Air conditioning and refrigeration

CIBSE Guide B3

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CIBSE

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Note from the publisher

This publication is primarily intended to provide guidance to those responsible for the design, installation, commissioning, operation and maintenance of building services. It is not intended to be exhaustive or definitive and it will be necessary for users of the guidance given to exercise their own professional judgement when deciding whether to abide by or depart from it.

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Foreword

Guide B provides guidance on the practical design of heating, ventilation and air conditioning systems. It represents a consensus on what constitutes relevant good practice guidance. This has developed over more than 70 years, with the Steering Groups for each edition of the Guide expanding and pruning the content to reflect the evolution of technology and priorities.

Since the last edition of Guide B in 2005, the European Energy Performance of Buildings Directive has been introduced. This requires national building energy regulations to be based on calculations that integrate the impact of the building envelope and the building services systems, formalising what was already recognised as good design practice. In addition, the use of voluntary energy efficiency and sustainability indicators has increased.

These changes have influenced the content of Guide B, but the emphasis remains on system design. The guidance in Guide B is not in itself sufficient to cover every aspect of the effective design of HVAC systems. Energy (and carbon emission) calculations will also be needed, and a range of other environmental criteria may specified by the client. These may, for example, include whole-life costing or assessments of embodied energy or carbon. The balance between building fabric measures and the energy efficiency of HVAC systems is important, as is the balance between energy use for lighting and for heating, ventilation and cooling. More detailed information on energy efficiency and sustainability can be found in Guides F and L respectively. The Guide does not attempt to provide step by step design procedures: these can be found in appropriate textbooks.

Structure of the Guide

Guide B deals with systems to provide heating, ventilation and air conditioning services, and is divided into several chapters which are published separately. It will usually be necessary to refer to several — perhaps all — chapters since decisions based on one service will commonly affect the provision of others.

— Chapter B0: *Applications* focuses on how different types of building and different activities within buildings influence the choice of system. This chapter is not available in printed form, but can be downloaded from the CIBSE website. For many activities and types of building, more detailed design information is available in specialist guidance.

Chapters B1 to B4 address issues relating to specific services. There are usually several possible design solutions to any situation, and the Guide does not attempt to be prescriptive but rather to highlight the strengths and weaknesses of different options.

— B1: *Heating*, including hot water systems and an annex on hydronic systems, which is also applicable to chilled water systems
— B2: *Ventilation and ductwork*
— B3: *Air conditioning and refrigeration*
— B4: *Noise and vibration control for building services systems* (applicable to all systems)

When all chapters have been published, an index to the complete Guide B will be made available.

The focus is on application in the UK: though many aspects of the guidance apply more generally, this should not be taken for granted. The level of detail provided varies: where detailed guidance from CIBSE or other sources is readily available, Guide B is relatively brief and refers to these sources. Examples of this are the treatment in the Guide of low carbon systems such as heat pumps, solar thermal water heating and combined heat and power. On-site energy generation such as wind power and photovoltaics are not covered.

Regulatory requirements are not described in detail in the Guide — the information varies between jurisdictions and is liable to change more rapidly than the Guide can be updated. Instead, the existence of regulations is sign-posted and their general scope explained. Sometime example tables are shown, but readers should note that these are simply examples of the type of requirement that is imposed and may not be current.

While there is some discussion of relative costs, no attempt is made to provide detailed cost figures as these are too project-specific and variable with time and location.

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3.1 Overview

Section 3.1 sets out areas common to both air conditioning and refrigeration. Sections 3.2 and 3.3 deal with air conditioning and refrigeration respectively.

The Guide describes requirements and provides guidance on system selection with respect to generic building systems and relates primarily to office air conditioning systems. Guidance on specific requirements for other building types is provided in CIBSE Guide B0: *Applications* (CIBSE, 2016a).

3.1.1 General

Air conditioning and refrigeration systems contribute to effective building performance and occupant satisfaction. For the purposes of this guide refrigeration is defined as the process of removing heat and therefore includes various forms of 'free' and environmental or low-energy cooling.

Designers must be conscious of the requirement for air conditioning and refrigeration to be designed with efficiency in mind to ensure that energy usage/cost and thereby carbon emissions are minimised.

Air conditioning and refrigeration installations must be designed in accordance with local regulations, for example Part L of the Building Regulations (England and Wales) (TSO, 2010), the F-Gas Regulation (EU, 2014) and the Pressure Equipment Directive (PED) (EC, 1997).

Air conditioning and refrigeration systems must be designed so that they can be controlled effectively to deliver the required comfort conditions in an efficient manner.

Whole-life costs (section 3.1.3) must be taken into account, because air conditioning and refrigeration systems are required to operate throughout the life of a building. Operating costs invariably exceed initial capital expenditure savings.

This guide is intended to be used by practising designers who hold a basic knowledge of the fundamentals of building physics and building services engineering.

3.1.2 Energy efficiency and minimising carbon emissions

There has been a significant increase in energy consumption related to air conditioning in recent years. To meet the overall UK targets for reduced carbon emissions, it is particularly important to give detailed attention to the energy efficiency of air conditioning systems.

Governments are increasingly committed to significantly reducing carbon emissions. The UK has a target of 34 per cent cut based on 1990 levels by 2020. Schemes include:

— The Climate Change Levy came into force in the UK in 2001 as a mechanism to encourage non-domestic building users to improve the efficiency of their systems. The levy was originally set at 0.15 p/kW·h for gas and 0.43 p/kW·h for electricity. At present the levy is offset by a cut in employers' National Insurance Contributions and 100 per cent first-year Enhanced Capital Allowances.

— The CRC Energy Efficiency Scheme, in which organisations consuming electrical power in excess of a benchmark value (6000 MW·h in 2011/2012) must purchase carbon credits from the Government in arrears at a nominal cost per tonne of carbon dioxide (£12 in 2011).

— Enhanced Capital Allowances for investment in energy efficient products.

— The Energy Performance of Buildings Directive (EPBD) (EU, 2010), which introduces requirements for energy efficiency and inspection.

— In England and Wales, Part L of the Building Regulations (TSO, 2010), which sets significantly more demanding targets for energy conservation aspects of buildings than has hitherto been the case.

The combined effect of these regulatory measures is expected to be a significant improvement in energy performance, certainly in new buildings and those undergoing refurbishment.

Designers should consider adopting energy efficient components and systems (e.g. high-efficiency chillers, heat pumps, heat recovery from cooling systems, free-cooling water-cooled heat rejection, etc.) where possible. These technologies may be at additional capital cost but can be demonstrated to achieve a payback in terms of whole-life cost or can assist in achieving energy benchmarks (e.g.

Building Research Establishment Environmental Assessment Method (BREEAM), Leadership in Energy and Environmental Design (LEED) and Estidama).

3.1.3 Whole-life cost

Whole-life cost (see OGC, 2007; Allard, 2001) or lifetime cost refers to total of investment, running, maintenance and disposal cost over the life of the installation. Analysis frequently shows that running costs are dominant, and this is a strong incentive for reducing energy consumption at the design stage.

Proper design and maintenance of air conditioning and refrigeration systems can significantly reduce the whole-life costs of the system. Costly modifications and alterations can be avoided by ensuring that the system requirements are properly defined and the design fully addresses the requirements.

Buildings have to adapt and change in response to business needs. Taking account of this at the design stage can also help to ensure that the system is designed to enable such adaptations to be carried out in the most cost-effective manner, again reducing the whole-life costs of the system.

3.1.4 Establishing key performance requirements

3.1.4.1 General

The design team should expect to agree requirements with the client at the outset of the project to optimise the choice of strategy. If the client is unable to advise on the precise needs, they must at least be made aware of any limitations of the chosen design.

The design team should also be able to advise the client of the cost implications (on a whole-life basis if requested (OGC, 2007)) of meeting their stated requirements.

Table 3.1 Considerations for establishing performance requirements

Issue	Requirement/comment
Client brief	To be developed in the context of the other issues
Integrated design	Co-ordinated approach by the architect and other specialists from outline design
Energy/environmental targets	Regulatory requirements, specifications or appropriate advice from the design team required, see section 3.1.4.1 Compatibility with indoor environment standards
Indoor environmental standards	Existing standards or appropriate advice from the design team required (see chapter 2 of CIBSE Guide A: *Environmental design* (CIBSE, 2015a)) Areas or objects with special requirements
Provision of controls	Individual, local, team, zone or centralised Control tolerances (e.g. of temperature, humidity, air quality, airflow) Interaction of the end user with the building services Basis of control, e.g. temperature, CO_2, CO or other
Demands of the building occupants	Building use (refer to the applications section) Work patterns Space planning—flexibility, cellular and/or open plan Occupancy numbers and anticipated maximum occupancy Average occupancy density and any areas of high or low density Internal loads—dependent on space use and application Anticipated diversity of internal loads
Investment criteria	Constraints imposed by 'letability' requirements
Value engineering and whole-life costs	Understanding of the client's priorities towards capital cost and issues of whole life costs (Allard, 2001) Requirements for calculations to be carried out on systems or system elements and the basis for these calculations (OGC, 2007) Has the client been involved in discussions of acceptable design risk? The importance of part-load performance
Resilience	Specified levels of resilience (application dependent)
Maintenance requirements	Ability of client to carry out, or resource, maintenance Can maintenance take place in the occupied space? Requirement for 'standard' or 'familiar' components
Associated systems	Implications of any particular requirements, e.g. fire, security, lighting and acoustic consideration
Security	Restrictions on size and location of any openings
Future needs	Adaptability, i.e. future change of use Flexibility, i.e. future changes in work practices within the current building use Acceptable design margins: it is important to distinguish, in collaboration with the client, between design that is adequate for current requirements (which may not be currently accepted best practice) and design that makes sensible agreed allowances for future changes and over-design (CIBSE Guide A (2015a))
Aesthetic considerations	The need for system concealment Restriction on placement of grilles, diffusers, etc.. Restrictions imposed by local authorities, building listing, etc..
Procurement issues	Time constraints Programming constraints, particularly for refurbishment

Requirements may subsequently be adjusted over the course of the project to meet financial constraints or changing business needs. The design team must also be able to advise on the impact of any such changes on building performance.

An appreciation of the issues shown in Table 3.1 (above) is an essential part of the briefing process.

3.1.4.2 Energy and environmental targets

The chosen air conditioning strategy influences, or is influenced by, the setting of appropriate energy and environmental targets and the selection of suitable indoor environmental standards. For example, meeting a stringent energy target may not be compatible with the provision of close control of temperature and humidity.

Initial agreement should be reached on the standards required. Checks should be carried out continuously by the design team to ensure that the implications of any changes made during design, construction or subsequent fit-out are understood and mutually acceptable.

Documents are available to assist in setting energy and environmental targets for a number of domestic and non-domestic building types, including the following.

— CIBSE Guide F (2012b) provides energy benchmarks and target assessment methods for dealing with banks and similar agencies, hotels, offices and mixed-use buildings. Table 3.2, reproduced from CIBSE Guide F, provides energy usage benchmarks for ‘good practice’ and ‘typical’ performance, based on two generic air-conditioned office classifications.

— CIBSE TM44: *Inspection of air conditioning systems* (2012a) provides guidance for air conditioning inspectors carrying out air conditioning systems inspections in accordance with Article 15 of the Energy Performance in Buildings Directive (EU, 2010) (a current legal requirement, which is required at least every five years).

— CIBSE TM46: *Energy benchmarks* (2008) describes the statutory building energy benchmarks prepared to complement the Operational Rating procedure developed by the Department for Communities and Local Government (DCLG) for Display Energy Certificates for use in England, Wales and Northern Ireland under the Energy Performance of Buildings (England and Wales) Regulations 2007 (TSO, 2007a). It describes the benchmarks and explains the approach to their development and use.

— CIBSE Guide F: *Energy efficiency in buildings* (2012b) covers both the energy requirements committed by the design and the energy costs in use, as design and management cannot be separated.

— The Energy Consumption Guides, published under the Government’s Energy Efficiency Best Practice Programme (discontinued), provide energy benchmarks and targets for industrial buildings, offices, public houses, hotels, hospitals, nursing and residential homes, and other non-domestic sectors. Many of these are available on the CIBSE website*.

— Building Maintenance Information’s report *Energy Benchmarking in the Retail Sector* (BMI, 1999) provides energy benchmarks within the retail sector.

— BREEAM (www.breeam.org) covers various schemes, e.g. industrial units, offices, superstores and supermarkets.

— BSRIA’s COP 6/99: *Environmental code of practice for buildings and their services* (1999) provides a guide to, and case studies on, the consideration of environmental issues during the procurement process; this guidance is applicable to all types of property.

— The Simplified Building Energy Model (SBEM) is a calculation developed by the Building Research Establishment (BRE) that can be used for:
  - Energy Performance in Buildings Directive calculations

* http://www.cibse.org/knowledge/archive-repository/archive-repository

Table 3.2 Office system and building energy benchmarks (CIBSE, 2012b)

Fuel/application	Annual delivered energy for stated office classification / kW·h·m^{-2}			
	Type 3*		Type 4†	
	Good practice	Typical	Good practice	Typical
Fossil fuels:				
— gas/oil heating and hot water	97	178	107	201
— catering (gas)	0	0	7	9
Electricity:				
— cooling	14	31	21	41
— fans, pumps and controls	30	60	36	67
— humidification	8	18	12	23
— lighting	27	54	29	60
— office equipment	23	31	23	32
— computer room	14	18	87	105
Total gas or oil	97	178	114	210
Total electricity	128	226	234	358

* Standard air conditioned
† Prestige air conditioned

- CO_2 emission calculations for Part L of the Building Regulations (England and Wales) (TSO, 2010) and equivalent regulations in Scotland, Northern Ireland, the Republic of Ireland and Jersey
- generation of Energy Performance Certificates.

— *Improving the energy efficiency of our buildings* (DCLG, 2008).

3.1.5 Cooling loads

One of the key factors in the design of an air conditioning and refrigeration system is to meet the cooling load. The magnitude of cooling load will have a significant effect on the design of the cooling system, as air conditioning and refrigeration equipment is tailored towards meeting specific ranges of cooling load.

Cooling loads are defined in chapter 5 of CIBSE Guide A: *Environmental design* (2015a), and include loads associated with:

— external gains determined by infiltration, ventilation, conduction and solar radiation

— internal gains determined by lighting, equipment, occupants, fabric and thermal mass.

Building design and operating requirements have a significant impact on cooling loads. The building services engineer should play a key role in the design of buildings to minimise cooling loads and thereby minimise power consumption and carbon emissions.

3.2 Air conditioning

3.2.1 Introduction

Air conditioning is a form of indoor air treatment, which includes filtration and control of temperature and humidity.

The strategic considerations that apply to distribution of the treated air to the occupied space are discussed in section 3.2.2.

This is followed in section 3.2.3 by a description of the systems that prepare the treated air for distribution.

Finally, section 3.2.4 deals with the equipment used to facilitate these processes.

Methods for generating the cooling itself are presented in section 3.3 on refrigeration.

The overall process of design development, from the initial outline design through to system selection and detailed equipment specification, is summarised schematically in Figure 3.1. A flow chart to aid choice of system type is given in Figure 3.2.

3.2.1.1 Comfort cooling and air conditioning

Comfort cooling may be defined as the use of mechanical cooling to maintain control over the maximum air temperature achieved in the space. As a consequence, there may be some incidental dehumidification of the supply air.

Air conditioning involves control over the humidity within the conditioned space as well as temperature control.

A further refinement is ‘close control’ air conditioning. There are many definitions of what is meant by ‘close’. For example (see Figure 3.2), ‘tight’ temperature control is defined as ±1 K (air temperature) and ‘close’ control of humidity as better than ±10 per cent. Specific circumstances may require more precise control, e.g. ±5 per cent RH and ±1 K, or ±2 per cent RH and ±0.5 K in critical areas. It is therefore important for the client and designer to have agreed these parameters.

A broad categorisation of comfort cooling and air conditioning systems is given in Table 3.3; the performance characteristics of individual systems within the broad categories will vary greatly. It is also possible for systems to differ in whether, for example, they:

— operate as single or multiple zone

— employ full fresh air or recirculation

— have humidification or dehumidification potential.

The choice of the optimum system will depend on the particular circumstances and client’s own priorities and, in the case of a refurbishment project, the existing building services. Table 3.4 provides some assessment criteria that might be used to compare systems from the perspective of both the client and the design team. These may be supplemented to suit the context.

3.2.2 Strategies

3.2.2.1 Introduction

This section summarises the key issues and performance targets that need to be addressed during design and is not intended to provide step-by-step design guidance. The guidance contained in this section should be read in conjunction with CIBSE Guide A: *Environmental design* (2015a) and CIBSE Guide F: *Energy efficiency in buildings* (2012b). For details of refrigeration methods, see section 3.3.

3.2.2.2 Room air distribution strategies

3.2.2.2.1 Room air diffusion: criteria for design

The effectiveness of air conditioning systems depends on the method by which supply air is introduced to, and vitiated air removed from, the space.

The parameters that influence the quality of the air at any point in the room are the following:

— air supply velocity

— temperature differential between the room and supply air

— air quality (cleanliness)

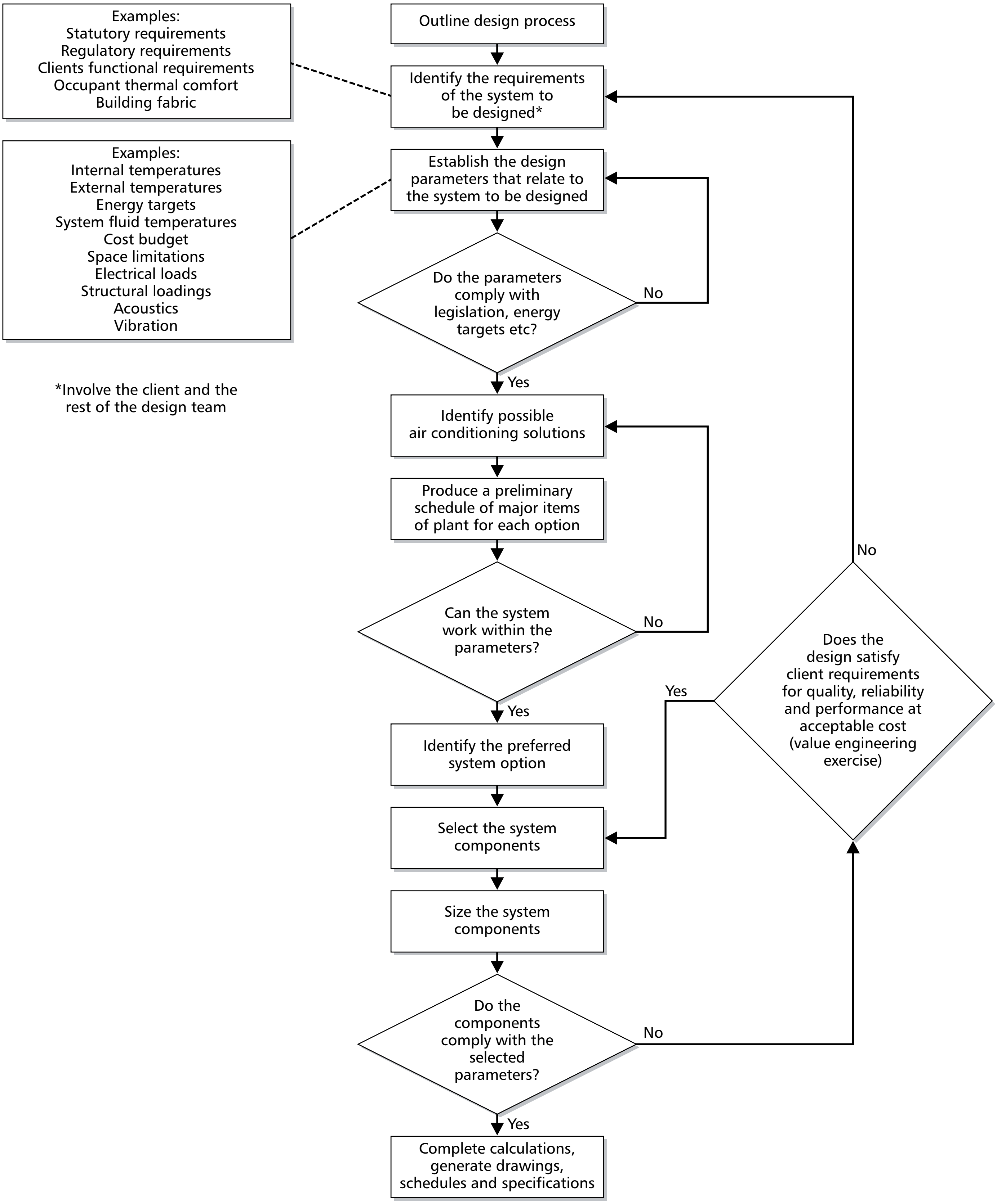


Figure 3.1 Outline design process; air conditioning

- position of the air supply terminals
- room shape and geometry, including projections
- position, size and shape of all sources and sinks for heat and contaminants
- temperature of any heat sources and sinks
- rates of evolution and sorption of contaminants
- other factors influencing air movement, such as movement of the occupants and machinery, and air infiltration.

If terminal devices are poorly selected or positioned, this can result in draughts, stagnation, poor air quality, inappropriate mixing, large temperature gradients and unwanted noise. The terminal type and layout may be affected by architectural or structural considerations,

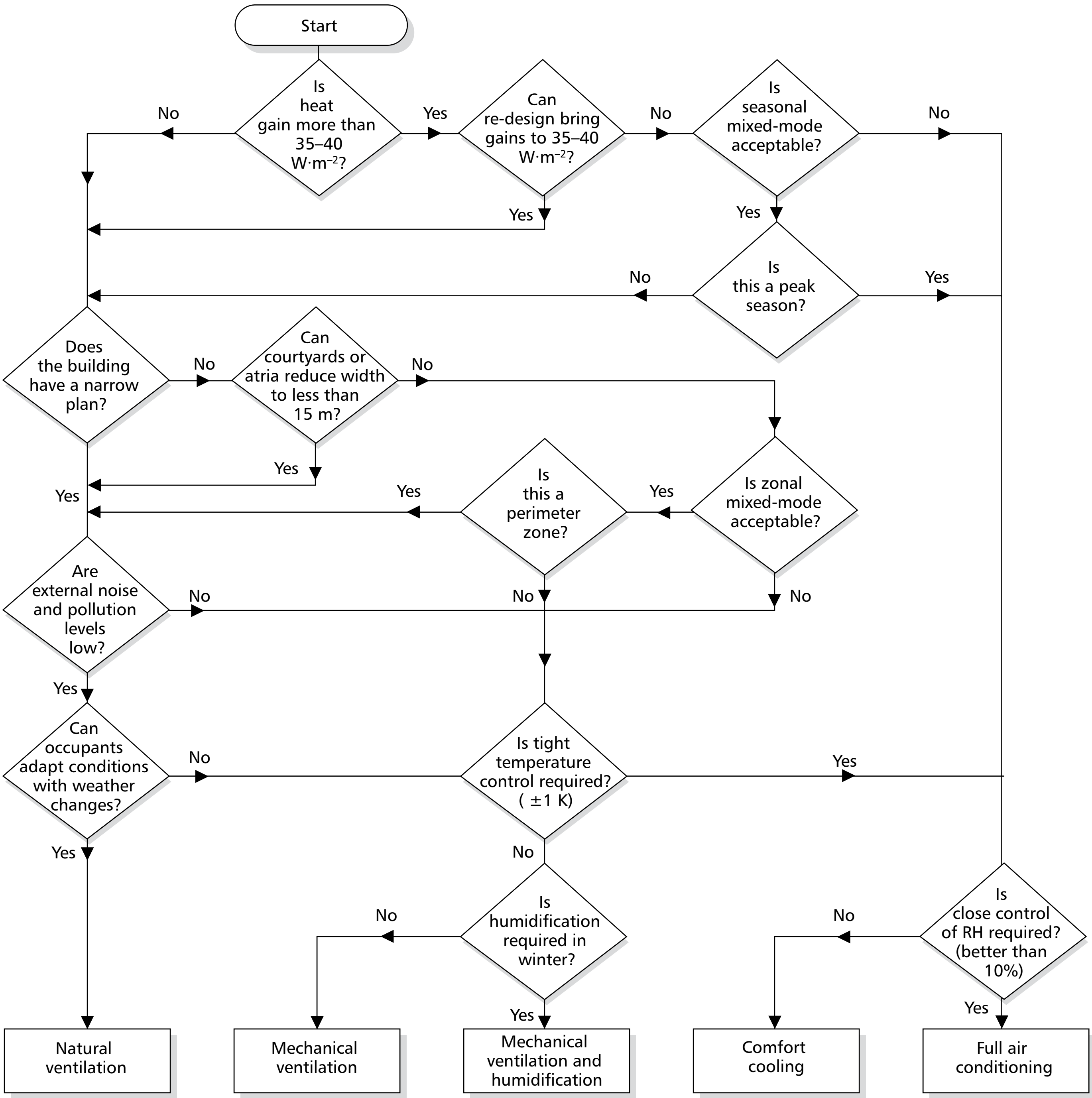


Figure 3.2 Selecting an air conditioning strategy

Table 3.3 Broad categorisation of comfort cooling and air conditioning systems

Type	Description	Typical systems
All air	Employing central plant and ductwork distribution to treat and move all the air supplied to the conditioned space. These systems can have fine tuning of the supply temperature or volume occurring at the terminals.	Variable air volume (VAV) and its variants, dual-duct, hot-deck/cold-deck. Constant air velocity, constant flow variable temperature.
Air/water or air/refrigerant	Usually employing central plant to provide fresh air only, terminals being used to mix recirculated air with primary air and to provide fine tuning of the room temperature	Fan coils, chilled beams, variable refrigerant flow (VRF) units, induction units, reversible heat pumps, chilled ceilings.
Unitary	Small-scale versions of single-zone systems within packaged units	Fan coils, reversible heat pumps, split systems, room air conditioners.

Table 3.4 Possible system assessment criteria

Criterion	Comment
Air conditioning and cooling performance	System efficiency and power consumption Risk of draughts Noise generation Maximum cooling load that can be handled Ability to be zoned Ability to cope with frequent variations in load Ability to cope with semi-permanent variations in load Potential for use in mixed mode systems
Control	Suitability for precise temperature control Suitability for precise humidity control
Design	Availability of guidance to assist in system design Ease of design Availability of performance data
End-user acceptability	Availability of end-user control
Robustness to poor design	Familiarity of client with proposed system Level of tailoring required for standard system to suit particular context
Indoor air quality	Ability to provide an appropriate quality of indoor air, free from contaminants and odours
Economic performance	Capital costs Life-cycle costs Energy costs
Installation, commissioning and handover	Installation time Ease of installation Ease of commissioning
Flexibility	Ability to cope with changes in space layout Ability to be upgraded
Reliability	Ability of the air conditioning systems to deliver required volumes and quality of air with no more than the specified levels of downtime
Ease of maintenance	Ease of cleaning Ease of replacement Requirement for maintenance in the occupied space Risks associated with transport of water or refrigerant around the building Risk of Legionnaires disease
Integration	Impact on floor-to-ceiling height Minimum plant space requirements Impact on distribution Need for high levels of airtightness Encroachment into workspace Constraints imposed on other services Constraints imposed by other services
Other issues	Ease of procurement Carbon emissions Refrigerant usage Aesthetics

however it is for the system designer to ensure that particular room air-diffusion requirements form part of the integrated/co-ordinated building design and/or structure (e.g. floor supply).

The occupants' perception of the effectiveness of the system will normally be determined by the:

— velocity of air adjacent to any uncovered or lightly covered skin (e.g. neck and ankles)

— temperature of airstream in relation to that of the still air adjacent to other parts of the body

— level of activity

— occupants' clothing

— air quality (cleanliness) in the breathing zone

— subjective individual temperature requirements

— appearance and positioning of any ventilation devices or openings

— system noise and vibration

— intrusion upon occupied space (wall, floor areas taken up).

The above are discussed in detail in section 1.4 of CIBSE Guide A: *Environmental design* (2015a).

BS EN ISO 7730: *Ergonomics of the thermal environment. Analytical determination and interpretation of thermal comfort using calculation of the PMV and PPD indices and local thermal comfort criteria* (BSI, 2005) recommends that during cooling the mean air velocity should be less than 0.25 $m{\cdot}s^{-1}$ for moderate thermal environments with light, mainly sedentary, activity and that during heating it should be less than 0.15 $m{\cdot}s^{-1}$. No minimum velocity is suggested, although stagnant zones could result in temperature gradients between the ankle and the neck greater than the 3 K recommended. It is likely that sufficient air movement will be generated by other means (e.g. convection).

The occupied zone can be defined as a region, the outer limits of which are described by an envelope 1.8 m from the floor and 0.15 m from the walls. However, in the case of low-level supply terminals (e.g. displacement ventilation) the occupied zone is any region where the occupants are likely to linger for significant periods. For desk terminals, mixing occurs over the desk surface and for seat-back terminals, mixing occurs in the regions above and between the seats.

An assessment of predicted percentage dissatisfied (PPD) (BSI, 2005) for a wide range of activity levels, clothing, body temperatures and velocities shows that, even at low activity levels, velocities as high as 1.0 $m{\cdot}s^{-1}$ can be acceptable in offsetting high temperatures. This technique has been applied to the concept of spot cooling in some industrial applications (Hwang et al., 1984) whereby heat stress in the workers is avoided by keeping the local conditions below an agreed value of wet bulb temperature.

3.2.2.2.2 Ventilation efficiency

Uneven temperature distribution and contaminant concentrations can occur within occupied zones due to local convection currents and the uneven distribution and mixing of contaminants within a space. If heat transfer and fresh-air provision can occur, the condition of the space above this zone is usually unimportant. Displacement ventilation systems exploit this concept (see section 3.2.2.2.5 'Displacement ventilation'). Conventional air conditioning systems, however, use dilution ventilation whereby mixing occurs outside the occupied zone and, under ideal conditions, all the air in the space is at the same temperature and of the same quality. The efficiency of the ventilation therefore depends on effective local removal of heat and contaminants from the space and the total energy requirements of the supply and extract systems required to achieve this. Careful account needs to be taken of potential contaminant sources within the occupied space, which will reduce the efficiency of the ventilation system.

3.2.2.2.3 Air distribution (HEVAC, 2013)

Air distribution is covered in detail in CIBSE Guide B2 (2016b).

In general, air can be supplied to a space in a number of ways (grilles, louvres, diffusers and air distribution textile ducts).

The principal types are diffusers and perpendicular jets. Airflow patterns for both types of terminal are strongly dependent upon the presence or absence of the Coanda effect (see below).

Diffusers may be radial, part-radial or linear and normally utilise the Coanda effect and or/swirl to reduce the risk of excessive room air movement.

A perpendicular jet is formed by discharging air through grilles, louvres, nozzles or any opening that allows perpendicular flow. Direction and spread adjustment can be provided using blades and/or swivel adjustment.

Supply air terminal devices can be incorporated into any room surface, e.g. ceiling (flat or sculptured), floor, wall (high or low level), desk top, seat back or under seats. Air terminal devices in other types of equipment are considered in section 3.2.4.2. Further guidance can be obtained from *Guide to air distribution technology for the internal environment* (HEVAC, 2013).

(*a*) Air terminal phenomena

Many studies of jets and their effect on room air movement have been undertaken. Figure 3.3 shows the predicted airflow patterns for various types and positions of air terminal device (ASHRAE, 2009).

It should be noted that these patterns are based on stylised terminals. For predictions of air movement appropriate to specific air terminals, the manufacturers' data must be consulted. For non-standard situations it may be necessary to model room air movement using a mock-up sample room. In most cases it will be necessary to allow for on-site adjustment of airflow pattern, either during commissioning or during operation by the occupant (e.g. desk-mounted terminals).

(*b*) Air diffusion terminology

ISO 3258: 1976: *Air distribution and air diffusion. Vocabulary* (ISO, 1976) (withdrawn) gave definitions and standard terminology used in connection with air movement. Some of the more important parameters are listed below.

— *Throw*: a free jet having a given momentum on discharge will establish velocity profiles known as isovels, the shape of which depends on the geometry of the terminal, the temperature of the jet and any other disturbing influences. The velocity decays with increasing distance from the terminal. Throw is defined as the distance from the terminal (measured perpendicular or parallel to the face of the air terminal device depending on the predominant direction of flow) to the 0.5 $m{\cdot}s^{-1}$ isovel.

  Lower velocities are normally required for air entering the occupied zone — typically 0.25 $m{\cdot}s^{-1}$ for cooling and 0.15 $m{\cdot}s^{-1}$ for heating. Reference should be made to manufacturers' literature for throw data and recommended mounting distances from solid surfaces and neighbouring terminals.

  The maximum throw for an air terminal device depends upon the characteristics of the device, the mounting height and the influence of neighbouring devices.

— *Spread*: the spread of a horizontal jet is defined as the width of the 0.5 $m{\cdot}s^{-1}$ isovel. Note that most manufacturers give the width of the 0.25 $m{\cdot}s^{-1}$ isovel, which is generally of more use to the designer.

— *Drop*: the drop is defined as the vertical distance from the centre-line of the terminal to the bottom edge of the 0.25 $m{\cdot}s^{-1}$ isovel.

(*c*) Entrainment, mixing and boundaries

Frictional forces cause a momentum transfer to take place between the jet and adjacent room air, which draws the room air in the same direction as the jet. The jet expands with distance from the terminal as it entrains adjacent room air. Hence kinetic energy is expended in creating turbulence, which transfers thermal energy and assists the dilution of contaminants. This process of diffusion may be enhanced by the introduction of a rapidly expanding jet and still further by imparting a swirling motion to the jet.

A jet that is constrained by the walls of a room, such as a full width slot, will entrain less room air and expand more slowly than a free conical jet (HEVAC, 2013; ASHRAE, 2013a).

(*d*) Effect of temperature differential

Figure 3.4 shows that a jet that is not influenced by the proximity of a solid surface follows a path that is a function both of velocity and temperature. A warm jet tends to rise until it attaches itself to a horizontal surface, whilst a cold jet falls. Care must be taken to ensure that this does not lead to unacceptable temperature gradients in the occupied zone during heating and excessive air velocities during cooling. The terminal must be mounted such that the 0.25 $m{\cdot}s^{-1}$ isovel does not enter the occupied zone.

The difference in temperature between the supply and return air may be greater than that between the supply air and the occupied zone, particularly with a low-level supply designed to encourage high-level stratification. This temperature difference is related to sensible heat gain and supply air mass flow, as follows:

$$q_s = m \, C_{ph} \, \Delta T$$

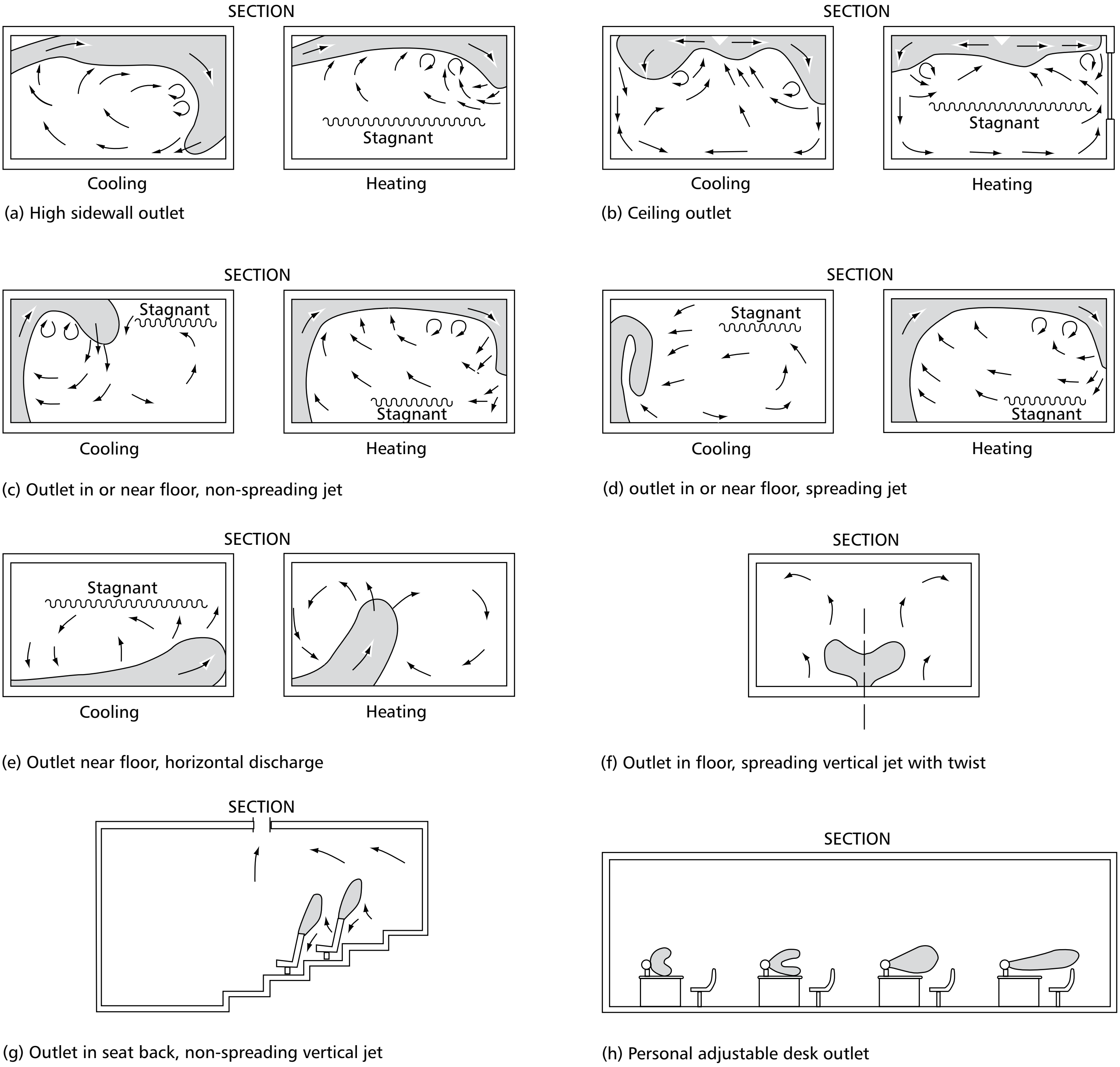


Figure 3.3 Predicted airflow patterns (adapted from ASHRAE, 2009)

where q_s is the total sensible heat gain (kW), m is the mass flow rate of supply air (kg·s^{-1}), C_{ph} is the specific heat capacity of the air and water vapour mixture (kJ·kg^{-1}·K^{-1}) and ΔT is the room air to supply air temperature differential (K).

Therefore, the mass flow rate, and hence the cost of air handling, will depend upon the temperature difference chosen by the designer. This decision will also be influenced by the evaporator temperature and the level of control of humidity. For example, a displacement system with low-level input can supply air at 18 °C with a temperature difference of about 10 K. This can be achieved with high evaporator temperatures and correspondingly low compressor power. However, high-level humidity control will suffer unless the supply air is over-cooled and reheated, normally an undesirable combination at peak load. Alternatively, a permanent bypass around the cooling coil can be provided and, if motorised dampers are incorporated at the coil face and in the bypass, part-load control supply temperature can be achieved by damper modulation.

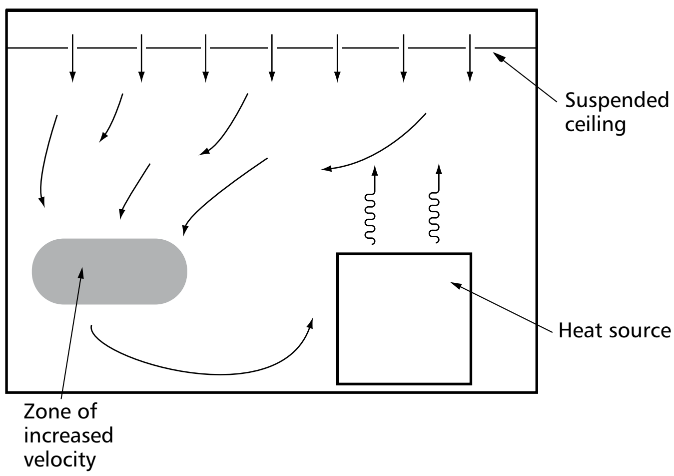


Figure 3.4 Effect of room convection currents

For comfort applications, air change rates are unlikely to exceed 10 air changes per hour, corresponding to a cooling temperature differential of 8–12 K. A free horizontal jet from a rectangular grille is likely to create down-draughts if providing more than 8 ACH with a cooling temperature differential greater than 8 K.

A maximum cooling differential of 10 K can be applied when:

— the presence of the Coanda effect (see below) is assured
— for a free jet, mixing of supply air with room air outside the occupied zone can be assured without promoting discomfort.

Table 3.5 gives general guidance on the maximum air change rates that can be achieved using various air terminal devices supplying air with a cooling temperature differential of 10 K.

Table 3.5 Typical maximum air change rates for air terminal devices

Device	Air change rate / h^{-1}
Sidewall grilles	8
Linear grilles	10
Slot and linear diffusers	15
Rectangular diffusers	15
Perforated diffusers	15
Circular diffusers	20

If sufficient mixing between terminal and occupants cannot be guaranteed (e.g. with low-level supply), then the minimum supply temperature of 18 °C applies, with a temperature differential in the occupied zone of 4–5 K. However, the cooling temperature differential is ultimately determined by the maximum exhaust air temperature (Sodec, 1984), see Table 3.6.

Table 3.6 Typical cooling temperature differentials for various applications

Application	Maximum temp. differential, K
High ceiling >2.8 m (large heat gains/low level air input)	12
Low ceiling <2.4 m (air handling luminaires/low level air input)	10
Low ceiling <2.4 m (downward discharge)	5

The larger temperature differential indicated for high ceilings is possible due to the smaller influence of ceiling temperature on the mean radiant temperature experienced by the occupants.

Downward discharge is generally only satisfactory for very high air change rates, and hence small temperature differentials, or where room convection is not significant. An exception is the specific case of split systems, where temperature differences can be as high as 20 K. Particular care is therefore needed in their specification and application.

High-level supply jets must overcome the buoyancy forces in the room air generated by heat emitters, solar gain, occupants etc., whereas low-level input utilises these forces to assist the supply jet. For this reason, low-level supply is most satisfactory for applications with high room gains and high ceilings. For low ceilings, the radiant heating effect of the ceiling itself may be significant. This may also be a problem where the ceiling void is used as an exhaust air plenum, carrying air heated by air-handling luminaires.

Free descending jets are not recommended for normal use, since the low velocity approaching the occupied zone would cause instability. This could result in localised high velocities due to deflection by convective forces elsewhere in the room (see Figure 3.4 above). An exception is the case of laminar downflow cleanrooms (BS EN ISO 14644: *Cleanrooms and associated controlled environments* (BSI, 1999–2013)) where an even velocity across the full area of 0.4 $m \cdot s^{-1}$ should be maintained from ceiling to floor. However, even in these circumstances, sources of extremely buoyant upflow should be avoided.

(*e*) Coanda effect

When a jet is discharged from a terminal device adjacent and parallel to an unobstructed flat surface, the jet entrains air from one side only resulting in deflection of the axis of the jet towards the surface. This phenomenon, known as the Coanda effect, is due to frictional losses between the jet and the surface.

The effect diminishes with distance from the terminals as increasing volumes of air are entrained from the room side of the jet, resulting in a reduction of jet velocity. However, the Coanda effect is maintained despite temperature differences between the jet and the room air. It is a critical factor influencing the selection and positioning of supply air terminals, particularly for rooms with low ceilings that have little space above the occupied zone in which mixing can occur.

If the Coanda effect is not present, the maximum throw for any terminal is reduced by approximately 33 per cent. The main factors that influence whether or not the Coanda effect will occur are:

— the distance between terminal and surface
— the width of jet exposed to surface
— the velocity of the jet
— the presence of projections and other disturbing influences.

The most important influence on the performance for side-wall terminals is temperature difference, i.e. buoyancy effects. For the usual range of temperature differences for cooling of 8–12 K, the opening should be within 300 mm of the surface to guarantee attraction. For systems designed to make use of the Coanda effect, provision should be made for on-site adjustment of the jet.

When a jet adheres to a surface, dust particles will be deposited on the surface leading to staining, hence supply air cleanliness is of paramount importance (see section 3.2.2.5). Cleanliness of the exhaust air is difficult to control and some staining of surfaces near to exhaust openings is inevitable.

Techniques exist for predicting the influence of projections, such as downstand beams and surface mounted luminaires, on a jet flowing across an otherwise smooth surface. An obstruction may cause the jet to separate completely from the surface, hence destroying the Coanda effect, or it may separate and join some distance downstream of the obstruction.

The critical distances at which these phenomena are likely to occur depend on the depth and shape of the obstruction and size of the supply opening. The influence of supply air to room air temperature differential is small but depends upon the extent to which mixing has occurred before the jet meets the obstruction.

Figure 3.5 shows the effect of a horizontal surface on a jet rising close to the vertical surface. The Coanda effect is maintained after the change in direction, provided that the velocity is adequate, particularly in the case of cooling jets, and that the temperature differential between supply and room air is not too large. Guidance for selecting optimum supply velocities and temperature is provided in CIBSE Guide A: *Environmental design* (2015a).

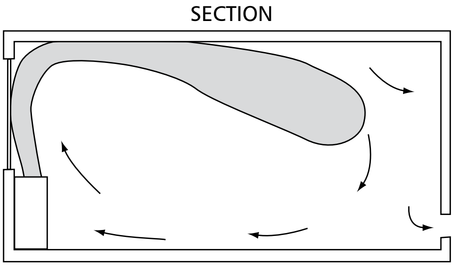


Figure 3.5 Effect of a horizontal surface on a jet

(*f*) Interaction between jets

Figure 3.6(a) shows possible room air velocity patterns for two jets directed towards each other along a 3 m high ceiling. The individual velocities of the two airstreams must not be greater than the 0.25 m·s^{-1} at the boundary of the jets otherwise discomfort may occur due to excessive down-draughts.

The envelopes of two converging jets may also interfere with each other, combining to form a single, wide jet with a maximum velocity at the new axis between the two jets (see Figure 3.6(b)). A similar phenomenon occurs with two jets moving in tandem (see Figure 3.6(c)). The downstream jet entrains and accelerates the decaying upstream jet and forms a wider jet with an axis further from the neighbouring surface. The cumulative effect of a series of single-way jets can result in a deep jet that intrudes into the occupied zone resulting in unacceptably high room velocities.

Figure 3.7 below shows examples of possible layouts for ceiling diffusers. The main problems likely to be encountered are those described above. Down-draughts may be encountered in areas marked 'X'. This problem may be eliminated by avoiding terminals with excessive throw, particularly in large spaces where stagnation between terminals is unlikely to occur. The layout shown in Figure 3.7(c) may cause convergence problems with long rooms.

For side-wall applications, the spacing of diffusers should be in accordance with manufacturers' recommendations. However, in the absence of such recommendations, Table 3.7 may be used in conjunction with throw and deflection data to determine the diffuser spacing. For a terminal mounted close to a wall, spacing should be halved to give the minimum distance from the centre line to the wall. Table 3.8 below (Sodec, 1984) indicates typical turndown limits for various types of fixed air terminal device.

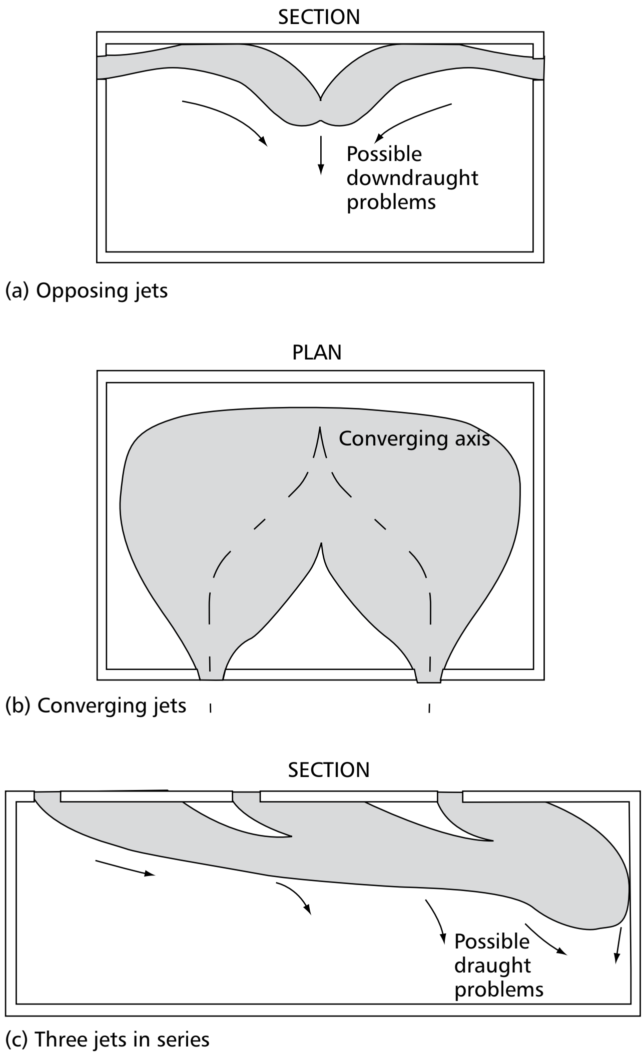


Figure 3.6 Room air velocity patterns; interactions between jets

Table 3.7 Data for determining spacing of ceiling diffusers

Deflection / deg.	Spacing / m	
0	0.20 L_x	0.33 L_y
22.5	0.25 L_x	0.50 L_y
45	0.30 L_x	1.0 L_y

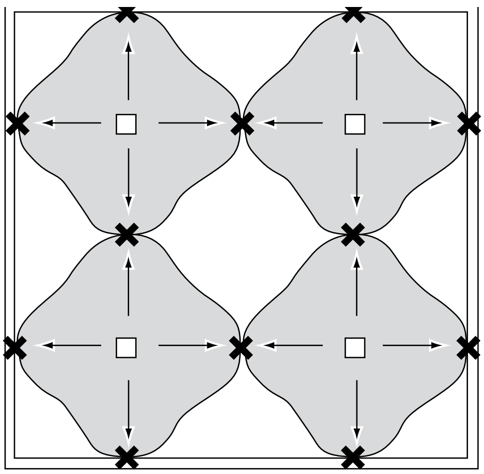
Note: L_x = throw (m) where axial velocity has decayed to 0.25 m·s^{-1}; L_y = throw (m) where axial velocity has decayed to 0.5 m·s^{-1}

3.2.2.2.4 Duct and plenum design

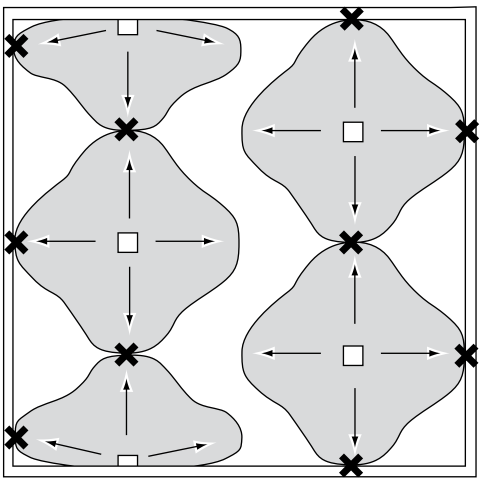
Air terminal devices will only perform as intended if the neck or inlet air velocity of the diffuser is uniform. If the duct connections and/or volume flow regulators created eddies at the terminal, the following problems may arise:

- unpredictable throw, spread and drop
- breakdown of Coanda effect
- high noise levels
- balancing being difficult or impossible.

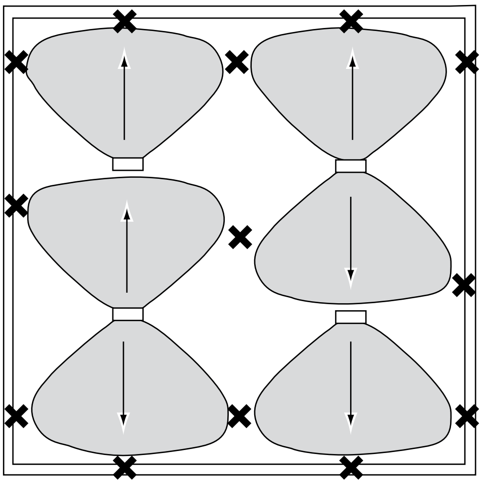
For design procedures for duct and plenum connections to various types of air terminal, refer to *Guide to air distribution technology for the internal environment* (HEVAC, 2013).



(a) Four-way ceiling diffusers, symmetrical layout



(b) Four-way ceiling diffusers, off-set layout



(c) One-and two-way ceiling diffusers, contra-flow layout

Figure 3.7 Supply terminal layouts for open plan spaces

If the ceiling void is to be used as an exhaust plenum, it is important to create a uniform negative pressure throughout the whole ceiling void to ensure even exhaust throughout all terminals. This is particularly important where exhaust is by means of air handling luminaires, the performance of which varies with airflow rate.

Table 3.8 Turndown limits for types of fixed air terminal device (Sodec, 1984)

Type of outlet	Maximum turndown / %
Ceiling mounted:	
— not using Coanda effect	50
— using Coanda effect	40
Floor-mounted outlets:	
— perforated plate and fixed bar grille	60
— free jet outlets	50
— outlets with swirl	40
Desk outlets:	
— linear type	50
— ball type	50
— outlets with swirl	40

Ceiling voids should be made as large as possible and, if obstructed by luminaires, ductwork etc. exhaust stub ducts should be provided to ensure even exhaust over the full ceiling area.

3.2.2.2.5 Displacement ventilation

In buoyancy-driven displacement-flow systems, air is supplied at a low velocity from low-level wall-mounted or floor-mounted supply air terminal devices directly into the occupied zone, at a temperature slightly cooler than the design room air temperature. The air from a wall-mounted terminal flows downward to the floor due to gravity and moves around the room close to the floor, creating a thin layer of cool air. Natural convection from internal heat sources, such as occupants and equipment, causes upward air movement in the room. The warm, contaminated air forms a stratified region above the occupied zone, which is then exhausted at high level. The height of this layer depends upon the relationship between the incoming airflow and the rate of flow in the plumes. The boundary will stabilise at a level at which these two flow rates are equal.

The airflow in displacement systems has both horizontal and vertical air movement characteristics. Horizontal air movement occurs within the thermally stratified layers that are formed between the upper (warm) and lower (cool) air layers in the room. Vertical air movement is caused by the presence of cold and warm objects in the space. Warm objects, such as people, create upward convection currents; cold objects, such as cold windows and walls, cause downward currents.

For given rates of fresh-air supply and pollutant discharge, the air quality in the occupied zone of a room with a displacement system can be higher than that using a mixed-flow air supply system. In displacement systems, air movement above the occupied zone is often mixed and it is when this mixed region extends down into the occupied zone that the air quality becomes similar to that in a mixed-flow system.

With a displacement system, a vertical temperature gradient is unavoidable. BS EN ISO 7730 (BSI, 2005) recommends a vertical temperature gradient for sedentary occupants of less than 3 K. This equates to approximately 3 $K \cdot m^{-1}$ if workers are assumed to be seated, although a limit of 1.8 or 2 $K \cdot m^{-1}$ is often proposed for offices (i.e. 5 K limit for a typical floor-to-ceiling height of 2.5 m). However, as 30–50 per cent of the overall supply-to-extract temperature difference occurs between the supply air and that at ankle level in the main space, a limiting difference between floor and ceiling height for typical office applications can be taken as 7–10 K. The supply air temperature is typically between 16 °C and 18 °C. It is also recommended that the limits of variation of temperature across the room should be within a temperature range of 3 K, i.e. ±1.5 K about the mean room air temperature.

A combination of near-floor temperatures below 22 °C and airflows in excess of 0.15 $m \cdot s^{-1}$ may cause discomfort due to cold feet, so occupants should be located a sufficient distance from diffusers. Equipment manufacturers should be consulted for detailed performance characteristics.

The zone around a supply air diffuser within which the supply air conditions have the greatest effect is labelled the near-zone. The permitted near-zone extent together with the maximum allowable comfort temperature at the near-zone perimeter for a given supply air temperature dictates the air volume per diffuser and its size. In an office, the near-zone may be 1 m; in a commercial application or in a foyer it may be 3 m. The maximum cooling load that can be delivered by a displacement system is therefore limited to 25 $W \cdot m^{-2}$ due to discomfort considerations (Sanberg and Blomqvist, 1989).

Displacement systems can be employed for many applications and building types. They are used in conjunction with chilled ceiling or chilled beam systems. However there are conditions under which the system is less effective than traditional mixed-flow systems. These include:

— where ceiling heights are low, i.e. <2.7 m

— where disturbance to room airflows is unusually strong

— where surface temperatures of heat sources are low, i.e. <35 °C.

(*a*) Displacement system devices

Pure displacement terminals

Pure displacement terminals aim to get air into the room with a minimum of eddies, room air mixing and temperature pick-up before it reaches the occupants. Hence there is a very small temperature difference between the supply air and that of the occupied zone. It is possible, under these circumstances, to maximise the use of outdoor air for free cooling and this may reduce the operation time of mechanical cooling.

Induction-type diffusers

Induction-type diffusers are intended to promote various levels of eddy mixing of the room air at the diffuser face. This allows lower supply air temperatures and hence marginally greater displacement cooling capacity. They have a larger approach temperature, generally require some mechanical cooling and impart a higher turbulent intensity with potential discomfort. A substantial diffuser open area is needed to obtain low velocities.

Swirl-type diffusers

Swirl-type diffusers introduce air at far higher velocities, promote full mixing in the occupied zone and disrupt buoyancy plumes. Thus they lose many of the displacement benefits. Horizontal diffusers need to be considered carefully in the light of their impact in terms of high velocities and sub-room temperatures near to occupants.

Effect of extract grilles

Extract grilles have a relatively minor impact on the system operation. The main consideration is their frequency with varying ceiling heights. The higher the ceiling, the greater the possible depth of the polluted air layer and so the air can travel further in order to reach a grille without gaining sufficient depth to encroach into the occupied zone.

(*b*) Control of displacement systems

The main forms of control are as follows:

— *Constant supply air temperature, constant airflow rate:* in which the supply air temperature is maintained constant at a design value selected to be at least 1 K below the required zone mean air temperature. Variations in heat gain will affect the temperature gradient within the space so that, provided the maximum heat gain does not create a temperature gradient in excess of comfort limits, acceptable conditions will be maintained. This form of control can be supplemented with a space sensor, which resets the supply air temperature set-point.

— *Constant supply air temperature, variable airflow rate:* the supply airflow rate may be adjusted to accommodate higher variations in heat load and maintain a substantially constant temperature gradient within the occupied zone. This adjustment can be automatically controlled to maintain a constant difference between the room air temperature and supply air temperature.

— *Variable supply air temperature:* this form of control is not as effective in displacement ventilation systems as it is in mixed-flow systems, because the supply air temperature required to maintain an acceptable mean room air temperature is not so directly related to internal heat gains.

Using a control system to maintain substantially constant thermal conditions within a room requires a temperature sensor located in a position that provides a reading that is representative of the occupied zone. In view of the vertical temperature gradients associated with displacement flow, the room air temperature sensor is best placed at about head height in a temperature neutral location free from significant draughts.

3.2.2.3 Mechanical air distribution

3.2.2.3.1 General

This section outlines general issues that should be taken into account during the selection and development of a mechanical air distribution strategy.

3.2.2.3.2 *Mechanical air distribution strategies*

The most common arrangement for supply and extraction of air in mechanical air distribution systems is a balanced supply and extract system in which the extract and supply systems are installed as two separately ducted networks. This offers the maximum flexibility by permitting contaminants to be removed at source and allowing for heat recovery.

Effective building sealing is required, as the system is designed to operate with a marginally positive pressure to reduce infiltration. The following methods are used to introduce air into the occupied space.

(*a*) Displacement systems

For details see section 3.2.2.2.5 'Displacement ventilation'.

(*b*) Mixing ventilation

This is based on the air being supplied into the room in a manner that creates sufficient turbulence for the contaminants within the space to be equally distributed. Contaminated air is extracted from the space and the supply dilutes the concentration of pollutants within the space. Mixing systems allow for recirculation, although the mixing within the space must be uniform. The system performance is not dependent upon room height or room layout. Air can enter the space either via the floor or via the ceiling.

(*c*) Floor-based supply

A floor-based supply is usually selected if raised floors are already in place, typically for IT systems. Floor-based systems allow the ceiling mass to be exposed. They may however restrict the furniture layout unless any underfloor units or distribution grilles are designed for easy relocation. Access for maintenance is, in theory, easy.

(*d*) Ceiling-based supply

Ceiling-based systems allow greater flexibility of furniture layout and also allow heat to be more efficiently extracted from light fittings.

3.2.2.3.3 *System considerations* (Liddament, 1996)

Increased system efficiency, i.e. reduced specific fan power, can be achieved by the following measures:

— Select efficient fans at or near to their maximum efficiency point.

— Select appropriate attenuation, filtration and heat recovery devices to reduce system pressure drops.

— Choose appropriate ductwork and system velocities to reduce system pressure drops.

— Vary the volume of air through the system, e.g. through the use variable speed fans. This can be achieved through variable speed drives.

— Ensure local extraction by the appropriate location of plant in order to minimise duct runs and hence fan power.

— Use intelligent zoning to avoid the system operating to suit the needs of one small area.

— Switch off systems when they are not in use or not required. Systems may run for longer than intended for various reasons, e.g. controls may have been overridden and not reset afterwards; automatic controls (e.g. frost thermostats or hidden hardware or software interlocks) may have switched on systems unnecessarily as a consequence of poor setting, calibration or programming. Suitable fault detection should be incorporated, e.g. by reporting the running hours of devices and systems during periods when they are programmed to be off.

— Control fan operation according to occupancy in either variable or constant volume systems.

— Log hours of operation of systems to identify if systems are operating unintentionally, particularly outside the occupied period. Anticipatory systems (e.g. for optimum start or night cooling) are prone to such behaviour.

— Take care to avoid parasitic loads that may increase energy consumption. Examples include:

  • heat recovery systems that break down unnoticed (or continue to operate when cooling is required)

  • free-cooling control systems that introduce the wrong proportions of outside air

  • unnecessary heating of air intended for night cooling.

Ideally, the performance of such systems should be automatically monitored against the design intentions. Alternatively, systems can be designed deliberately to allow such technical problems to be reported to a building management system (BMS) automatically.

The supply of air to a space can be controlled by a number of manual or automatic means. The general principles of these were considered above in 'Control of displacement systems'. The most popular options are:

— *CO_2 sensing:* useful in buildings where there are wide variations in the ventilation requirement, e.g. bingo halls, cinemas, theatres and meeting rooms.

— *Temperature sensing:* useful where it may be advantageous to increase the flow of air when conditions are favourable to take advantage of free cooling.

— *Humidity sensing:* fresh-air rates can be increased when internal humidity levels are too high, an option used for example in areas where moisture is produced, e.g. kitchens and bathrooms.

— *Occupancy sensing:* this enables systems to be switched off when rooms are not occupied.

3.2.2.4 Mixed mode air distribution

3.2.2.4.1 *Introduction*

Mixed mode ventilation is where both mechanical and natural ventilation are present. Selection criteria for mixed mode ventilation are detailed in CIBSE AM13: *Mixed mode ventilation* (CIBSE, 2000b); the following issues should be considered.

— *Costs:* capital and operating costs are highly variable. A balancing factor is to what extent supplementary mechanical systems have been installed.

— *Maintenance:* poor designs could result in excessively complex maintenance requirements.

— *Operability:* as above, poor designs in terms of controls complexity can result in inefficient and misunderstood system operation.

— *Window design:* a mixed mode approach might allow this to be less complicated and more robust than in buildings designed for natural ventilation alone.

— *Energy efficiency:* in relation to fully air conditioned buildings, mixed mode systems should use less energy for fans, pumps and cooling.

— *Occupant satisfaction and comfort:* mixed-mode buildings offer the potential for a high level of occupant satisfaction in that they provide more options for correcting a situation.

The selection process is illustrated in Figure 3.8.

The mixed mode approach should not be seen as a compromise solution. It needs to be chosen at a strategic level and the appropriate option selected. The ability to provide general advice on applicability is limited because the final design can range from almost fully naturally ventilated with a degree of fan assistance for still days, to almost fully air conditioned with the option to revert to natural ventilation at a later date.

CIBSE Guide A: *Environmental design* (2015a) provides a calculation and the British Council for Offices' *Guide to Specification* (BCO, 2014) provides fixed values for acceptable temperatures for mixed mode systems.

The BCO *Guide to Specification* recommends the following temperature control regime:

— less than 25 °C for 95 per cent of occupied hours

— less than 28 °C for 99 per cent of occupied hours.

Mixed mode can take a variety of forms and it is essential to be clear about the chosen strategy, i.e.:

— contingency

— complementary (either operated concurrently or in a changeover manner)

— zoned.

These strategies are outlined below.

3.2.2.4.2 Strategy

(*a*) Contingency designs

Contingency designs are usually naturally ventilated buildings that have been designed to permit the selective addition of mechanical air conditioning where these may be needed at a later date. Occasionally the passive measures may themselves be the contingency plan, with an initially fully air conditioned building designed to be amenable to subsequent naturally ventilated operation, either in part or in whole.

(*b*) Complementary designs

Natural and mechanical systems are both present and are designed for integrated operation. This is the most common variety of mixed mode system. Complementary designs can operate in two modes.

— *Concurrent operation:* the most widely used mode, in which background mechanical supply systems, with or without cooling, operate in parallel with natural systems. Often the mechanical system suffices, controlling draughts and air quality and removing heat, but occupants can open the windows if they so choose.

— *Changeover operation:* natural and mechanical systems are available and used as alternatives according to need, but they do not necessarily operate at the same time. Changeover may be on the

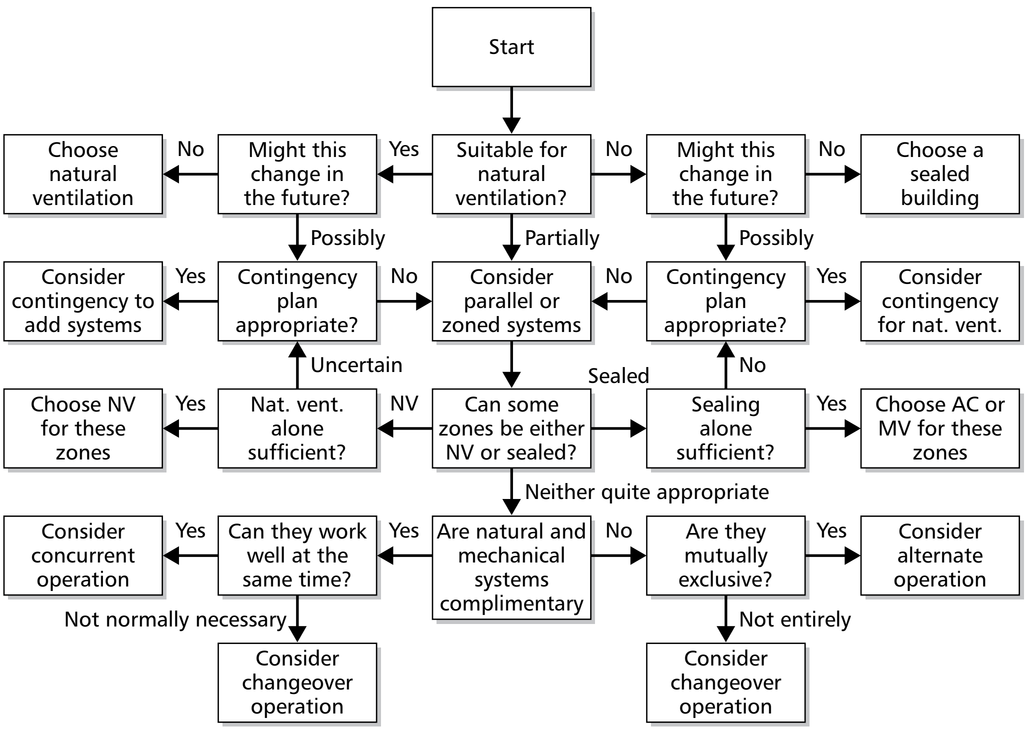


Figure 3.8 Mixed mode selection chart

basis of a variety of conditions as suggested in section 3.2.2.4.3(e) 'Control'.

The chosen control strategy must guard against the risk that changeover systems may default to concurrent operation. Problems of this kind tend to increase with the complexity of the proposed operating strategies.

(*c*) Zoned designs

Zoned designs allow for differing servicing strategies to be implemented in different parts of the building. Many buildings operate in this manner, e.g. a naturally ventilated office with an air conditioned computer room and a mechanically ventilated restaurant and kitchen. Mixed mode increases the range of options available, e.g. offices with openable windows at the perimeter and mechanical supply in core areas. The zoned approach works best where the areas are functionally different, or where the systems are seamlessly blended.

3.2.2.4.3 General issues

This section covers:

- building fabric
- combining natural and mechanical systems effectively
- flexibility
- choice of HVAC system
- energy efficient operation of mixed mode systems
- control
- performance assessment.

General issues that should be taken into account during the selection and development of a mixed mode strategy are outlined. The range of circumstances encompassed by the term 'mixed mode' system is extremely broad. It encompasses, for example, a building that is almost entirely naturally ventilated except for areas of high heat or moisture production served by mechanical systems, to one that is entirely served by air conditioning with the intention that this might in the future be converted to natural ventilation. Hence the guidance provided here must be considered in the light of the specific strategy, or its derivative, as determined in section 3.2.2.5.2 'General system considerations'. Furthermore, this section cannot be treated in isolation but should be read in conjunction with other sections in 3.2.2, which consider the principles of the individual operating modes.

(*a*) Building fabric

Mixed mode is a term describing servicing systems that combine natural ventilation with any combination of mechanical ventilation, cooling or humidification in a strategic manner. In common with buildings that are solely naturally ventilated, this approach requires that suitable benefit be obtained from the building fabric.

The presence of mechanical systems means that a suitable balance needs to be drawn, using value engineering principles, between investment in the relatively long-lived fabric and expenditure on the shorter-lived (and easier to modify/replace) building services, components of which can subsequently be added when and where necessary.

Although the building services in a mixed mode system should usually cost less than in a fully mechanically serviced building, some additional investment may be needed to improve their efficiency, responsiveness, control and adaptability. The initial cost of the mechanical services and the openable windows combined can be greater than that for a sealed building.

Obviously, the greatest economies are made if the improvements to the fabric allow the building services system to be completely eliminated from part or all of the building. For example, reducing fabric and internal heat gains may allow mechanical cooling to be avoided. A highly insulated and airtight fabric with low-powered mechanical ventilation (and heat recovery) may allow both mechanical cooling and perimeter heating to be avoided. The effective use of external night-time temperature differentials can permit any excess heat built up during the day to be removed at night, using natural and/or mechanical ventilation, thereby reducing or eliminating the need for mechanical cooling during the daytime.

In the particular case of zoned systems, a consideration may be to introduce 'localised' fabric enhancements to reduce heat gain, e.g. additional treatment of the roof fabric to ameliorate solar heat gains or additional solar shading of selected windows. A further option might be to introduce 'assisted passive' measures before employing full mechanical systems. This might take the form of a fan in selected natural ventilation 'stacks' for use under peak conditions or on days when inadequate external forces are available, or possibly simple desk fans.

(*b*) Combining natural and mechanical systems effectively

Within complementary systems, the balance between the operation of the natural and mechanical system elements needs to be optimised. This requires a 'trade-off' between the extent of passive and active features, e.g. the number and location of the openable windows will depend upon the extent of mechanical ventilation. The processes by which this balance can be achieved are given in CIBSE AM13: *Mixed mode ventilation* (2000b).

In the case of zoned systems, it requires an understanding of the problem areas that will require mechanical assistance. These might include:

- zones adjoining inferior environmental conditions, such as top floors, corner rooms, internal areas, areas local to non-openable façades or areas where partitioning inhibits bulk air movements
- toilet areas
- areas where heat- or odour-producing equipment is located such as areas containing photocopiers or drinks machines, tea rooms or cleaners' cupboards
- restaurants or kitchens
- areas with dense occupation or high equipment heat loads that may require comfort cooling or close control air conditioning such as meeting rooms, electronic data processing rooms, dealer rooms etc.
- atria.

(*c*) Flexibility

The design of contingency systems must give consideration to flexibility for subsequent adaptation through addition or omission of either centralised or localised supplementary mechanical systems to ensure that the design indoor environment can be achieved.

The extent to which systems are initially installed, or allowance made for them, will depend upon the context but the decision must be taken in the light of the ease and speed of subsequent installation and the likely extent of upgrades, sub-tenancies or critical areas.

Plant rooms

Consideration should be given to allocation of supplementary plant space. Plant room locations should consider ease of installation and access for maintenance. Options include fixed size prefabricated plant rooms or pre-design plant skids.

In some cases, plant rooms may be obtained on hire and 'plugged-in' with minimum site disruption.

Distribution routes

The availability of space for routing services to and around individual rooms often determines the overall level of flexibility. The recommended heights of exposed ceiling soffit slabs to facilitate natural ventilation can often provide adequate space for a future suspended ceiling void or bulkhead, capable of accommodating a wide range of HVAC systems. A raised floor may also allow direct expansion, chilled water and condensate pipes to be routed to any potential 'hotspot'. With appropriate initial sizing, the floor void also has the potential to become a floor supply plenum, from which rooms or larger areas can be supplied with air.

It is important to ensure continuity of the routes between the various parts of the system. A clear route without constrictions is needed from the spaces designated for main plant, via the risers, to the tertiary run-outs. Care should be taken to avoid inadequate space for connection between risers and the floors they are to serve.

Water-based systems

Water-based distribution systems might need to include strategically located provisions for future connections, complete with isolating valves or proprietary, self-sealing couplings. Where appropriate, these basic systems need to be tested at initial completion to confirm their integrity.

(*d*) Choice of HVAC system

The choice of HVAC system will depend upon the clients' functional requirements, see section 3.2.2.5. In the case of zoned or contingency systems, the choice between free-standing or centralised systems is dependent upon:

— the size and distribution of the zones to be treated

— planning restrictions on the use of the façade

— the availability of space for logical horizontal and vertical distribution routes.

(*e*) Energy efficient operation of mixed mode systems

The principles for achieving energy efficient operation in mixed mode systems are a combination of those applied to buildings operating in either natural or mechanical ventilation modes.

Prioritisation of these principles depends upon the extent to which mechanical systems for ventilation, cooling or humidification have been installed.

Additionally, consideration needs to be given to the following issues.

— Mechanical systems should be used only when and where required. The fan power increases with air change rate. Furthermore, as the air change rate increases, the occupants are more likely to notice the difference between when the system is operating and when it is not. This may reinforce the tendency for it to be left running unnecessarily. The use of zoned mixed mode systems helps to overcome the need for whole systems having to operate in order to service small demands.

— Natural and mechanical systems should not conflict in their operation, e.g. mechanical systems competing with air coming in through the windows, or simultaneous humidification and dehumidification. Such situations can be reduced through making users aware of the rationale behind the operation of the system and having suitable trigger points for changeover operation. The state and performance of the system should be monitored and system conflicts reported.

— Systems should not default to a non-optimal state, e.g. switched on when they could be switched off or, at least, operating at reduced output. This risk can be minimised by avoiding over-complex design.

(*e*) Control

The control strategy for mixed mode systems is context dependent, but aims overall for energy efficient operation, maximum staff satisfaction and ease of building management.

This is achieved through:

— maximisation of the natural operating mode

— integration of natural and mechanical systems to avoid system conflicts, wasteful operation and discomfort

— simple and effective controls for the building management that are easy to commission and operate on occupation of the building

— provision of mechanisms to allow occupants to influence their local environment either directly or by feedback to the building operator.

3.2.2.4.4 Performance assessment

Some aspects of mixed mode design may be difficult to resolve or to optimise using normal calculation methods and rules of thumb. More detailed simulation may be desirable:

— in appraising options

— in developing new concepts and testing their robustness under all foreseeable conditions

— in demonstrating the capabilities of an option to clients

— in refining a chosen approach.

Appendix 3.A1 considers the techniques of dynamic thermal simulation and air movement analysis. In applying them specifically to mixed mode systems the designer must consider the following:

— the full variety of potential (often overlapping) operational modes and control variables

— the trigger points for each control strategy element

— the potential actions of occupants

— uncertainty concerning the actual operation of the building compared to the intent and the consequent robustness of the solution

— possible differences between parts of the building and areas of particularly demanding localised conditions, which place particular demands on the ventilation system

— possible adverse interactions between adjacent zones in different operating modes

— possible adverse effects of facilities designed for one mode and operating in another, e.g. facilities designed for summertime ventilation and cooling may not work well in cold weather, possibly leading to draughts or excessive heat losses.

The selection of appropriate weather data and treatment of heavyweight buildings within thermal models is discussed in Appendix 3.A1.

3.2.2.5 Strategic cooling considerations

3.2.2.5.1 Introduction

There is a wide range of comfort cooling and air conditioning plant available. Guidance on the key potential advantages and disadvantages of specific systems is provided in sections 3.2.3.

General guidance on the relative merits of the more innovative systems is available in *Low Energy Cooling — Technology Selection and Early Design Guidance* (Barnard and Jauntzens, 2001). CIBSE Guide F: *Energy efficiency in buildings* (2012b) suggests the classification system for HVAC systems given in Figure 3.9 and discusses issues relating to the energy efficient design of system families.

The main system families are discussed in detail in sections as follows:

— section 3.2.3.2: centralised all-air systems (page 3-21)

— section 3.2.3.3: partially centralised air/water systems (page 3-30)

— section 3.2.3.4: local systems (page 3-39)

— section 3.2.3.5: ground source systems (page 3-44)

— section 3.2.3.6: legacy systems (page 3-49).

3.2.2.5.2 General system considerations

This section outlines general issues that should be taken into account in the selection and development of a comfort cooling or air conditioning strategy. It should be read in conjunction with section 3.2.2.3 on mechanical ventilation and section 3.2.2.4 on mixed mode systems. Design guidance for individual HVAC systems is given in section 3.2.3.

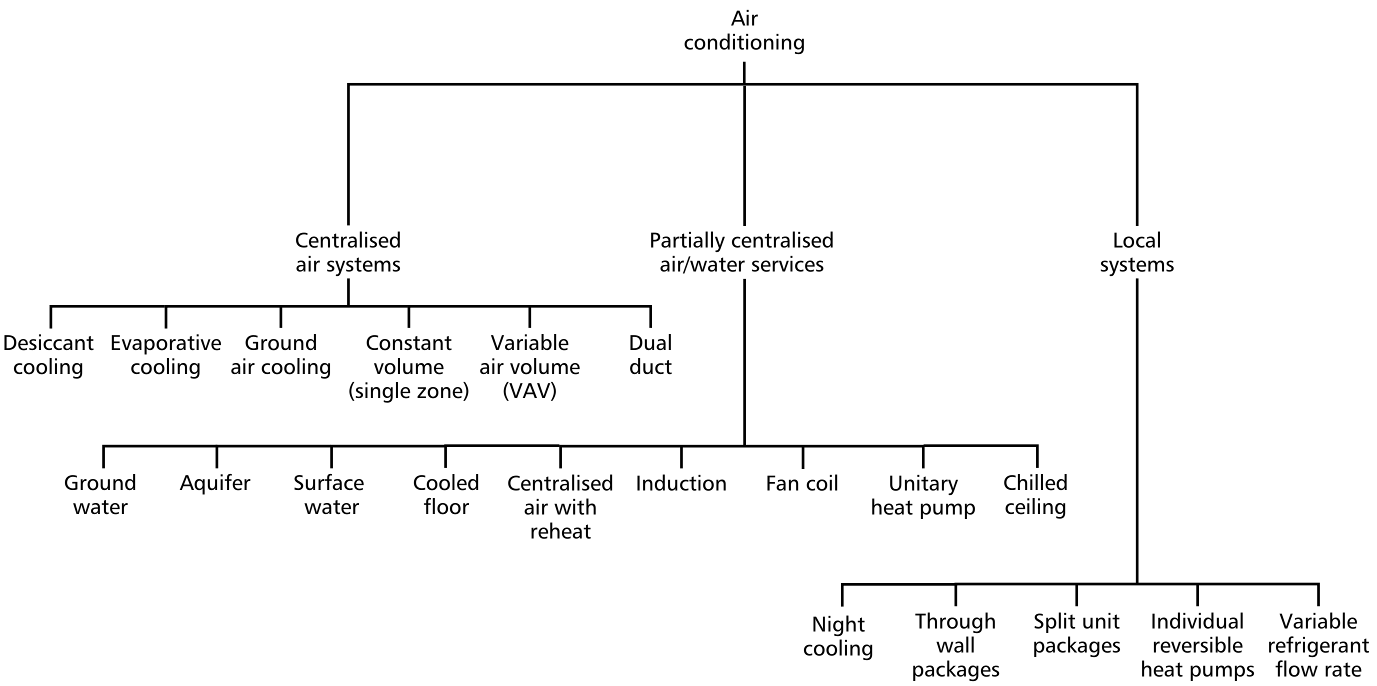


Figure 3.9 Classification of air conditioning systems

(*a*) Energy efficient operation

For the cooling and humidification processes:

— Ensure plant is not oversized (see BSRIA GN11/97: *Oversized air handling plant* (Brittain, 1997)).

— Consider switching off humidifiers when humidity control is not critical. Allow the humidity to drift between 40 and 65 per cent if possible.

— Electric steam humidification can have severe implications for electricity consumption, CO_2 emissions and electricity costs. The peak use of humidifiers tends to coincide with the coolest weather when electricity is also at its most expensive. Alternative humidifier solutions should be considered.

— Avoid simultaneous heating and cooling unless providing close control of humidification.

— Check control settings to ensure that set-points are suitably high in summer and low in winter.

— Ensure that cooling is shut down in winter when it is not required.

— Turn off reheat systems in all areas during the summer unless close control of humidity is being provided.

— Ensure maximum use is made of recirculated air and fresh air for free cooling as appropriate, see 'Free cooling' in the following section.

(*b*) System control

Figure 3.10 summarises the various control options for comfort cooling or air conditioning systems in single-zone applications. Control options for full fresh-air systems are similar to those for recirculation systems but must include provision for frost protection upstream of the filters. The following notes discuss some aspects of control peculiar to air conditioning systems.

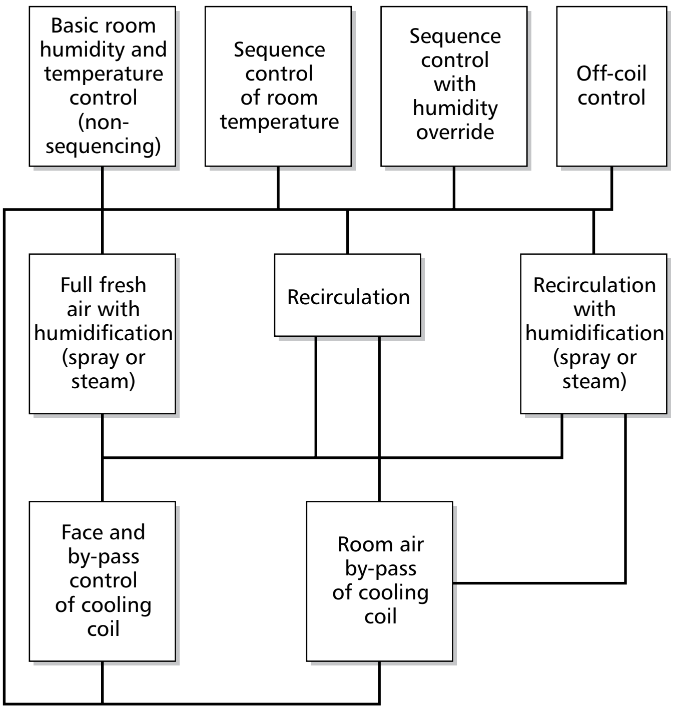


Figure 3.10 Comfort cooling or air conditioning control

'Free cooling'

Before considering a system that depends upon mechanical cooling, every opportunity should be taken to use 'free cooling', of which fresh air is the most obvious source. Cooling systems with low-level input and high-level extract (see section 3.2.2.2) may use higher supply temperatures for summer cooling and can occasionally take away the need for mechanical cooling by a combination of the following:

— drawing outside air from a north-facing aspect

— drawing outside air from a point clear of the 'heat island' at ground level

— drawing outside air through a buried, earth-cooled duct

— supplying the cooling coil with indirectly or evaporatively cooled water from a suitable source.

In the latter case, the potential hazards of microbiological contamination must be considered; refer to CIBSE TM13: *Minimising the risk of Legionnaire's disease* (2013a).

If mechanical cooling is not provided, humidity control will be difficult to achieve since little dehumidification is available from the above, largely passive, sources of cooling. However, with low-level air supply into a space, moisture from the occupants will not mix thoroughly but will be carried to a high level with the upward momentum of the air.

If heat gains are moderate, it may be possible to use all air systems without cooling to limit the rise in internal summertime temperatures, in which case larger air change rates would be required than for air conditioning.

In most recirculation applications it will be worthwhile incorporating motorised dampers, sequenced with the coils, so that outside air, when available at an appropriate condition, may be used to achieve the desired room conditions with minimal load on the central plant. It may also be worth incorporating a means of holding the mixing dampers on full fresh air, cycling to minimum fresh air when outside air enthalpy (h_o) is greater than room enthalpy (h_r) (CIBSE, 2009), see Appendix 3.A2.

Free cooling is also available via cooling towers providing cooling water without the need to operate the chillers. See also BSRIA BG8/2004: *Free cooling systems* (de Saulles, 2004).

Frost protection

Frost protection is required upstream of the filters in both full fresh-air and recirculation systems. Systems may suffer during damper sequencing from a room sensor with inherent time lags under high gain conditions in winter. Stratification through the mixing box may also be a problem (see section 3.2.4.3.2 'Mixing boxes'). In these cases, electric or low-temperature hot water (LTHW) coils should be provided, switched at 4–5 °C from a downstream thermostat.

Face and bypass control

Simultaneous heating and cooling can be avoided by bypassing the cooling coil with either outside, mixed or room air. This relies on accurate damper positioning for control over room conditions and may produce elevated room humidity.

Other control components, not indicated on the system schematics in section 3.2.3, may be required to deal with early morning boost, heat recovery and variable occupancies. It should be borne in mind that the more complex the control scheme, the greater the capital cost and the greater the chances of control malfunction. In particular, humidity sensing is prone to inaccuracy and drift.

Humidity control

An air conditioning system need not provide continuous humidification of the supply air since there will be many occasions when this facility is unnecessary in meeting the comfort needs of the occupants; see chapter 1 of CIBSE Guide A: *Environmental design* (2015a) and chapter 6 of CIBSE Guide F: *Energy efficiency in buildings* (2012b). Guide A suggests that at design temperatures normally appropriate to sedentary occupancy, the room humidity should, if possible, be above 40 per cent. Lower humidity is often acceptable for short periods. Humidity of 35 per cent or below may be acceptable but precautions must be taken to limit the generation of dust and airborne irritants. An upper limit for humidity of 60 per cent is proposed to minimise the risk of mould growth or condensation in areas where moisture is being generated. This can be extended to 70 per cent in terms of maintaining comfortable conditions.

For comfort air conditioning, it is usually satisfactory to supply air with sufficiently low moisture content to cater for maximum latent gain and limit the room percentage saturation by overriding either the humidity sensor or the temperature sensor in the air leaving the cooling coil, as appropriate. Close-control air conditioning is difficult to achieve with multiple zone systems, since each zone requires a dehumidifying cooling coil, re-heater and humidifier to give total control of humidity.

Humidity sensors can be used to limit humidity rise by:

— controlling the output of a cooling coil by proportional control (with integral action if required)

— overriding the action of a temperature sensor controlling some combination of heating coil, cooling coil, mixing dampers and/or extract air heat recovery device

— overriding control over the re-heater so that the sequencing room temperature sensor calls for further cooling and hence dryer air is supplied

— overriding control over the re-heater in a variable air volume zonal reheat terminal so that the zonal temperature sensor calls for a larger volume of dry air to be supplied to that zone

— resetting the off-coil humidity sensor.

In the last case, the supply air moisture content is controlled by the dry bulb temperature sensor. This gives accurate humidity control providing that the cooling coil is efficient and the variation in room humidity is predictable. A humidity sensor can be incorporated to override the cooling coil operation if the occupancy increases above the usual level. Also, simultaneous cooling/dehumidification and reheating will occur for much of the cooling season. With a system serving externally influenced spaces, the off-coil sensor set-point may be reset when the moisture content of the incoming air falls below that required to deal with latent gains. Dew-point control, as applicable to chilled beam installations, is dealt with in CIBSE Guide H: *Building control systems* (2009).

If control of high humidity is not required, the limits of the proportional band of a sequence controller can be the winter and summer design room conditions. Otherwise, different conditions for summer and winter can only be achieved by using integral action to remove any offset and by resetting the set-point of the room temperature sensor in response to an outside temperature sensor. Sequential control will normally require a wide proportional band, particularly if mixing dampers are included.

Humidity sensors can be used to control low humidity by:

— providing step or on/off control of a steam humidifier (see Figure 3.12 below)

— providing proportional control of a pre-heater and/or mixing dampers to provide appropriate on conditions to a water spray-coil humidifier with the spray pump running continuously (see Figure 3.15 below)

— switching on a spray washer pump or spinning disc humidifier and providing appropriate on conditions by proportional control over the pre-heater and/or mixing dampers.

If off-coil sensors are not employed, a low-limit sensor may be required to bring in the heating coil if the supply air temperature falls below the minimum design value. This is necessary where room or return air temperature sensors are likely to be slow to respond to low supply conditions.

(*c*) Fan position

The air handling unit (AHU) fan is normally positioned downstream from the heating and cooling coils to maximise coil efficiency, minimise risk of water droplet carryover and mix the air leaving the heat exchangers that is otherwise stratified (see Figure 3.11(a)).

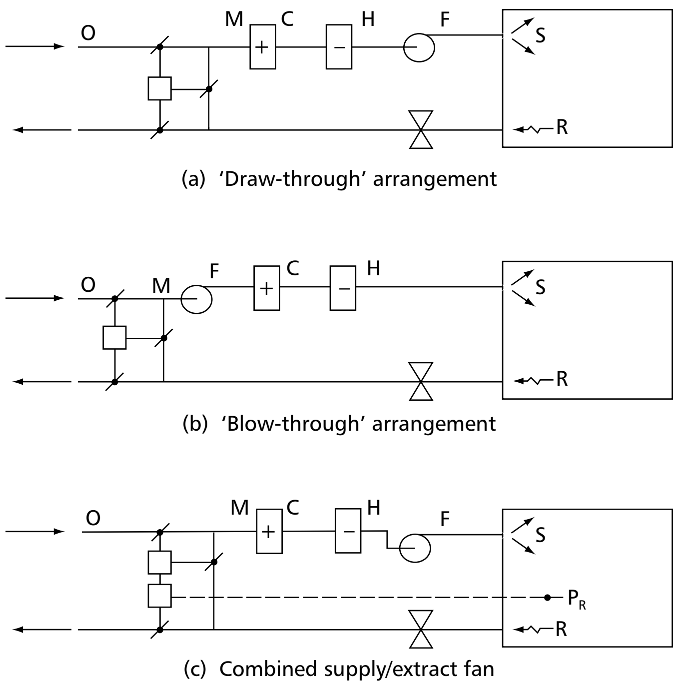


Figure 3.11 Supply air handling plant; alternative arrangements for position of fans

Alternative arrangements are possible and should be considered in the context of the overall design.

'Blow-through' central plant

The main advantages of positioning the fan upstream of the cooling coil are that:

— a lower supply air moisture content can be achieved at a particular apparatus dew-point and chilled water temperature (see Figure 3.11(b))

— the risk of drawing airborne contaminants from the drainage system or plant room into the system is reduced.

The main disadvantages are that:

— since the cooling coil is under positive pressure there is a greater risk of condensate leakage through the casing

— an additional plenum or transition piece is required at the fan discharge to reduce the air velocity to an appropriate value at the coil face and ensure an even distribution of air over the face area.

Soiling of the fan may be reduced by locating the filter upstream of the fan, see Figure 3.11(a).

Combined supply/extract fan

A single fan can both draw air through the extract system and blow air through the supply distribution system, providing that a balance can be achieved between extract and intake pressure losses using an appropriate combination of fixed resistance and damper in the intake.

In most cases, free cooling from full fresh air will be required. Therefore, means must be provided for varying the proportion of return air to outside air at the mixing box by damper modulation. Some means of pressure relief will be required in the building or system and Figure 3.11(c) shows a relief damper controlled from a room pressure sensor (P_R). Pressure relief will only be effective with very low duct resistance between the room return point and exhaust damper. For extract systems with low resistance, this damper could be replaced with simple, weighted pressure relief flaps (also see section 3.2.4.3.2).

Zoning

The loads on an air conditioning plant are rarely constant due to changes in solar gain, occupancy or the use of lights etc. If the loads throughout the building vary together (i.e. in phase), or the variations are not large enough to drift outside of the acceptable limits, single-zone control can be adopted. However, if different areas experience load changes that are out of phase, the supply air must be provided at a rate or condition appropriate to each zone.

Most deep-plan buildings require division into perimeter and internal zones. The depth of perimeter zones mainly depends on the penetration of sunlight and daylight which is determined by orientation, external shading, shape and size of windows, characteristics of the glass and the type and pattern of use of blinds.

For a typical multiple-zone application, the following points should be noted:

— For a constant volume flow rate to be maintained to each zone the system must be capable of supplying air at various temperatures at any one time. This may involve simultaneous heating and cooling of the supply air.

— All rooms with similar solar gain patterns can be zoned together provided that heat gains/losses are similar. However, the number and position of the zonal sensors will be important. Corner rooms pose further problems.

— North-facing rooms experience less variation and can be grouped with internal zones for cooling provided that heating is dealt with by other means.

The success of an air conditioning system depends largely on wise zoning and careful positioning of sensors in relation to the sources of heat gains.

3.2.2.5.3 Performance

Appendix 3.A1 considers the techniques of dynamic thermal simulation and air movement analysis. For England and Wales, Building Regulations Approved Document L2 (NBS, 2013a) includes specific performance requirements for air conditioning systems (see section 3.2.2.4.4 'Performance assessment'). A number of other guidance documents or techniques exist that can be used to provide target energy benchmarks, or for comparative purposes, at early stages of the design process. These include:

— CIBSE Guide F: *Energy efficiency in buildings* (2012b)

— ASHRAE BIN method (ASHRAE, 2013b)

— Energy Efficiency Best Practice Programme (EEBPP) Energy Consumption Guides*.

3.2.3 Systems

3.2.3.1 Introduction

This section summarises the various system types.

3.2.3.2 Centralised all-air systems

These consist of:

— constant volume (single or multi- zone)

— variable air volume systems.

Central plant and duct distribution are employed to treat and move all of the air supplied to the conditioned space. In constant volume systems, the heating and cooling loads of the building are met by changing the temperature of the supply air. In a variable air volume system, it is the airflow that is altered to meet the requirements of the space.

3.2.3.2.1 Single duct constant air volume systems

(*a*) Description

While maintaining a constant air volume, single-duct constant volume systems vary the supply air temperature in response to space conditions. The simplest system is a supply unit serving a single-temperature control zone; a

* http://www.cibse.org/knowledge/archive-repository/archive-repository

single-zone system. Applications include large rooms such as lecture theatres. They should not be used for multiple zones with different heating/cooling loads because control of conditions will be very poor and they will be very inefficient in operation.

Single-zone systems with room control maintain temperature closely and efficiently. The same systems with off-coil control are also used where air is to be supplied to a number of zones at the same conditions. Examples of this are displacement systems (see section 3.2.2.2.5 'Displacement ventilation') and systems that provide fresh air in conjunction with space conditioning systems, such as fan coil units.

The multi-zone reheat system is a development of the single-zone system. Conditioned air is supplied by the central unit, generally at a fixed cold temperature. This air is then reheated as required by heaters in the supply ductwork to each zone. This provides space temperature control for zones of unequal loading. However, energy wastage can occur when air cooled by the central unit is subsequently reheated. Where the total air supply is greater than the outside air requirement, recirculation is normally used to minimise energy requirements. For full fresh-air systems, heat recovery devices should be considered (see section 3.2.4.3).

(*b*) Design

Single-zone room control

The typical arrangement for a simple single-zone system is shown in Figure 3.12. The temperature sensor T_R controls the cooling coil and re-heater in sequence within its proportional band (see Figure 3.13).

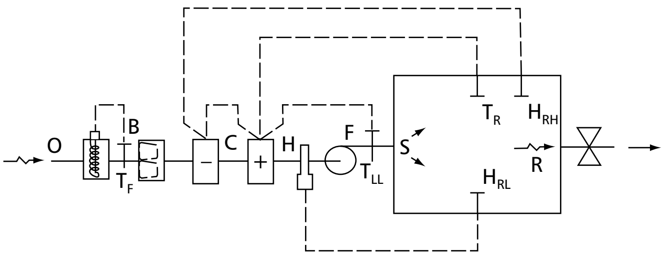


Figure 3.12 Full fresh-air system with steam humidification: sequence control with humidity override

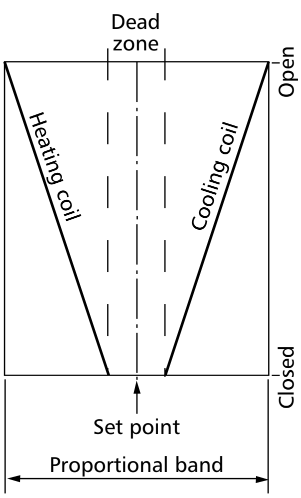


Figure 3.13 Sequential control of heating and cooling units

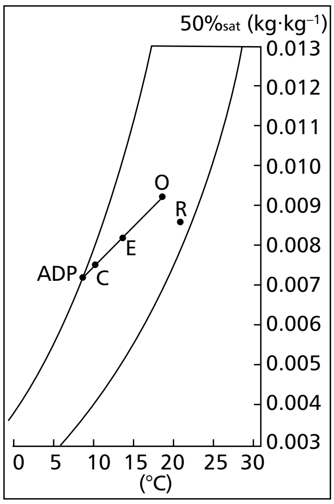


Figure 3.14 Full fresh-air with steam humidification: sequence control with face-and-bypass dampers; psychrometric process

The humidity sensor H_{RH} will bring in the cooling coil out of sequence and T_R will call for simultaneous reheat to deal with overcooling.

Energy wastage by reheating after dehumidification can be reduced by using face-and-bypass dampers. In the scheme shown in Figure 3.15 and Figure 3.16, T_R positions the dampers in sequence with the heating coil to provide an appropriate supply condition rather than controlling cooling directly via the cooling coil. When combined with appropriate cooling media temperatures (see Figure 3.14), this method provides adequate humidity control without wasteful reheat. Room humidity will rise, particularly at low sensible heat loads. However protection against high humidity can be provided by using a humidity sensor (H_{RH}) to override damper control, the re-heater being brought in to deal with resultant over-cooling.

Figure 3.16 shows a typical arrangement with recirculation. The temperature sensor T_R controls the cooling coil, mixing dampers and re-heater in sequence within its proportional band (see Figure 3.17).

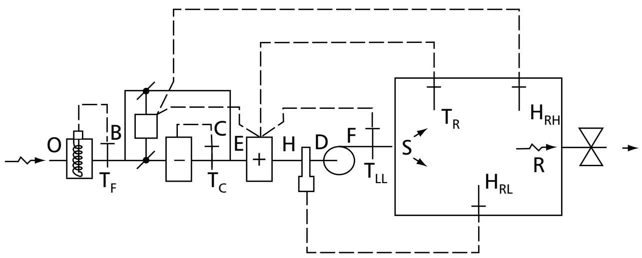


Figure 3.15 Full fresh-air system with steam humidification; sequence control with full face-and-bypass dampers

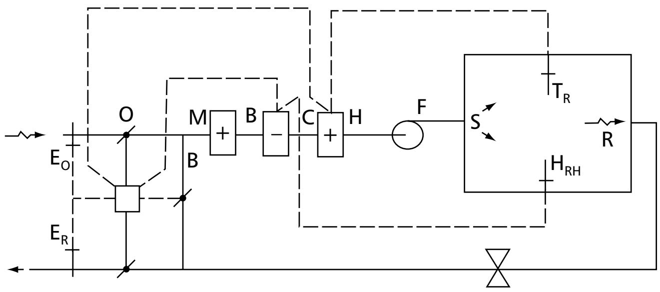
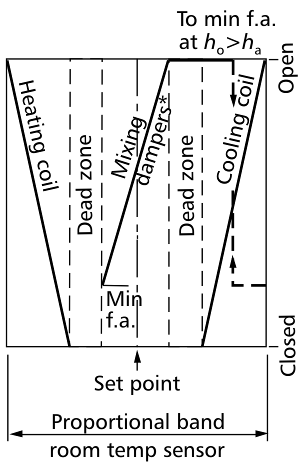


Figure 3.16 Recirculation: sequence control with humidity override



* For a full fresh air system with heat recovery, the mixing dampers are replaced by the heat recovery device in the above sequence.

Figure 3.17 Sequential control for recirculation systems

In the scheme shown in Figure 3.18, instead of directly controlling flow of the cooling medium through the cooling coil, an appropriate supply condition is provided by positioning the bypass and recirculation dampers in sequence with the heating coil in response to an appropriate signal from T_R. This gives closer control of room humidity than face-and-bypass dampers because air extracted from the conditioned room air only is bypassed around the cooling coil.

A part-load analysis of mass flow and temperature balance can be used to determine the on- and off-coil conditions for the cooling coil and hence the resultant room percentage saturation (see Figure 3.19). Control is otherwise similar to face-and-bypass control.

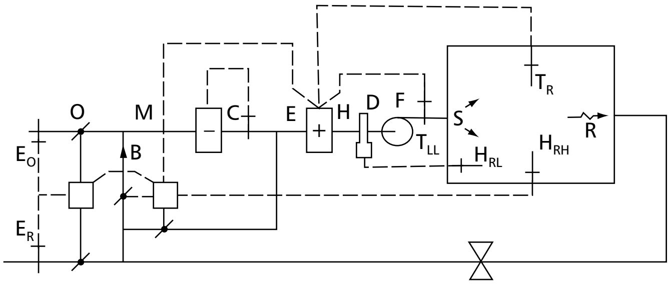
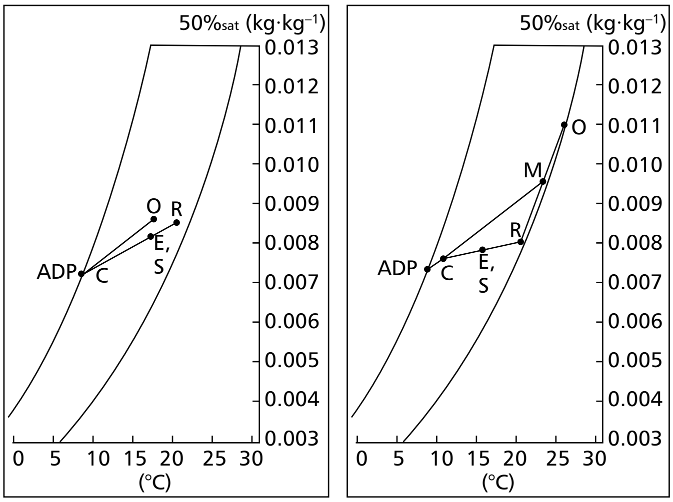


Figure 3.18 Recirculation with steam humidification: sequence control with room air bypass



(a) t_o = 18 °C (b) t_o = 27 °C

Figure 3.19 Recirculation with steam humidification: sequence control with room air bypass; psychrometric chart

Single-zone off-coil control

Figure 3.20 shows an arrangement in which the off-coil dry bulb temperature sensor T_C controls the cooling coil, pre-heater and re-heater in sequence to achieve its set-point, adjusted against the outside temperature sensor T_O, if appropriate. Alternatively, the room temperature sensor T_R can be used to control the output of the re-heater to achieve the desired room temperature.

Figure 3.21 shows the typical arrangement with recirculation. The off-coil dry bulb temperature sensor T_C controls the cooling coil, mixing dampers and heating coil in sequence within its proportional band (see Figure 3.17). The pre-heater (shown dotted) is incorporated into the sequence only if large fresh-air rates promote high mixing ratios, hence low winter temperatures, through the air-handling plant. Alternatively, a low-limit sensor could be used to bring in the heating coil as necessary. If adiabatic humid-

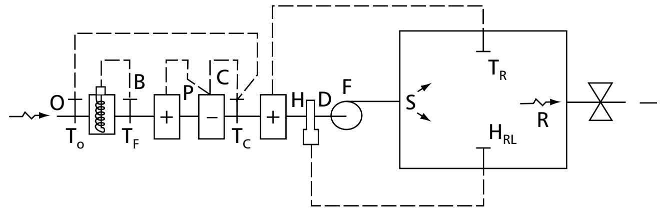


Figure 3.20 Full fresh air with steam humidification; off-coil control

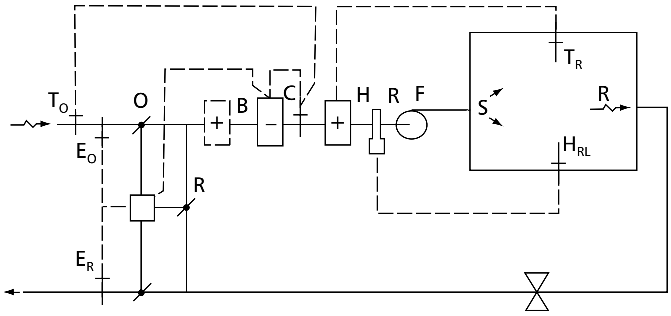


Figure 3.21 Recirculation; off-coil control

ification is employed to deal with the associated low winter moisture contents, a pre-heater may be necessary to heat mixed air. The pre-heater is optional, but if it is not present, a low-limit sensor should be provided to bring in the re-heater to prevent cold draughts on start-up during wide load variations. Off-coil control before the fan can be difficult due to potential air stratification.

Multi-zone reheat system

Figure 3.22 shows a typical arrangement for a terminal reheat system. Air is treated centrally and distributed at a common temperature and moisture content such that:

— the temperature is sufficiently low to deal with the greatest sensible heat gain (or lowest net loss)

— the moisture content is at a level that will satisfy the zone having the lowest sensible heat ratio

— adequate fresh air is provided to the zone having the highest mixing ratio of local fresh air to supply air.

For any zone that experiences over-cooling by the centrally treated air, the room temperature sensor T_R brings in the respective zonal re-heater.

The condition of the distribution air can be varied with outside temperature when the system is serving perimeter zones only. Internal zones are likely to experience high cooling loads even at low external temperatures, hence the air leaving the central plant must be kept at the minimum design condition. Serving perimeter and internal zones from one plant can prove wasteful of energy unless humidity control necessitates low supply temperatures or it is possible to achieve low supply temperatures by utilising sources of free cooling.

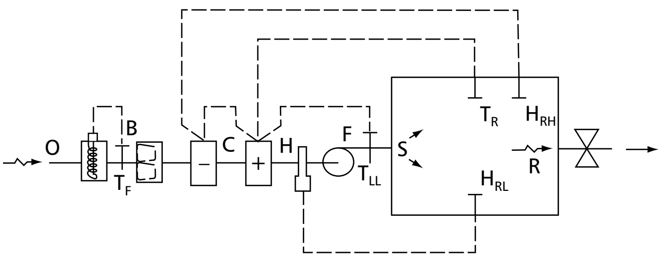


Figure 3.22 Terminal reheat

In order to reduce unnecessary reheat, control signals from the re-heater control actuators can be analysed centrally, the resetting schedule for the off-coil sensor being based on the zone requiring the lowest supply air temperature, i.e. minimum reheat requirement.

Dew-point systems provide saturated air at the cooling coil at all times to provide very stable humidity conditions when air is reheated to the desired space temperature. However, these systems are only necessary for special applications such as laboratories, and should normally be avoided as they can be very inefficient.

Construction

See section 3.2.4 for equipment requirements.

3.2.3.2.2 Single duct variable air volume (VAV) systems

(*a*) Description

VAV systems control the temperature in a space by varying the quantity of air supplied rather than the supply air temperature. Terminal devices at the zones modulate the quantity of supply air to the space. The supply air temperature is held relatively constant, depending on the season. VAV systems can provide reduced fan energy consumption in comparison with constant volume systems. They are primarily suited to applications with a year-round cooling load, such as deep plan offices. Potential problem areas include: humidity control, provision of sufficient outside air and air movement. Where close humidity control is critical, e.g. laboratories or process work, constant volume airflow may be required.

(*b*) Design

The control of VAV systems is considered in detail in CIBSE Guide H: *Building control systems* (2009). Varying the volume of air supplied to a space has the following consequences:

- the ability to offset sensible heat gains is reduced
- the ability to offset latent heat gains is reduced
- the mixing ratio remains constant, its ability to dilute odours, CO_2 etc. is reduced
- unless special air terminal devices are utilised, the ability to create room air movement is reduced.

The volume of supply air is normally varied in relation to room air temperature (sensors T_{RE} and T_{RW} in Figure 3.23) and will respond only to changes in sensible gain. Hence, unless the main load variations are caused by occupancy changes, unacceptable humidity rise and depletion of fresh air can result. The effect on room air movement will depend largely on the turndown efficiency of the terminal device (see section 3.2.2.2.1 'Room air diffusion: criteria for design'). Generally, humidity rise on turndown can be kept within acceptable limits provided that a cooling differential of about 8–12 K is used (see Figure 3.24). Limiting turndown and incorporating reheat may be used in zones with particularly high latent gains, such as conference rooms.

Fresh-air rates on turndown can be maintained at the central plant by means of an inlet velocity sensor to control the position of the mixing dampers.

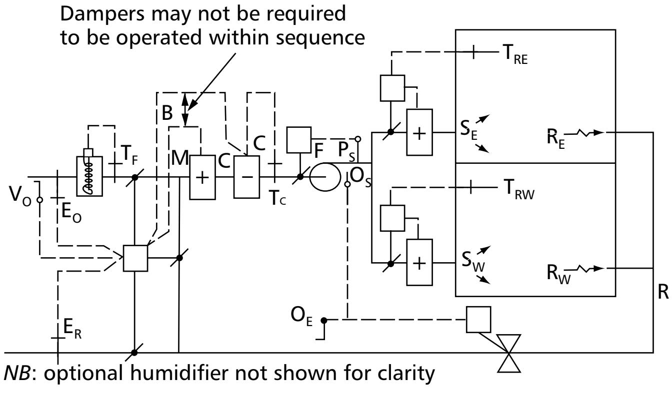


Figure 3.23 VAV with terminal reheat

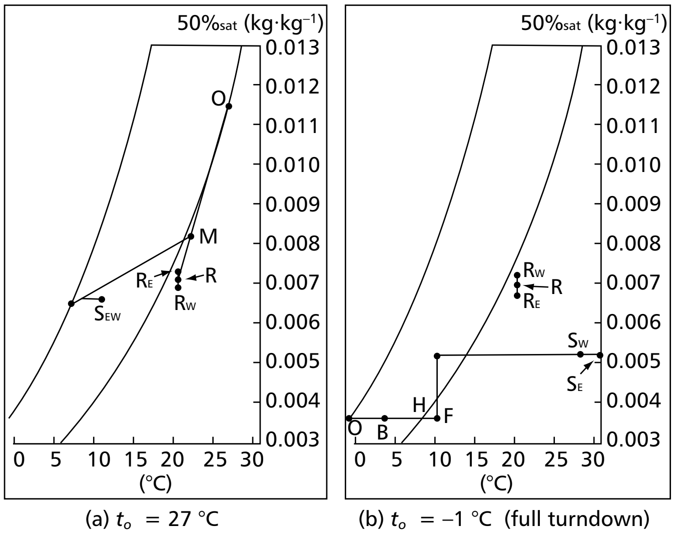


Figure 3.24 VAV with terminal reheat; psychrometric process

The efficiency with turndown depends on:

- the position selected for sensing flow changes
- the mechanism employed for reducing total flow rate
- the mechanism by which flow-dependent signals are converted to movement at the actuator.

If the supply fan duty is to be modulated from a static pressure sensor in the supply ductwork, the sensor must be in a position that gives a reasonable indication of total flow requirements. Medium to high duct velocities are needed to improve sensor sensitivity to flow changes (CIBSE, 2009; Jones, 2000; Hittle et al., 1982).

The extract system must respond to changes in the supply flow rates to avoid over/under-pressurisation of the building. This may be dealt with at two levels:

- *Individual control zones*: if zones are separated by solid partitions any imbalance in supply flow rates between zones must produce corresponding changes in extract flow rates. Thus the extract duct for each zone will contain a damper controlled to follow changes in supply volume. In the case of a multi-storey open-plan building, this may be necessary on a floor-by-floor basis.
- *Choice of fan characteristics*: the supply and extract fans will usually be of different types to cope with dissimilar system pressure requirements. Hence, their characteristics will differ accordingly.

AHU fans normally achieve variable volume by variable speed drive.

For perimeter zones where minimum loads fall below the potential cooling at full turndown, some means of heating will be necessary to avoid over-cooling. (*Note*: this may also be a consideration for internal zones with intermittent loads, e.g. meeting rooms.)

If a step change in load from net cooling to net heating occurs in all zones simultaneously, a changeover coil in the central plant may be used to supply either constant temperature heated or cooled air. Where there is no step change, the system can be controlled to cycle between heating or cooling depending on the requirement of the majority of the zones using the thermal inertia of the building to limit hunting. Alternatively, it may be possible to reset the set-point of the off-coil sensor in the manner of a variable temperature system, typically by scheduling against outside air temperature. This has the advantage of expanding the range of loads that the system can accommodate and eliminating some of the disadvantages of turndown. However, fan running costs increase because of the reduced turndown over the whole year.

(*c*) Terminal reheat

To meet heating requirements, re-heater batteries are provided in the terminal devices. These are normally controlled in sequence with airflow from a room temperature sensor. As the requirement for cooling reduces (and heating increases), airflow is reduced to a minimum and the re-heaters are brought on. Compared with constant volume reheat, this reduces energy consumption, as the amount of air being cooled and then reheated is reduced.

(*d*) Perimeter heating

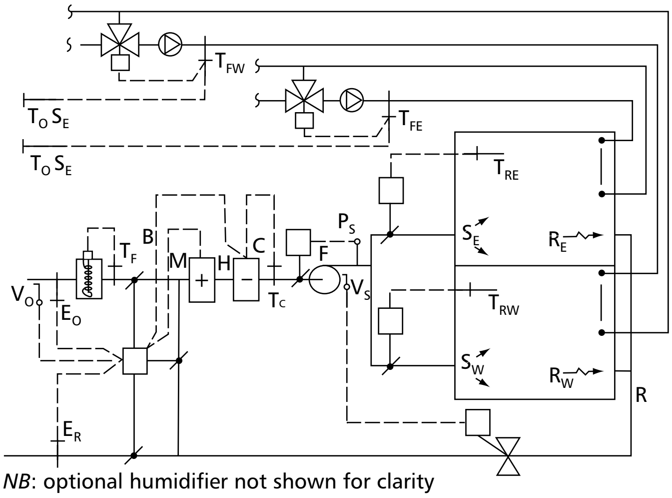
If significant perimeter down-draught is likely (large glazed surfaces), under-window heating may be desirable (see Figure 3.25. The output of the heating system must be controlled in such a way as to prevent the heat appearing as a cooling load. One solution is to control the heating and cooling in sequence from a common temperature sensor. Water temperature should also be scheduled against outside air temperature and compensated for different orientations if appropriate. The resetting schedule is shown in Figure 3.26 and is based on providing sufficient heating to deal with the greatest potential cooling at maximum turndown. An extension of this principle is to utilise a VAV system for internal zones and a variable temperature air conditioning system to deal with perimeter loads.

(*e*) Induction VAV: air terminal

A separate constant-volume, primary air duct or system is used to encourage constant throw from supply air terminals. A separate source of primary air can be used to provide a constant fresh-air supply, scheduled against outside air temperature as appropriate. Primary air is discharged at a constant volume through induction nozzles or slots at the variable volume supply outlet, which may be in the form of a side-wall grille, ceiling diffuser or induction nozzle.

(*f*) Induction VAV: ceiling plenum

Air from the central unit is mixed with air from the ceiling void, which has been heated through exhaust luminaires. Primary air reduces with cooling load whilst total air supply volume is kept relatively constant. This results in good fan



NB: optional humidifier not shown for clarity

Figure 3.25 VAV with perimeter heating

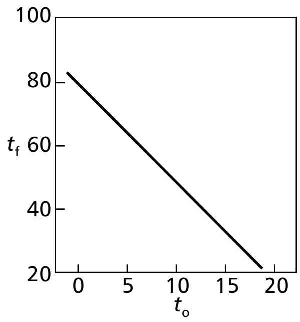


Figure 3.26 VAV with perimeter heating; example of resetting schedule for flow temperature sensors against outside air temperature

economy whilst room air movement is greater than that obtained from conventional throttling devices.

(*g*) Fan-assisted VAV

In principle, this system is similar to the ceiling plenum induction system but uses a fan within each terminal unit to enhance room air movement on turndown and blend warm air from the ceiling void with that from the central unit.

There are two arrangements in common use, whereby the fan and VAV damper are connected either in parallel or in series (see Figure 3.27(e) and (f), below, respectively). The parallel arrangement requires the fan and damper to be controlled in sequence; the fan being brought in only on full turndown. With the series configuration, the fan runs continuously, thus maintaining constant room air movement (and noise generation) with varying proportions of air drawn from the ceiling void. A reheat coil can be incorporated into the device if insufficient heat is available from the luminaires.

Fan-assisted terminals with reheat can be used for early morning pre-heat with the central plant held off.

(*h*) Construction

See section 3.2.4 for equipment requirements.

The VAV device for varying airflow to a control zone may either form part of the air distribution system and serve a number of conventional air terminal devices, or it may be the air terminal device itself. In the latter case it may incorporate some means of maintaining reasonably constant throw. Figure 3.27 below shows examples of these devices.

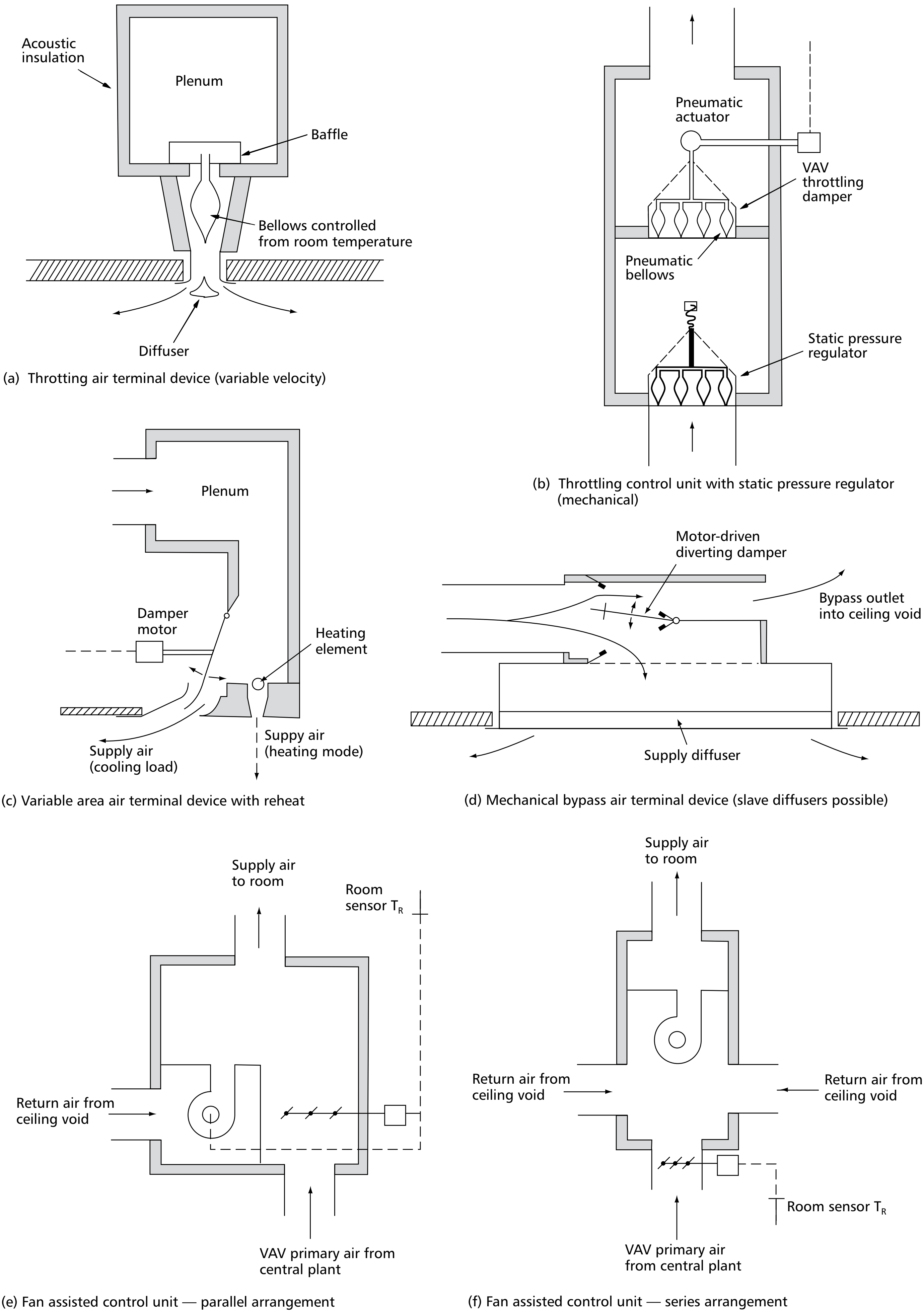
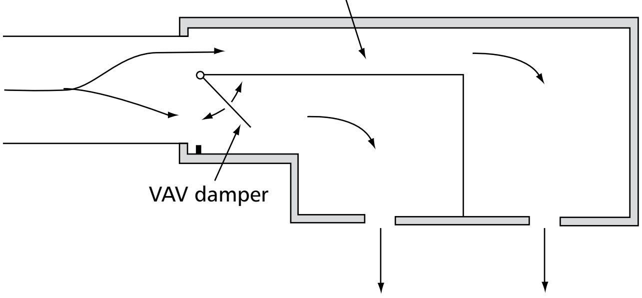
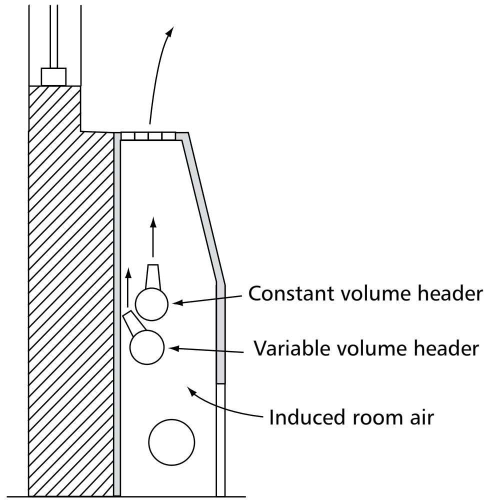


Figure 3.27 VAV devices



(g) Induction air terminal device using slots



(h) Induction air terminal device using nozzles — located under window

Figure 3.27 VAV devices — *continued*

Flow rate may be modulated by:

- throttling by dampers
- throttling by variable area
- mechanical bypass by diverting supply air back to the air handler (constant volume fan).

Control of the device can be achieved by the system being operated, utilising the pressure available in the supply duct, or by the use of an external power source, either electric or pneumatic.

The device may also incorporate some means of system balance under varying flow conditions, normally by automatic damper adjustment from a static pressure sensor. Alternatively, this function may be fulfilled by a separate damper box.

VAV systems often use velocity reset VAV boxes. Primary air volume is set between minimum and maximum settings in relation to space temperature. These are pressure dependent and the system is essentially self-balancing.

VAV devices may be actuated mechanically, by means of a spring-loaded regulator that closes as pressure increases, or pressure-actuated using the changes in branch pressure to position a throttling damper. Both types increase fan pressure requirement by 100 to 200 Pa.

3.2.3.2.3 Evaporative cooling (direct and indirect)

(*a*) Description

In evaporative cooling systems, the evaporation of water is used to decrease the dry bulb temperature of air. There are two main categories of evaporative cooling (see Figure 3.28):

- *Direct evaporation:* water is evaporated directly into the supply airstream, adiabatically reducing the airstream's dry bulb temperature but increasing its absolute humidity. Direct coolers may operate using spray air washers or wetted media.
- *Indirect evaporation:* two airstreams are used. A secondary airstream is cooled directly using evaporation and then exhausted. This secondary stream may be outdoor or exhaust air. The cooled, moist, secondary air cools the primary supply air indirectly through an air-to-air heat exchanger. (When the secondary stream is exhaust air, the heat exchanger can also be used to preheat outdoor air in the winter.) Hence indirect evaporative cooling provides sensible cooling without increasing the latent capacity of the supply air. There is an efficiency penalty when compared with direct evaporation.

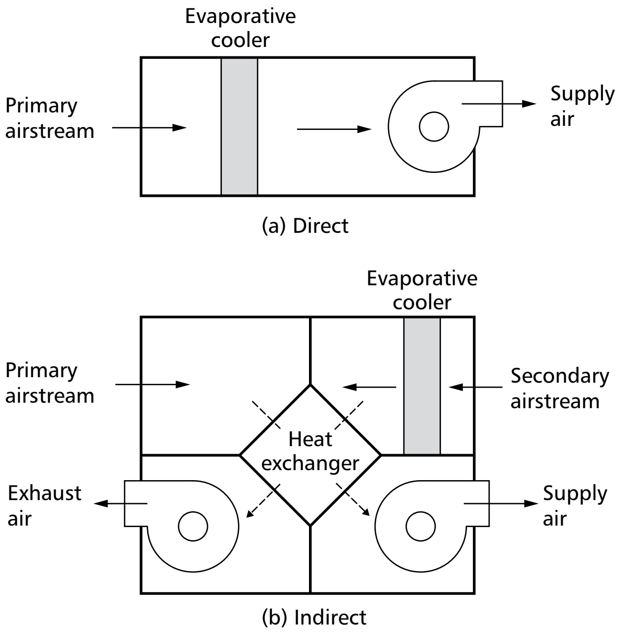


Figure 3.28 Direct and indirect evaporative cooling

When designed as a standalone system, an evaporative cooling system requires three to four times the airflow rate of a conventional air conditioning systems. Because of the higher airflow rates, larger ducts are required. The higher airflow rates and the absence of recirculated air promotes good indoor air quality. In practice, because of the limited cooling capacity of an indirect evaporative cycle, the primary air is often cooled again by direct evaporation or by a mechanical cooling system. This is called a two-stage or indirect–direct system. In practice, within the UK, the technology is used as a supplementary cooling measure only, or in combination with desiccant cooling.

(*b*) Performance details

The lowest temperature that can theoretically be achieved is the dew point temperature of the treated air. The resulting cooling depends on the wet-bulb depression and the cooler effectiveness. Evaporative cooling will perform best under high-temperature, low-humidity conditions. These periods are relatively infrequent in the UK and, as a result, evaporative cooling can meet moderate sensible cooling loads under dry conditions with no latent cooling loads. It cannot address latent cooling loads. With high wet bulb temperatures, evaporative cooling systems will not deliver the required cooling. These systems are ineffectual in humid climates.

The saturation effectiveness depends on the equipment design (e.g. contact time, area and airstream velocity, condition and adjustment). For direct spray types, effectiveness is 50–90 per cent, with the higher values associated with double-spray arrangements. Direct wetted media coolers could have an effectiveness of 85–95 per cent. Typically, indirect pre-cooling stages achieve 60–80 per cent effectiveness. System effectiveness should be considered where more than one stage takes place.

(*c*) System enhancements

Evaporative cooling may be enhanced by:

— combining it with other technologies to provide supplementary cooling, e.g. hollow core systems

— using it in systems that have low cooling loads, e.g. displacement ventilation

— using it to pre-cool condenser air

— using an indirect evaporative pre-cooler to recover heat energy in winter.

(*d*) Control strategies

The control strategy depends on the number of stages (up to three) in place. Control is related to the set-point temperatures of the operating modes of the different system components. The operation of that component can be on/off or modulated within its operating range.

(*e*) Critical design factors

The following factors must be considered.

— Evaporative coolers need to be shaded.

— The effect of design conditions; close control is difficult.

— Higher air velocities are required for a stand-alone system, although the temperature depression is less and its humidity exceeds the room air, hence comfort conditions should not be adversely affected.

(*f*) Maintenance

Extra maintenance is required in comparison with a conventional system in terms of the preventative care needed to drain the system and flush the wetted media to prevent the accumulation of mineral deposits. This is particularly important when the system is turned off after summer.

Guidance on the treatment of water used within direct evaporators would suggest that water treatment should not be undertaken since the water will need to discharge to waste. Also see CIBSE TM13: *Minimising the risk of Legionnaire's disease* (2013a) for guidance on measures to reduce the risk of Legionnaires' disease. No distinct guidance is given for indirect evaporative systems.

(*g*) Applications

Evaporative cooling is most often used in buildings with relatively small cooling loads or buildings that do not require tight humidity and temperature control, such as warehouses. It can be used with retrofit applications provided that ducting requirements can be met.

(*h*) Space allowances

Evaporative coolers are somewhat larger than conventional HVAC units for a smaller cooling capacity. Moreover, space may also be required for larger air ducts, typically 15–30 per cent for a direct system.

(*i*) Maintenance and health

In the UK, designs have used indirect evaporative cooling systems. Corrosion and scaling of the indirect evaporative cooling coil tubes can occur. These should be rust-resistant, copper-bearing, galvanised iron. Scaling in and around the spray area may need to be controlled with chemicals.

3.2.3.2.4 Desiccant cooling systems

(*a*) Description (Warwicker, 1995)

Desiccants are hygroscopic materials that readily absorb or give off moisture to the surrounding air. They can be solids or liquids, although application of desiccant technology in the UK is currently based on the use of solid material. They may be natural or synthetic substances.

The moisture containment of a hygroscopic material in equilibrium depends upon the moisture content of the surrounding air and varies widely for different desiccants. The moisture content also varies for different temperatures at the same relative humidity.

If the desiccant material contains moisture in excess of the surrounding airstream, it will release moisture to the air with the absorption of heat and there will be a cooling effect equal to that of evaporation. If the desiccant material contains moisture below that of the surrounding air, it will absorb moisture from the air. Heat will be released corresponding to the latent heat given off if a corresponding quality of water vapour were condensed.

Desiccants transfer moisture because of a difference between the water vapour pressure at their surface and that of the surrounding air. As the water content of a desiccant rises, so does the water vapour pressure at its surface.

Both higher temperatures and increased moisture content boost the vapour pressure at the surface. When the surface vapour pressure exceeds that of the surrounding air, moisture leaves the desiccant. After the heat dries the desiccant, its vapour pressure remains high so that it has little ability to absorb moisture. Cooling the desiccant reduces its vapour pressure so that it can absorb moisture once again. This is referred to as 'regeneration'.

Desiccant systems can be applied where:

— high latent loads are present that would otherwise require very low chilled water temperatures, e.g. supermarkets

— a source of low-grade energy such as waste heat or solar energy can be used to regenerate the desiccant.

(*b*) Design and operation

Figure 3.29 shows a typical air conditioning plant using solid desiccant technology.

Outside air (A) passes through the filter before entering the desiccant wheel where moisture is removed from the air (B). During this absorption process, the temperature of the air rises and is then cooled by the thermal wheel (C). The air is now drier and cooler and may be further cooled by either evaporative cooling (D) or mechanical cooling (D1), dependent upon the required final condition.

The cooler, dehumidified air is then introduced to the space where it provides all the latent cooling requirement and some sensible cooling, depending on the type of system chosen, either all air or air/water.

The return air leaves the space via a filter (E) before entering an evaporative cooler (F). This cool humid air enters the thermal wheel (which acts as the cooling for the supply air) and is heated by the supply air (F). It is then further increased in temperature by the heater (G), where it regenerates the desiccant wheel. In order to save energy, some of the air bypasses the heater and the desiccant wheel (J). The psychrometric process is shown in Figure 3.30.

(*c*) Performance

Like any other system, performance is dependent on the external and internal conditions. The difference between desiccant systems and those based on refrigerant use is the impact of the ambient moisture content. Increased moisture content reduces the performance of the desiccant system to a greater degree than increased temperature, which can more easily be handled.

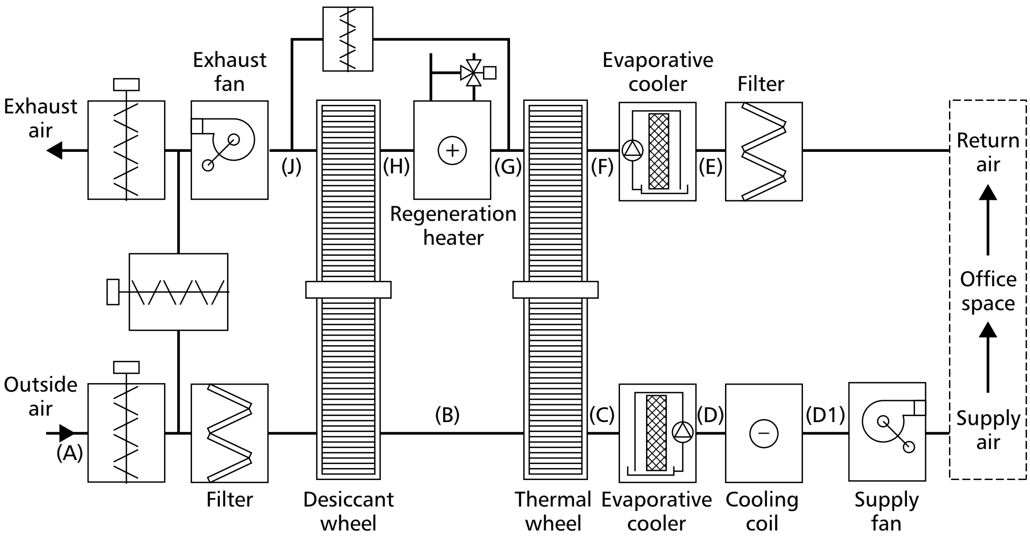


Figure 3.29 Typical air conditioning plant using solid desiccant technology

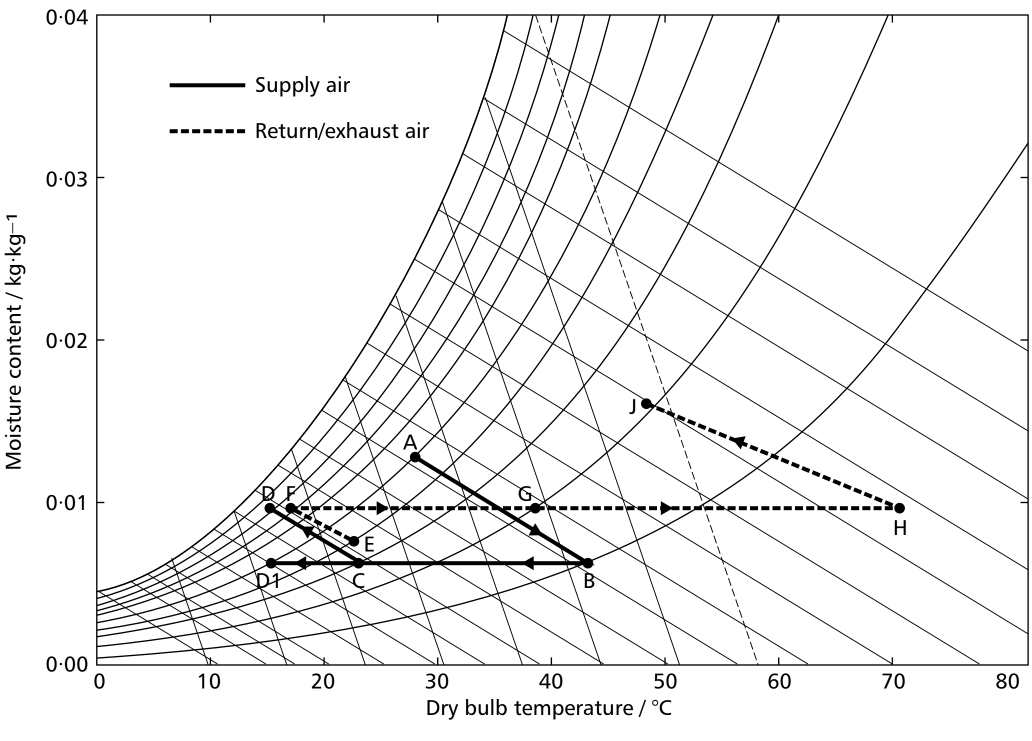


Figure 3.30 Psychrometric process for desiccant cooling

Performance is also dependent on the efficiency of the energy recovery system and humidifiers. By using a desiccant in conjunction with an energy recovery system and evaporative coolers, a supply air condition of between 12 °C and 19 °C at a chosen moisture content ($kg \cdot kg^{-1}$) can be achieved. The system can be used in all types of air conditioning systems, but is particularly effective in controlling dew point in radiant cooling either by chilled ceilings or fabric thermal storage.

The cooling and dehumidification capacity of a desiccant system is controlled by changing the temperature of the heater for the reactivation air. During the winter, when ambient conditions are low, the system operates in a heat recovery mode. Efficiencies in excess of 85 per cent can be achieved by using the desiccant wheel as a sensible and latent heat recovery unit in conjunction with the heat recovery wheel. This reduces energy consumption during the heating season.

A desiccant system may handle up to 50 per cent of the internal heat gain without any energy input by using only exhaust air evaporative cooling and the thermal wheel. At approximately 75 per cent of required capacity, the desiccant system provides 1 kW of cooling for each kW of regeneration heat input. However, at peak design load, the output of the desiccant system can drop to as low as 0.5 kW of cooling per kW of regeneration heat. As this is for a very small proportion of its operating period, detailed analysis may still reveal savings over more conventional systems. It is critical that the system-control philosophy is understood both by the designer and the operator of the system to ensure that the maximum potential savings are made.

The regeneration heater energy requirement is at its greatest during the summer months. However, in the case of a commercial office building, this is generally less than the winter heating load. The same equipment may therefore be used for both summer regeneration and winter heating.

(*d*) System enhancements

The system performance can be enhanced in terms of energy usage by:

- solar or gas regeneration of the desiccant
- when the desiccant is inactive, bypassing the wheel, so reducing system resistance and hence fan energy.

(*e*) Maintenance

The useful life of a desiccant material largely depends on the type of contamination in the airstreams they dry and the operational practice. A properly maintained system may last for 20 years.

(*f*) Capital and running costs

The capital cost of a desiccant plant is higher than that of a conventional plant, particularly for smaller systems (i.e. below 5 $m^3 \cdot s^{-1}$). This should be balanced against running cost and CO_2 production savings. A cost and environmental benefit analysis will be required for individual projects.

(*g*) Space requirements

The physical space requirement for the air handling plant is in the order of 20 per cent more than that for a conventional system, but savings can be made on reduced refrigeration plant depending upon the final air condition required.

3.3.3.3 Partially centralised air/water systems

These usually employ central plant to provide fresh air only. Terminals are used to mix recirculated air with primary air and to provide fine-tuning of room temperature. Examples include:

- fan coils
- chilled ceilings/chiller beams
- unitary heat pumps
- cooled surfaces
- sea/river/lake water cooling.

Both heating and cooling pumped water circuits are normally required to satisfy varying requirements.

Tempered fresh-air systems limit the humidification and dehumidification capacity. However, this is adequate for most applications and discourages attempts at unnecessarily close control of humidity, which wastes energy.

The AHU should be sized for fresh-air duty only to reduce energy consumption and heat recovery should be considered. Heat recovery from the chiller units can be employed to serve the terminal re-heaters. All partially centralised systems should have local interconnected controls to produce a demand-based response at the main plant.

Savings achievable due to reduced airflows must be balanced against the restricted free cooling from fresh air, the additional energy used due to higher pressures and local fan energy, and the energy required for heating and chilled water distribution pumps.

3.2.3.3.1 Fan coil units

(*a*) Description

A fan coil is a packaged assembly comprising coil(s), condensate collection tray, circulating fan and filter, all contained in a single housing. The fan recirculates air from the space continuously through the coil(s) either directly or via the void in which the fan coil is located. The units can provide heating as well as cooling of the space through the addition of a heating coil. Ventilation is usually provided by a separate central AHU or it can be drawn through an outside wall by the room unit itself.

Benefits provided by fan coil units include:

- significantly smaller ventilation plant and distribution ductwork than all-air systems
- individual zone control of temperature, if suitable controls are fitted
- high cooling capacity
- flexibility to accept future changes in load and layout (e.g. variable fan speed).

The fan energy requirement for central AHUs supplying fresh air only is normally considerably less than for an all-air system AHU. However, additional power is required by the fan coil units to circulate the room air. The centrifugal and tangential fans used in fan coil units typically have efficiencies far less than that of the most efficient AHU fans. Fan coil systems generally have relatively high maintenance costs and short operating lives (e.g. subject to bearing wear) and give rise to noise. The designer should be aware that there is potential for water leaks above the occupied space with fan coils installed in the ceiling void.

Fan coils are best suited to applications with intermittent medium to high sensible loads and where close humidity control is not required, e.g. offices, hotels, restaurants etc.

Fan coils are available in many configurations including:

— chassis units (normally horizontal units for mounting in ceiling void)

— cased units (normally vertical configuration for floor mounting against a wall)

— cassettes mounted through a false ceiling.

Vertical units require floor and wall space. Vertical units under windows or on exterior walls suit buildings with high heating requirements. Horizontal models conserve floor space but require adequate floor-to-ceiling heights to ensure that the void in which they are to be located is of sufficient depth.

Fan coil units with free cooling are suitable for some applications (on outside walls of low-rise buildings) and can provide additional economy of operation.

(*b*) Design

The types of fan coil system can be categorised as follows.

— *Two-pipe changeover*: a single coil is supplied with either chilled or heated water via a common water circuit connected to central heating and cooling plant via three-port changeover valves. This method is appropriate only where the summer/ winter transition is easily distinguishable, which is not normally the case in the UK.

— *Two-pipe non-changeover*: a single coil is supplied with chilled water only via a water circuit. Heating is normally provided either by a separate perimeter system or by electric heaters in the fan coil units. The use of electric re-heaters is not generally recommended for energy efficiency but may be appropriate where heat energy requirements are low, possibly due to high internal gains. Heating the ventilation air can also be used when heat energy requirements are low, although significant energy wastage through fan coil cooling of heated ventilation air can result if zone loads are not similar. Supply air temperatures are usually limited to a maximum of 45 °C.

— *Four-pipe*: four-pipe fan coils incorporate separate heating and cooling coils, fed by heating and chilled water circuits respectively (Figure 3.31). Ventilation air can be introduced in the following ways:

- distributed from a central AHU to stub ducts fitted with dampers located near to the fan coil inlets
- distributed from a central AHU to fan coil inlet plenums, although care must be taken to avoid the central unit fan pressure adversely affecting the fan coil fans
- distributed from a central AHU and introduced into the space separately via conventional air terminal devices
- drawn through an outside wall by the fan coil room unit itself.

The central AHU and distribution ductwork are normally sized to meet only the fresh-air requirements of the occupants and so are much smaller than those for an all-air system. Separate introduction of the ventilation air may have energy advantages in some applications by enabling the fan coils to be switched-off during mid-season when there is no requirement for heating or cooling.

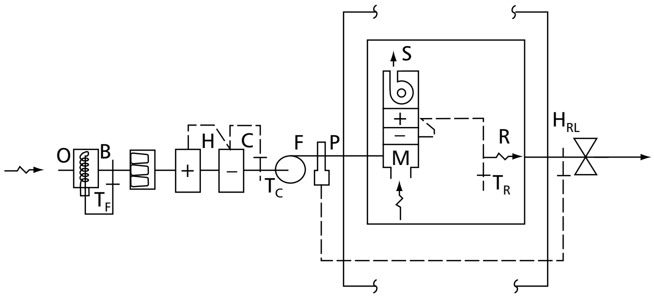


Figure 3.31 Four-pipe fan coil system

The central AHU is typically a full fresh-air system with off-coil control of heating and cooling coils, including humidification if required. The ventilation air will normally be supplied at a neutral temperature to minimise loads on the fan coils. This temperature may be scheduled down against outside air to provide an element of 'free' cooling in warmer weather. Where required, the unit may provide central control of humidity levels at the dictates of the supply of return air condition or a combination of the two. Refer to chapter 5 of CIBSE Guide H: *Building control systems* (2009) for detailed guidance on control.

The fan coils provide temperature control on a zone-by-zone basis. Depending on the chilled-water temperatures and space conditions, they can also provide some local dehumidification. Fan coil unit capacity can be controlled by coil water flow (waterside), air bypass (airside) or, occasionally, fan speed. Waterside control can be via four-, three- or two-port coil control valves. Airside control can be via air dampers with actuators supplied with the fan coil. It is potentially simpler to install and commission, and it can avoid maintenance problems caused by valves blocking but may require slightly larger units and can suffer from problems such as carryover. Airside control is generally less energy efficient than waterside control, as there is always a hot or cold coil operating simultaneously at full duty within the fan coil, and air leakage occurs at the coil dampers.

Water flow and air bypass can be controlled at the dictates of return-air or room-temperature sensors. Fan coil units can be supplied with integral return-air sensors. Control of room conditions can be coarse under certain conditions, as there may be a significant temperature difference between the ceiling void return-air temperature and the room temperature, resulting in a reduction of control accuracy. However this arrangement is regarded as an acceptable compromise for most applications because it is cheaper and easier to install than separately wired room sensors. Fan speed control may be automatic (BMS or power-enabled) or manual. Automatic control is usually on/off. Manual speed selection is normally restricted to vertical rooms units where there is access to the controls. Units are available with variable speed motors for step or modulated speed control. Room-temperature sensing is preferred where fan speed control is used, as return-air sensors will not give a reliable measure of room temperature when the fan is off. Room-temperature sensing may also enable the fans to be turned off if the room temperature is near to the set-point, thereby saving fan energy.

The size of the fan coil will normally be determined by the airflow required to cool the space and the water flow temperature. Where cooled ventilation air is introduced separately, fan coil sizes will be smaller. The fan coil cooling load should include dehumidification that may take place at the unit. This dehumidification is uncontrolled. Selection purely on sensible loads may lead to significant undersizing. In winter, humidified ventilation air may be dehumidified by the fan coils. Fan coil dehumidification can be reduced by running the chilled-water system at elevated temperatures. 'Wet' systems are based on flow/return chilled-water temperatures in the region of 6–12 °C. 'Dry' systems operate at higher temperatures in the region of 10–16 °C. This requires larger units to provide the same cooling but can improve the efficiency of the central cooling plant and provide increased opportunity for 'free' cooling.

Where air is returned via the ceiling void (see Figure 3.32), heat pick-up from light fittings may result in temperatures onto the coils being significantly higher than room temperature (ASHRAE, 2013c). This should be taken into account in unit sizing.

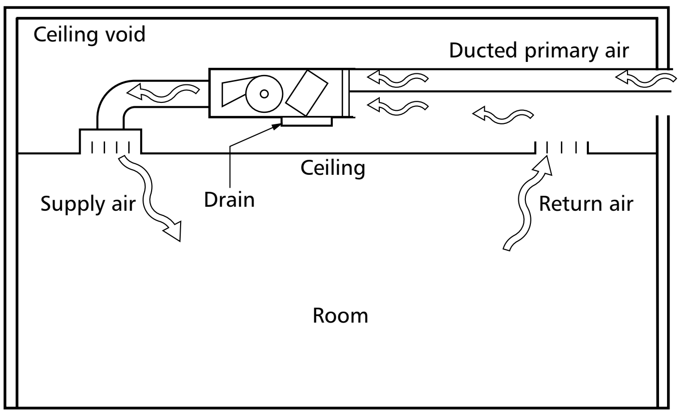


Figure 3.32 Ceiling void fan coil unit with separate primary air

Consideration should be given to avoiding conflict between heating and cooling to avoid unnecessary energy waste, particularly where a separate perimeter heating system is provided. One possible approach is to control the heating and cooling in sequence from a common temperature sensor; another is to ensure that there is an adequate dead band between heating and cooling. Care should be taken to avoid conflict between fan coil units that have separate control systems but are located in the same space. This can be overcome by operating several fan coils under a master/slave system from a master controller with sensor.

Where the ventilation air is used for heating, the supply air temperature may be scheduled against outside air temperature or to meet zone requirements. Increasing the supply air temperature may also be used in two-pipe changeover systems as the outside temperature drops to provide heating to zones with small cooling loads. Changeover to heating can then be delayed until all zones require heating. Fan coils provide the opportunity for early morning pre-heat with the primary AHU held off.

(*c*) **Construction**

Gravity condensate drain systems are preferred for 'wet' systems. Sufficient space should be provided in the ceiling void to achieve an adequate fall. Pumped condensate systems are available but will require maintenance and are inherently less reliable. It is considered good practice to provide condensate overflow systems on 'dry' fan coil systems to cope with accidental local moisture gains and as actual air psychrometrics can differ from the dry design situation. Condensate systems should be provided with suitable traps and air gaps. Drain pans should be fitted under each cooling coil (extending below the cooling valves) with a fall to a drain connection in the bottom of the pan. Drain pans should be removable for cleaning.

Attention should be paid to the combined inlet and casing noise levels and the discharge noise levels to ensure acoustic requirements are met. The information should be available from the fan coil manufacturer. For units installed in ceiling voids, return air grilles in the ceiling can be a particular source of noise. Return air grilles should not be grouped in such a manner that they accentuate noise levels.

Discharge ducting should be designed to avoid noise problems in the room. Generally, noise levels of fan coils will increase as external static resistance is applied across the unit, so external static resistance should be designed to be as low as possible. Allowance should be made for the use of flexible corrugated ducting and additional bends caused by site obstructions. There should be adequate return air grilles in the space being served, as modern partitioning systems can be comparatively airtight.

Filters are typically a pad type to G2/G3 (see CIBSE Guide B2, section 2.3.3.3 (CIBSE 2016b)) or a cleanable wire mesh or cardboard cartridge type that may offer maintenance advantages. Filters are primarily for protecting the coil fins from blocking and fans from build-up of dirt and debris.

Sufficient access should be provided for maintenance, particularly for the fan and motor, cleaning or changing of filters, and cleaning and inspection of the condensate drain pan and system (see section 3.2.4 for equipment requirements).

3.2.3.3.2 Chilled ceilings/chilled beams

(*a*) **General**

Conventional cooling methods such as fan coils and VAV systems provide cooling almost entirely by convective heat transfer. An alternative strategy is to provide cooling by a combination of radiation and convection using, for example, chilled ceilings. Such systems cool objects within the space, as well as the space itself. Although they are commonly known as radiant cooling systems, only 50–60 per cent (maximum) of the heat is transferred by radiation.

Chilled ceilings use chilled or cooled water as the cooling medium, normally 13–18 °C. There are many different types of chilled ceiling devices, but essentially they fall into three main categories (see Figure 3.33). These are:

— *radiant ceiling panels*: in which the cooling capacity is distributed across the ceiling using serpentine chilled-water pipework

— *passive chilled beams*: which have a more open structure and a heavier reliance on natural

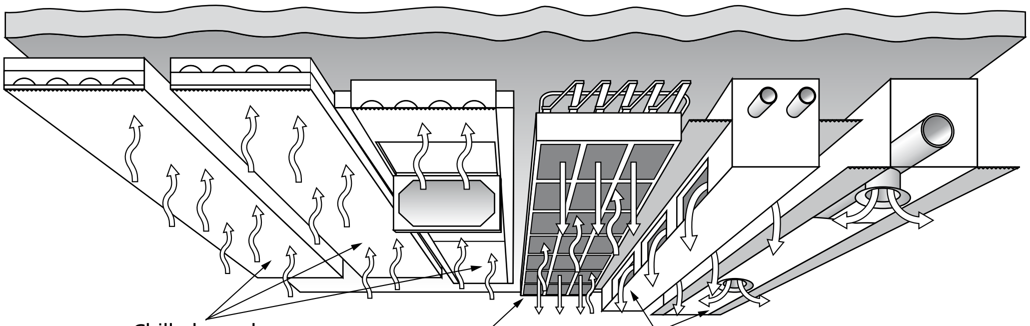


Figure 3.33 Chilled-ceiling categories

convective air movement; cooling is concentrated in finned coils similar to conventional heat exchangers

— *active chilled beams*: which are similar to the above but with the air movement through the beam being mechanically assisted.

With active chilled beams, fresh-air supply is an integral part of the beam, being induced by the central air-handling plant. However with passive chilled beams and panels, fresh air has to be introduced separately, either by mixed flow or more normally by displacement systems. Chilled beams can either be capped or uncapped, i.e. unconnected to the ceiling void or connected to the ceiling void. They can also be flush mounted to the ceiling or hung exposed from the ceiling, although care is needed to ensure that the required performance is achieved at the selected distance between the beam and the ceiling.

Chilled ceilings can be applied to both new-build and refurbishment projects. However, they are not suitable for situations where a close-controlled environment (i.e. temperature and humidity) is required. They may also be used in conjunction in mixed mode applications (Arnold, 1996). However, in this context, it is very important to consider condensation control.

(*b*) Design

Chilled ceilings and beams are often used in conjunction with displacement ventilation. Depending on configuration, cooling loads up to 120 W·m^{-2} may be achieved.

(*c*) Cooling performance

Cooling performance is highly dependent on the size and layout of chilled panels or beams. It is also a function of the room temperature. For cooling loads up to about 80 W·m^{-2}, displacement ventilation may be combined with chilled panels with the chilled panels providing 50 W·m^{-2} and displacement ventilation providing 30 W·m^{-2}. Providing this level of cooling from panels will require about two-thirds of the ceiling area to be covered. Passive chilled beams in combination with displacement ventilation can provide 70–120 W·m^{-2} of cooling.

For loads greater than 120 W·m^{-2}, active chilled beams are essential as they have a higher cooling capacity. Performance will be adversely affected by high heat loads directly below beams. It will also decrease with room temperature. For example, a system able to deliver 100 W·m^{-2} at a room temperature of 24 °C will provide no cooling at a room temperature of 14 °C. Care must also be taken to consider the possible effect of down-draughts from chilled ceilings delivering high cooling outputs. At these loads, physical testing or computational fluid dynamics (CFD) modelling of the design may be required. Further information on these systems is available elsewhere (Abbas, 1994).

The ratio of convective to radiative heat output for various systems is shown in Table 3.9.

Table 3.9 Convective and radiative proportions of heat output for chilled beams/panels

System type	Proportion of heat output (%)	
	Convective	Radiative
Active chilled beams	90–95	10–5
Passive chilled beams:		
— capped	80–90	20–10
— uncapped	85–90	15–10
Chilled panels	40–50	60–50

Systems can be used in conjunction with a low-quality source of cooling due to the relatively high cooling water temperatures required. Examples of this might be groundwater (see below: section 3.2.3.3.5 'Sea/river/lake/aquifer water cooling') or cooling towers. This will increase their coefficient of performance (Butler, 1998). As cooling is supplied within the space, this limits the requirements for the ventilation system to provide fresh air, thus also saving fan energy.

(*d*) Combination with displacement ventilation

Tests have shown that when chilled ceilings are combined with displacement ventilation, there is more downward convection than is the case with displacement ventilation alone, although upward convection should still be dominant in the vicinity of the occupants. The flow pattern resulting from chilled beams may give a more mixed condition in the occupied zone than chilled panels. Similarly, uncapped passive beams may result in stronger downward convection currents than capped passive beams.

A physical testing and CFD modelling study (Davies, 1994) shows that when displacement ventilation without chilled ceilings is employed, the airflow patterns are chiefly upward when the internal thermal loads are equivalent to the cooling capacity of the displacement ventilation system. On condition that the supply air temperature and air velocity are maintained within recommended values, a high order of thermal comfort and air quality are usually obtained. The addition of chilled beam devices to offset higher internal thermal gains progressively erodes the predominant upward airflow region as thermal loads are increased. When the cooling load of the chilled ceiling devices is about three times that of the displacement ventilation system, the flow pattern is similar to a conventional mixed airflow system, except in the vicinity of heat sources where upward convective plumes entrain air from the displacement cool air layer at floor level. When displacement ventilation is employed with chilled ceilings the radiant cold panels slightly increase the depth of the mixed warm and contaminated upper region but do not affect the displacement airflow characteristics of the lower part of the room. The environmental thermal comfort conditions, however, are generally of a very high order.

(*e*) Control strategies

Many of the advantages offered by chilled beam and ceiling systems are due to the simplicity of these systems, since they are inherently self-compensating in their thermal cooling. It is important that this level of simplicity is also maintained within the control system used, which is in many ways akin to a simple radiator heating system.

Ideally, beams should be controlled in groups using two-port, two-position control valves. These can be pulse controlled to vary the length of time open depending upon the variance between measured room temperature and set-point.

Most systems now have speed control (static inverter) on the pumps in order to maintain a constant system pressure as the system volume flow rate requirement varies. Where speed control is not being used, a simple pressure bypass valve on the end of the circuit should be used.

System controls are normally set up to mimic those of a fan coil system, i.e. two- or four-port valves on the outlet, either on/off or infinitely variable controller and a room sensor. The control strategy should ensure that condensation risk is eliminated (see the following paragraph). For central control, a three-port valve is needed to regulate the inlet water temperature. If two-port valves are used in rooms, then a buffer tank or low loss header between the chiller and pipework will ensure a constant flow rate to the chiller. A bypass valve at the end of each branch decreases the pressure in the pipework and is particularly important with a two-port valve system. This also ensures that a constant chilled-water supply is available.

(*f*) Condensation risk (Martin and Alamdari, 1997)

The avoidance of condensation on the surface of chilled panels and beams has been a major design issue in the UK, with fears over its relatively wet climate. It has been assumed that without dehumidification of the outside air, 'office rain' could occur.

Condensation detection should always be incorporated into the chilled beam control system. This should be considered as being ideal for active beam systems but essential for passive beams and chilled ceilings. In most buildings, it is unlikely that condensation will occur within an active beam system but it can occur as a symptom of a fault within the system. Occasions when coil condensation can occur include during commissioning when chilled water is being balanced before the chilled control/mixing system has been commissioned, if windows are left open or even broken, AHU dehumidifier pump failure or human error resulting in chilled-water temperatures being reduced.

Condensation detection should be by direct dew-point sensing using a device clamped to the pipework. It is not practical to measure independently the room temperature and humidity in order to calculate the dew point. The inherent lack of accuracy found in most humidistats is acceptable when measuring in order to maintain the humidity within a wide band for comfort. However, they should not be used for dew-point calculation.

When condensation is detected, the chilled water supply temperature should be ramped up one degree at a time to ride above the dew-point. Alternatively the chilled water should be shut off completely and alarms raised on the building energy management system (BEMS), since the condensation may be a symptom of a fault in the system.

Condensation will start to form if the ceiling surface temperature falls below the room air dew-point temperature. Various condensation avoidance strategies have been developed to minimise or eliminate the risk of this condition occurring. In principle, the selection of an appropriate control strategy and set-point should not allow the development and formation of condensation. Equally, it should not unduly limit the cooling output from the ceiling, nor its ability to be used within a mixed mode application.

The chilled-water pipework between the panels or beams should be insulated and provided with a vapour barrier to minimise condensation risk, as these surfaces will be cooler. This also increases the ceiling output, as the difference between the panel/beam temperature and the room temperature is larger. Alternatively, if the chilled water temperature is maintained at the same level, the room dew point can be allowed to increase to reduce the requirement for dehumidification of the ventilation supply air.

Control of the chilled-water temperature provides an effective means of avoiding condensation. Although it is more energy efficient than using supply air dehumidification control, it may result in a loss of comfort conditions if the room dew point becomes too high.

These techniques can be used in combination, for example dehumidification with supply air temperature control. Note that measurement of the room dew point temperature through a combination of the dry bulb temperature and relative room humidity measurements requires accurate sensors that are regularly recalibrated.

Condensation can also be avoided by reducing the room dew-point temperature by reducing the supply air temperature, forcing the cooling coil to increase capacity resulting in increased dehumidification. This may increase the risk of draught, particularly when using displacement ventilation.

(*g*) Maintenance

Chilled-water pipework is present throughout the building. Care must be taken with zoning and provision of sufficient drainage points. Providing that the system remains problem free, maintenance costs should be lower than those for conventional systems.

(*h*) Noise

Compared with fan coils and VAV systems, chilled-ceiling systems do not generate sufficient background noise to provide sound privacy. It may be necessary to increase the sound insulation in the partitioning system or increase the height of any partitions. Consideration should also be given to the use of electronic sound conditioning (broadband and characterless).

(*i*) Performance monitoring

Guidance on modelling the performance of chilled ceilings in conjunction with displacement ventilation is available at both early stage design and detailed design.

(*j*) Construction

— *Ceiling layout*: in practice, the high heat gains in modern office spaces are served by chilled beams and chilled panels, or chilled beams alone. It is important to consider the ceiling layout in terms of its effect on the overall performance, e.g. the positioning of beams at the perimeter areas, with panels being used in the inner zones (see Figure 3.34). It is also important to consider integration with the light fittings.

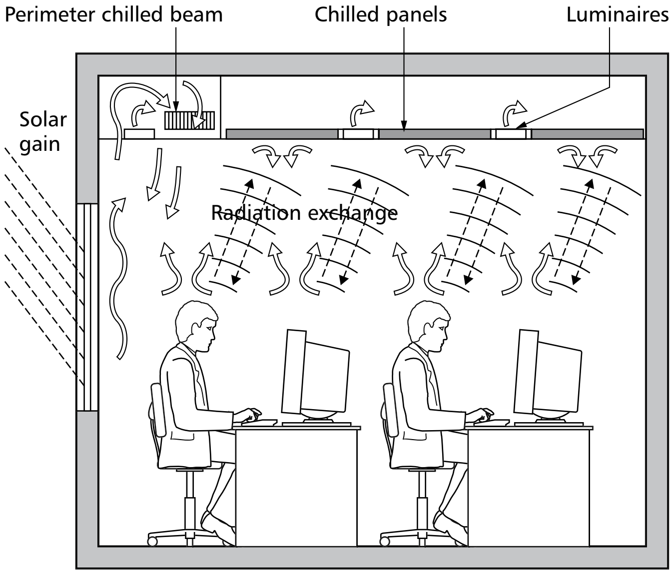


Figure 3.34 Typical ceiling layout incorporating chilled panels and chilled beams

— *Space allowances:* the requirement for ductwork space and associated ventilation plant can be reduced in comparison with conventional systems. However, space is required for the central cooling and distribution systems. For active chilled beams, an air supply must be allowed for. Passive chilled beams require space for overhead air recirculation and beam stack height below.

A floor-to-slab height of at least 2.4 m is required for passive and active chilled beams to ensure a high degree of thermal comfort. The height limitation must be determined from case to case depending on expected heat loads and the features of the beams such as their width and depth.

3.2.3.3.3 Unitary heat pumps

(*a*) Description

A unitary heat pump is a single-unit heat pump that can operate in either heating or cooling mode whilst using air or water as a heat source/sink (see Figure 3.35).

They are classified according to the type of heat source and the heat distribution medium used, e.g. air to water, air to air etc. Table 3.10 lists examples of heat source and distribution systems.

Table 3.10 Examples of heat source and distribution systems

Heat source	Heat distribution medium	Typical distribution system
Air (ambient, heat recovery)	Air	Air diffusers Individual units: — dehumidifiers
Water (surface, ground, industrial waste, process cooling water)	Water	Radiators: — underfloor coils — fan coils — induction units
Ground (closed loop)	Water	Radiators: — underfloor coils — fan coils — induction units

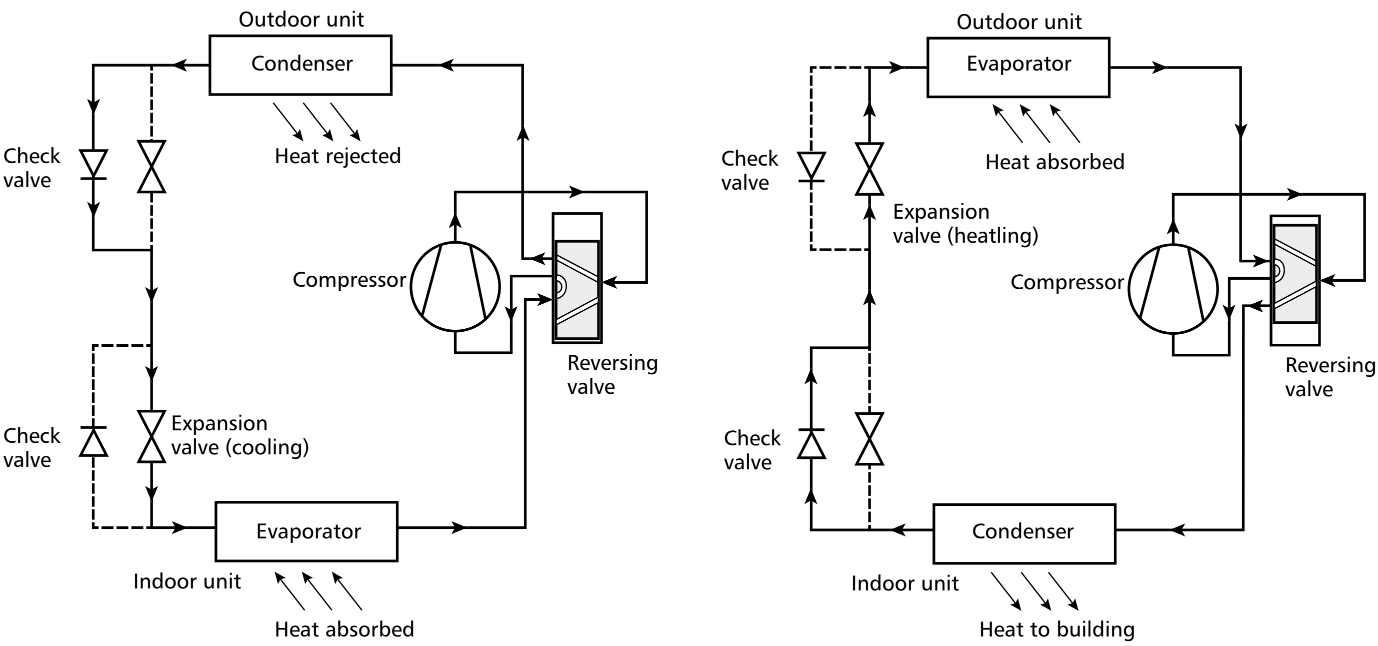


Figure 3.35 Schematic of reversible heat pump; cooling mode (left), heating mode (right)

Under certain circumstances, the heat from a source is transferred to the heat pump by a secondary medium and an intermediate circuit. The secondary medium is used to prevent cross-contamination and to protect the overall system in case of breakdown (e.g. pipe breakage) or freezing. The secondary medium can be brine or glycol, or a similar low-temperature medium.

Each of these systems can be applied as heating only, reverse cycle heating/cooling or simultaneous heating/cooling in the following situations:

— *commercial*: offices, shops, hotels
— *domestic*: institutional residential buildings, dwellings, conservatories
— *recreational*: leisure centres, pubs and clubs
— *industrial*: factories, warehouses and processing
— *educational*: schools and further education.

(*b*) Design

The operational efficiency of systems can be enhanced by employing:

— heat recovery from air, steam or water
— renewable primary energy sources, e.g. solar, wind, water, ground, geothermal etc.

(*c*) System performance evaluation

Like other refrigeration systems, the performance of heat pumps is expressed as a coefficient of performance (CoP). CoP is the ratio of energy or heat output to the energy input.

Theoretical CoP

For the vapour compression cycle where a cooling output is considered, CoP_R is the ratio of refrigeration effect to the work done by the compressor.

For the vapour compression cycle where a heating output is considered, CoP_H is the ratio of heat from the condenser to the work done by the compressor, i.e:

$$\text{CoP}_H = \frac{\text{Enthalpy change due to condensation of vapour}}{\text{Enthalpy change due to compression of vapour}}$$

Hence, from Figure 3.36, the theoretical coefficient of performance is given by:

$$\text{CoP}_H = \frac{H_3 - H_4}{H_3 - H} \tag{3.1}$$

The theoretical full load CoP gives an indication of the viability of a particular heat pump option and a full economic assessment is always necessary in final selection of equipment.

Seasonal performance factors (SPFs) may be used to account of variations in energy source conditions and any additional energy usage within the systems, pumps, defrost, distribution and running hours.

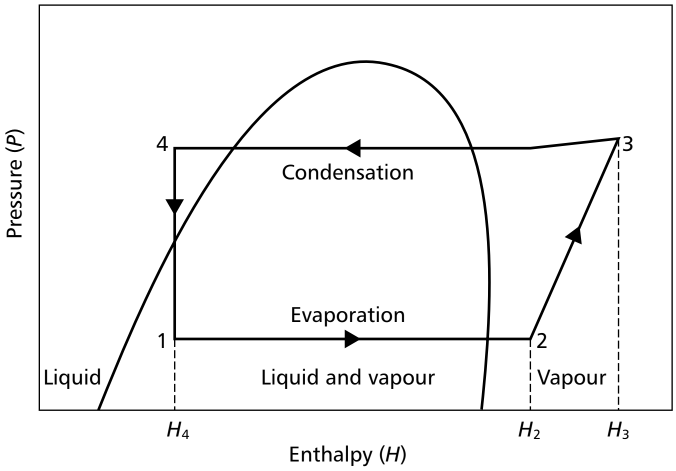


Figure 3.36 Pressure–enthalpy diagram for the vapour compression cycle

Practical CoP

The CoP for the heat pump itself is termed the appliance CoP. This is useful when comparing one heat pump with another. When considering heat pumps for heating, it has become accepted industry practice for the input energy to include energy used by outdoor fans or pumps required by the low temperature source in addition to the energy used by the compressor

The appliance CoP should not be used to determine the running costs of an installation but only as one of the criteria considered when selecting a particular heat pump.

Overall system efficiency and seasonal CoP

Overall system efficiency can be established and expressed as a CoP by including the energy input and output to supplementary heating and distribution fans or pumps as part of the total energy input and output. It is not a true CoP because items that are not part of the heat pump operation are also considered. But it does give an indication of the total energy used by the system compared with the heat output, enabling an estimation of consumption and running costs to be established.

The seasonal CoP of a heat pump is defined as the appliance CoP averaged over the heating season. The values of CoP are dependent on compression ratios, temperatures, cycle arrangements, source and distribution temperatures and will also vary depending on which of the coefficients of performance is being considered. Table 3.11 shows the variation of CoP values for a typical vapour compression air-to-air heat pump using ambient air as a source.

Table 3.11 Variation of CoP for a typical vapour compression air-to-air heat pump

Theoretical CoP		Actual CoP		
CoP_R	CoP_H	Appliance	System	Seasonal
4	5	3.0	2.3	2.5 to 3.0

(*d*) Control strategy

Correct control of the heat pump system is vital to maintain performance. Particular care must be taken with the heat pump system to avoid rapid cycling, as this is both harmful to the equipment and inefficient in energy usage. Controls can be divided into two groups: those installed for unit protection by the manufacturers and those for the correct operation of the unit and system.

(*e*) Critical design factors

To provide correct selection and application of the heat pump systems to ensure the operation is at maximum efficiency, consider the following:

— designing for heating only, cooling only or heating/cooling

— designing to suit heat source/sink (minimising refrigerant ΔP increases efficiency)

— high overall operational efficiency

— selecting a suitable primary power source

— environmental considerations

— controls for stand-alone systems, multiple systems or building management systems

— simplicity of the design (avoid over-complication).

3.2.3.3.4 Cooled surfaces (floors and slabs)

(*a*) Description

Cooling to the space is provided via radiation and convection heat exchange with cool exposed surfaces, usually floors and ceilings. A pipe network is used to cool the surface. This may be attached to a panel-type construction or imbedded in a slab if the slab surface is exposed. The panel-type systems are generally thermally lightweight systems that have a rapid response to load changes. The slab systems are heavyweight with the thermal capacity to store cooling but a slow response to load changes.

Cooled surfaces are most suited to buildings with low to medium heat gains and summer temperatures are permitted to rise. Sensible cooling only is provided. The cooling capacity of the system is a function of the space/surface temperature differential. Relatively high (e.g. 18 °C) water temperatures are typically used for cooling, permitting the use of low-grade cooling direct from sources such as cooling towers, dry air coolers (see Figure 3.37) or aquifers. This helps to avoid or reduce the requirement for mechanical refrigeration. The system may also be used for heating during winter. Indeed, most floor systems are selected for heating rather than cooling.

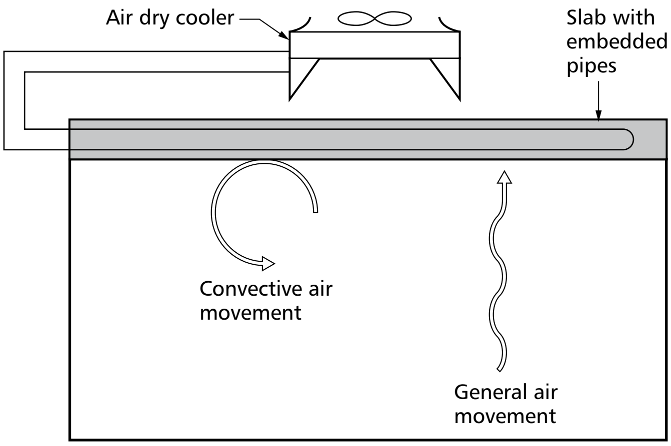


Figure 3.37 Cooled surface using dry air cooler

(*b*) Design

The cooling effect of the surface is a function of the surface–space temperature differential and the surface area. It can be estimated using equations for surface heat transfer given in CIBSE Guide A: *Environmental design*, Appendix 5.A8*, equation A8.74 (2015a). Manufacturers' data should be referred to for accurate performance data.

Permissible surface temperatures can be constrained by comfort requirements, minimising condensation risk and control practicalities. For comfort, radiant temperature asymmetry should be less than 5 K for cooled floors, 10 K for cooled walls and 14 K for cooled ceilings. Minimum surface temperatures should be such that they do not cause a significant condensation risk. This risk can be reduced if the system is being used in conjunction with a mechanical ventilation plant providing humidity control.

Increases in output for cooled ceilings may be achieved by profiling the surface. This provides a larger area for heat transfer. Convective heat exchange with the air in the space will increase approximately in proportion to the area. Radiant heat transfer will normally be limited if the profiling has a similar overall exposed surface area to that of a plane surface when viewed from the occupied space. The geometric exposure can be calculated using form (or shape) factors (Holman, 1986).

Exposure of soffits raises a number of issues that should be considered including aesthetics, acoustics and integration.

Design issues for the pipework system include the location and spacing of pipes. For panel-type systems, pipes are generally spaced 100–300 mm apart (ASHRAE, 2012b). Wide spacing under tile or bare floors can cause uneven surface temperatures. For slab systems, pipework may typically be located 40–100 mm below the surface at 150–450 mm spacing (ASHRAE, 2012b). To achieve effective storage and heat conduction to the surface, see Figure 3.38. Optimum values can be evaluated using conduction models.

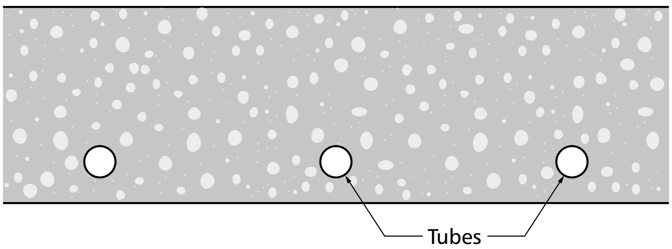


Figure 3.38 Cooling pipework in structural concrete slab

Surface finishes should not be insulative and should have high emissivity. Insulation to prevent perimeter and back heat losses should be considered.

Flexibility in operation (e.g. heating in perimeter zones with cooling internally) and future adaptability should be provided by suitable zoning of the pipework layout and the configuration of the pipework distribution system. The length of pipework runs should be determined to suit zoning and to avoid excessive pressure drop.

* http://www.cibse.org/knowledge/Guide-A-2015-Supplementary-Files/chapter-5

For lightweight systems, response to load changes will be fairly rapid. For slab systems having high thermal inertia, reaction to load changes will be slow. This should be reflected in the control strategy adopted. The slab temperature may be controlled to within the normal space comfort band to minimise the risk of overcooling, e.g. 20–22 °C. This can be achieved with cooling water temperatures in the region of 18 °C. Cooling water may be circulated during the day and/or night. This will be determined by a number of factors including the following.

— *Output required*: high outputs may require top-up cooling during day as well as cooling at night.

— *Cooling source*: the cooling source may be more energy efficient at night or only able to produce sufficiently low temperatures at night.

— *Energy tariffs*: cheaper tariffs may be available at night favouring night-time operation.

(*c*) Construction

Because of the inherent problem of access to repair leaks, considerable care should be taken during the construction process to minimise the likelihood of their occurrence. Plastic or plastic-coated pipework is normally used to avoid corrosion problems and silting. Longer lengths may also be used, which reduces the number of joints and associated risk of leakage.

Distribution to the pipes is often via supply-and-return manifold headers. Single continuous lengths of pipes between the supply-and-return headers are preferred to avoid joints and increasing the risk of leakage. The pipes are normally arranged in a serpentine configuration.

For panel-type systems, there are a variety of construction methods available (see Figure 3.39) including:

— support via joists or battens

— attachment to the underside of the floor

— support in a floating floor panel with suitable grooves for laying the pipework.

For floor systems, insulation should be considered to minimise downward heat flow. Providing a reflective finish below the pipework will help to promote upward heat flow. Heat diffusion and surface temperature uniformity can be improved by the addition of metal heat transfer plates, which spread the cooling beneath the floor.

For slab systems, construction can either be in-situ or in pre-cast units. Pre-casting in factory conditions may be preferred from the point of view of minimising the risk of leaks. The pipework may be supported by the steel reinforcement cage or on the bearing slab for floating slab applications (see Figure 3.40). The pipe ends may be located in a connecting box fixed to the shuttering with its opening facing upwards or downwards as required for connection to the distribution system (see Figure 3.41). It may be necessary to pressurise pipes to stiffen them until the concrete has set. During construction, pipework terminals should be capped to prevent debris getting into the pipes. Slab temperature sensors should be installed at a depth that is representative of the storage capacity of the slab. Temperatures close to the surface may be influenced by local effects (e.g. air blowing across the slab, hot plumes rising from equipment).

3.2.3.3.5 *Sea/river/lake/aquifer water cooling*

Water abstraction and return licences governing the water volume flow and ΔT may be required.

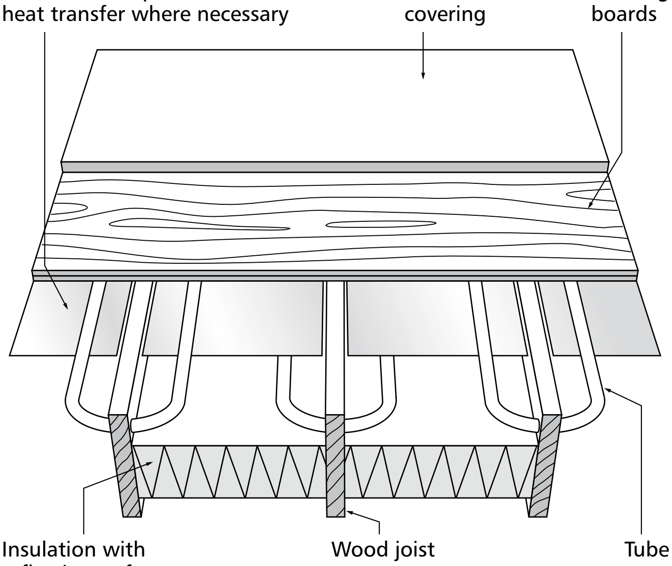


Figure 3.39 Cooling pipework in sub-floor

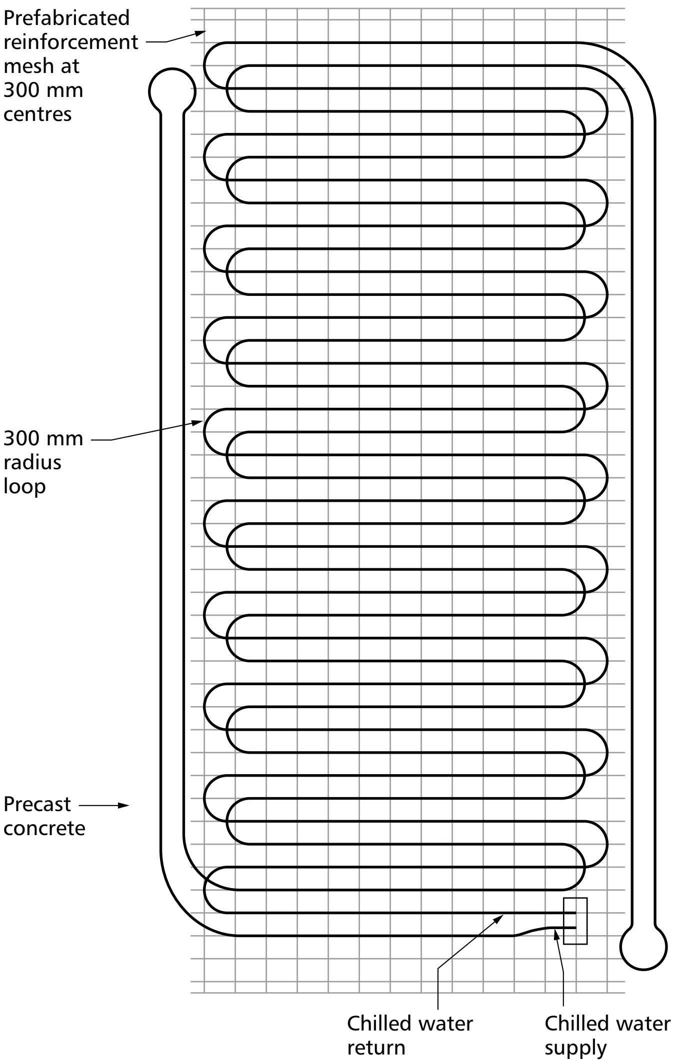


Figure 3.40 Example of cooling pipework supported by reinforcement cage

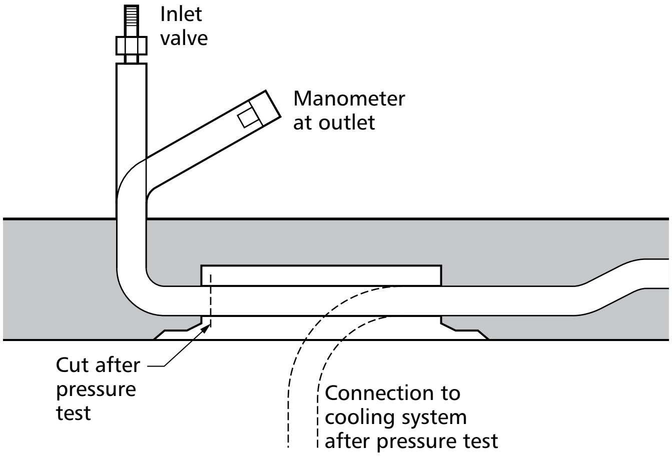


Figure 3.41 Pre-mounted connection box for cooling pipework

(*a*) Description

Water is pumped through an open-loop system and cooling extracted via a heat exchanger (see Figure 3.42). This cooling can either be used directly or indirectly. Direct applications include cooling the space (e.g. via chilled beams/ceiling, water-cooled slabs) or the supply air. Examples of indirect use are as condenser water or with heat pumps to provide heating and cooling. In winter, warm water returning to the heat exchanger can be used to pre-heat incoming fresh air. The primary benefits are free cooling and low operating costs. Such systems are restricted to buildings that are located near a water source with suitable temperatures and thermal capacity. (*Note*: small lakes can warm up significantly during the summer.)

(*b*) Design

Key design parameters include:

- the depth from which water is drawn
- water temperature
- water flow rate
- water filtration.

Generally, the greater the depth, the lower the water temperature. However, pump head will also increase with depth, and so the cooling benefits will need to be balanced against pump energy requirements. The water temperature will also determine the function for which the water can be used, i.e. direct cooling or condenser water cooling.

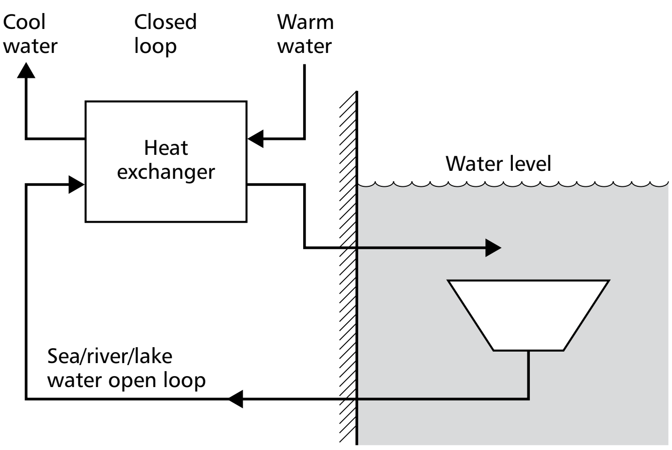


Figure 3.41 Schematic of sea/river/lake water cooling system

Equations for surface water heat transfer are provided in chapter 3 of CIBSE Guide C: *Reference data* (2007).

The water flow rate required will be determined by the water temperature and the cooling loads or heat rejection requirements. Operation of the system will generally be at the dictates of the cooling system. Temperature limits may be used to determine the operating mode, e.g. free cooling below, condenser water cooling above.

(*c*) Construction

Suitable materials should be selected and measures undertaken to minimise fouling, biological growth and corrosion, particularly in marine environments.

Possible corrosion-resistant materials include titanium and treated aluminium. Screens and filters should be provided to protect against fouling of the heat exchangers. Cathodic protection can be used to impede marine growth and corrosion. (*Note*: chlorine has been used to minimise biological growth but is harmful to the environment and marine life.)

3.2.3.4 Local systems

These include:

- split systems and variable refrigerant flow systems
- room air conditioner 'through the wall' units
- underfloor air conditioning systems
- night cooling and thermal mass.

Local systems can provide filtration, comfort cooling and heating, but not humidification.

3.2.3.4.1 Split and variable refrigerant flow systems

(*a*) General

Split systems (Jones, 2000; Oughton and Wilson, 2015) and variable refrigerant flow (VRF, also referred to as variable refrigerant volume or VRV) systems are room air-conditioner units, or small AHUs, incorporating a direct-expansion cooling coil, a filter and a fan to recirculate room air.

These are connected to a remote air- or water-cooled, condensing unit(s) via low-pressure vapour and high-pressure liquid refrigerant lines. The external units are normally roof or wall mounted and contain compressors, heat exchangers and air circulation fans.

In cooling mode, the external unit heat exchangers function as a refrigerant condenser, producing liquid that is circulated to the remote room units. This passes through the coils, absorbs heat, evaporates and the gas is returned to the compressors. When operating in a heating mode the functions are reversed.

Two-pipe and three-pipe systems operate as heat pumps and can incorporate low temperature hot water and over door air curtains as part of an overall building heating and cooling strategy. More complex systems can offer simultaneous heating and cooling and heat recovery ventilation, these tend to offer higher overall system efficiency. Applications include small commercial, retail and small, medium and large offices

(*b*) Performance

The maximum capacity of an external unit is of the order of 25 kW although VRV/VRF systems can generally be installed in parallel up to 150 kW.

The number of room terminals (indoor cooling coils) is dependent on the external unit capacity and system design diversity.

The room terminals typically have cooling/heating outputs in the range of 2.5–14 kW and may be served by one external unit or a combination of multiple external units.

(*c*) Control

Control options include:

- standard time clock control
- complex time and zone controls
- BMS integration via protocol gateways
- leak detection
- complex system for monitoring for performance and fault prognosis/diagnostics.

(*d*) Maintenance

Systems must be maintained to manufacturers requirements to ensure system performance and operating life. Indoor unit filters must be cleaned regularly as well out the condensing unit coils. Further checks on a system must be carried out by an F-Gas registered engineer and would fall in line with the standard procedures adopted for all air conditioning and heat pump equipment.

3.2.3.4.2 Room air conditioners (through-wall packages)

(*a*) General

Also known as window units and through-wall air conditioners, these packaged units incorporate a room airside evaporator (direct expansion cooling coil), an outside air-cooled conditioner, a compressor and an expansion device. Winter heating is often by electric coil, although some manufacturers offer a low-pressure hot water coil. Where appropriate, moisture penetration may be minimised by the use of high-efficiency weather louvres. Dust penetration may be minimised by the use of sand trap louvres. Their main advantage is that they are self-contained, requiring only an appropriate electricity supply and an outside wall in which to be mounted, normally at low level. No plant space is required. It is also possible to install heat pump versions for increased energy efficiency. A drawback can be noise level. Manufacturers' literature needs careful interpretation and corrections to ratings will normally be required to account for UK conditions.

(*b*) Control

In their basic form, these units offer the crudest form of air conditioning. Room occupants normally have control over the units through switching of the compressors. However, this gives consequent swings in room temperature, humidity and noise level.

3.2.3.4.3 Underfloor air conditioning systems

(*a*) General

Conditioned air is supplied from a number of vertical downflow fan coil units into the raised floor void, see Figure 3.43. The units can be positioned either within the office space or in service cupboards.

The air return path depends upon the room configuration but is most commonly direct or via a ceiling plenum.

To avoid draughts, air is not introduced to the space directly, but instead via a number of floor-mounted mixing boxes. Each of these incorporates a small room-air recirculation fan, together with modulating damper, which mixes in as much, or as little, underfloor air as necessary to meet the requirements of the immediate local area. These are often referred to as 'fan tile units'.

(*b*) Performance

Each zone unit typically circulates up to 1.5 m^3/s of air, with an associated cooling or heating capacity of around 30 kW, usually sufficient for a typical office 'zone' of up to 250 m^2.

Each fan tile unit typically circulates up to 0.15 m^3/s of air, with an associated cooling or heating capacity of around 3 kW, usually sufficient for a floor area of up to 25 m^2.

Air velocities are kept very low, whilst variable speed fans allow optimisation of specific fan power providing efficient part-load performance. Typical ambient noise levels are around NR35, avoiding the need for airside attenuators and associated increase in fan power.

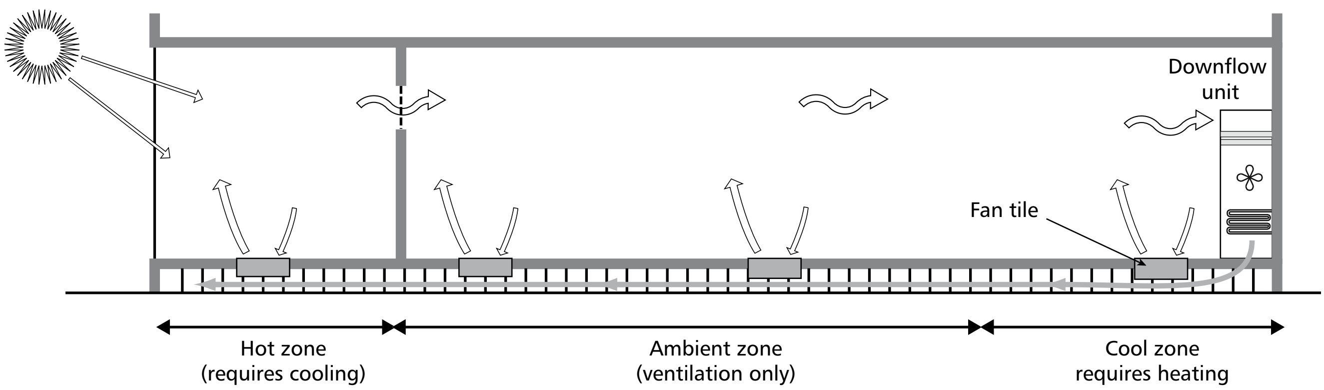


Figure 3.43 Underfloor air conditioning system

(*c*) Application and control

Requiring only a domestic power supply, the floor-mounted fan tile units can be considered effectively mobile and are easily moved around as office layouts change. This provides an extremely high degree of flexibility for different space allocations, or tenant requirements within a building. It can also allow occupants to control conditions within individual spaces, often simply from their own computers.

Heating and cooling coils within the zone units are very generously sized when compared with conventional AHUs or fan coils, which makes this system ideal for low energy heat rejection and heat scavenging or reclaim schemes.

Fresh-air free cooling can be easily incorporated where large cooling loads are present, and the floor slab is used as a large thermal mass to provide night time precooling in summer or off-peak pre-heating in winter.

(*d*) Maintenance

Compared with traditional fan coils, the relatively small number of zone units required simplifies maintenance. Only front access to cabinets is required and there are no components installed at high level.

A number of spare fan tiles are usually kept available on-site to quickly and easily replace any problem units. Any work on these, other than routine checking and cleaning, can thus be carried in a workshop as and when convenient.

3.2.3.4.4 Night cooling and thermal mass

(*a*) Description

Night cooling in combination with a thermally heavyweight building can be used as a means of avoiding or minimising the need for mechanical cooling in buildings. During the summer, ambient air is circulated through the building at night, cooling the building fabric. This stored cooling is then available the next day to offset heat gains.

Interaction between the mass and the air, see Figure 3.44, can be (*a*) direct via exposed surfaces in the space or (*b*) indirect where the air is passed through floor voids, cores or air paths.

For direct systems with exposed mass, heat transfer is both by radiation and convection. Indirect systems rely solely on convective heat transfer.

For natural ventilation, because of the low pressure drops available to drive the airflow, interaction between the thermal mass and the air is normally direct via exposed surfaces in the space. Most solutions use exposed soffits. External walls and partitions can be used to add thermal mass. Carpeting and/or a false floor will normally limit floor exposure.

Where mechanical ventilation is provided, direct and/or indirect interaction may be used. Additional fan energy will be expended to introduce cooling at night. For large systems (i.e. with pressure drops greater than 1000 Pa) this may exceed the mechanical cooling and pump energy saved (Barnard, 1994).

System pressure drops should be minimised to maximise energy efficiency.

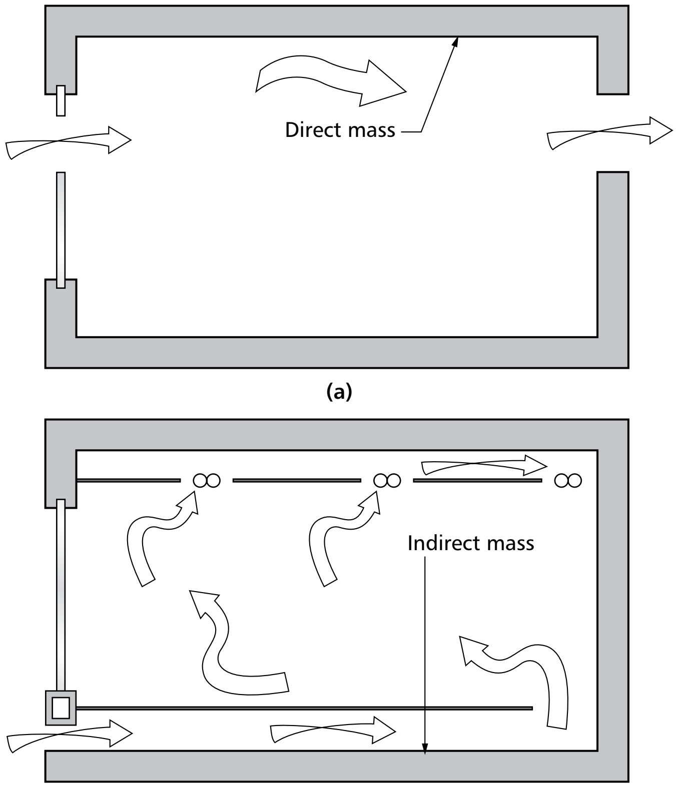


Figure 3.44 Heat transfer; (a) direct and (b) indirect

As the cooling provided is a function of the temperature difference between the thermal mass and the space, night cooling is most suited to buildings where the temperatures are permitted to rise during peak summer conditions. In the UK, night-cooled solutions can provide up to 50–60 $W \cdot m^{-2}$ depending on thermal storage in the building slabs.

Where mechanical cooling is provided, night cooling of the building mass can be by introducing outside air or by using the mechanical cooling system when outside temperatures are high. Scope exists in many mechanically cooled buildings for the controlled use of the building mass as an energy store.

It can provide the following benefits:

- a reduction in mechanical cooling requirements during the occupied period
- the opportunity to take advantage of cheaper night-time electricity tariffs
- improved comfort in low-capacity systems.

However, it is often the case that there is a requirement for space temperatures to be maintained below a maximum in the summer and not be permitted to rise. This will limit the benefit of night cooling in reducing mechanical cooling requirements.

(*b*) Design

Figures 3.45 and 3.46 below illustrate a number of design approaches that may be used for direct and indirect solutions. Specific design issues are addressed in the following sections and include:

- thermal storage performance
- conflict with air heating/cooling

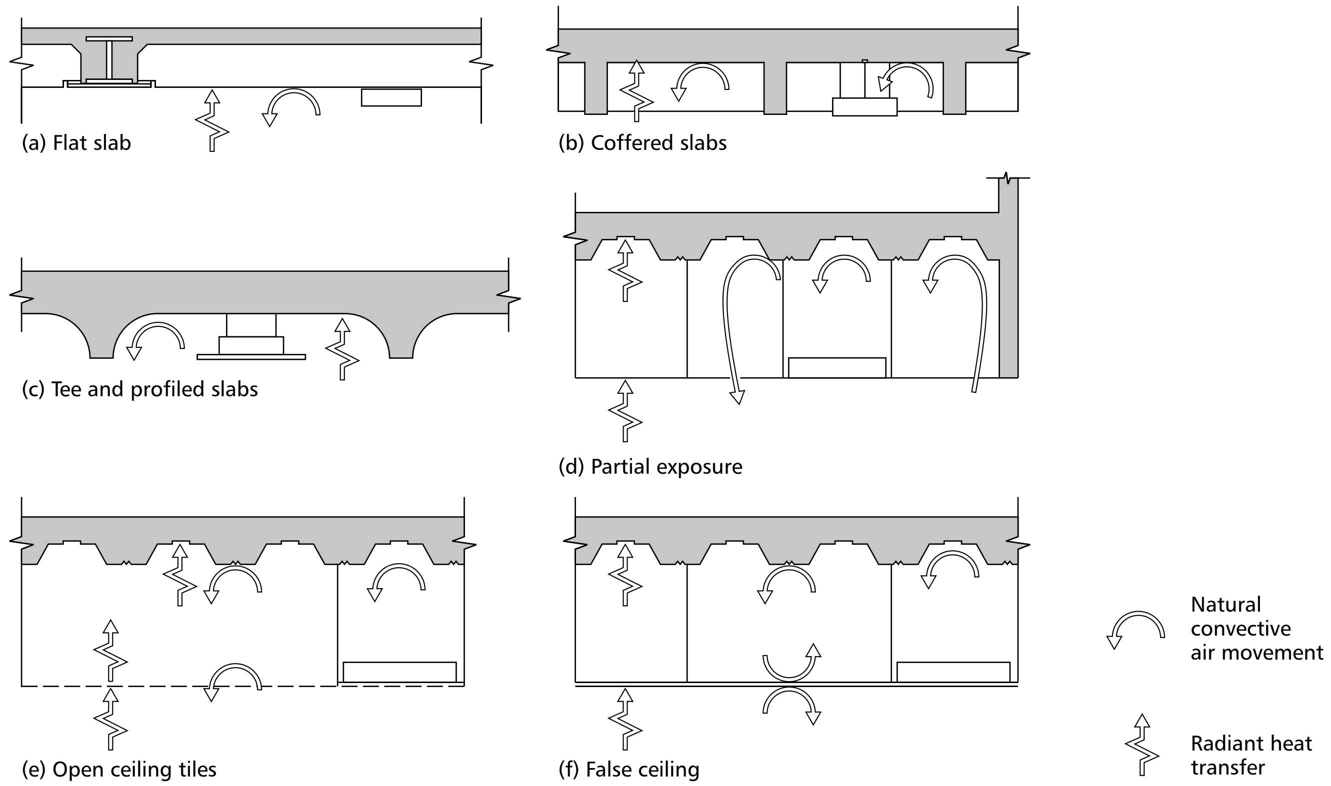


Figure 3.45 Design details for exposing thermal mass

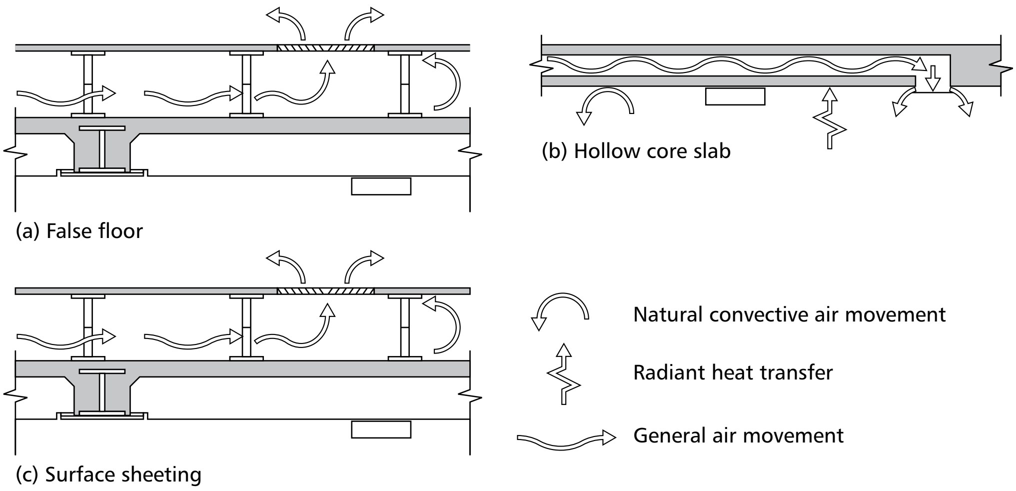


Figure 3.46 Design details for indirect interaction

— aesthetics

— acoustics

— integration

— control strategy.

Two common problem areas for design are top floors and corner/perimeter offices. Economic or structural constraints may mean that the roof cannot be designed to incorporate the same level of thermal mass as the other floors. In these cases it may be possible to add mass via other elements, or an alternative design strategy may need to be considered. For corner/perimeter offices high heat gains and losses may mean that supplementary cooling or alternative design strategies may be required.

It should also be recognised that exposing thermal mass may lead to a significant increase in the heating demand during the winter months due to the thermal mass acting as a store for unwanted infiltration and conduction heat losses at night (Braham et al., 2001). In contrast, there can be a reduction in heating demand during the summer as excess heat from internal gains can be stored for later use more effectively by the heavier constructions—a form of heat recovery. For lighter constructions, the excess heat will tend to be rejected to the outside rather than stored.

(*c*) Thermal storage performance

The thermal storage performance of a building element is dependent on two key factors:

— the ability of the element to conduct and store the thermal energy

— rate of heat transfer between the element and the air.

For most floor construction types, the ability of the concrete slab to conduct and store the thermal energy is superior to the rate of surface heat transfer. Therefore the surface heat transfer characteristics generally determine the thermal storage performance of a concrete floor slab.

Direct systems

For direct systems with elements exposed to the occupied space (e.g. the underside of a slab), surface heat transfer is by a combination of radiation and natural convection. Basic equations for these situations are given in chapter 3 of CIBSE Guide C: *Reference data* (2007). For exposed plane surfaces, typical values are 5 $W \cdot m^{-2} \cdot K^{-1}$ for radiation and 2–3 $W \cdot m^{-2} \cdot K^{-1}$ for natural convection. High surface emissivity is needed to achieve good radiant heat transfer. The degree of geometric exposure of the surface of an element to the space should also be taken into account for radiant heat transfer and is normally calculated using form (or shape) factors (CIBSE, 2007; Holman, 1986).

The high level of radiant cooling provided by an exposed element allows the same level of thermal comfort to be achieved at a higher air temperature.

The airflow within the space for night cooling should ideally be such that the contact between the cool air and the thermal mass is encouraged. Measures such as high-level vents may enhance interaction with exposed soffits.

Improvements in surface heat transfer can be achieved by increasing the surface area by forming coffers or profiling the surface. This can significantly increase the exposed surface area and hence convective heat transfer. Radiant heat transfer benefits will normally be limited if the profiling has a similar overall exposed area to that of a plane surface when viewed from the occupied space. Partial thermal exposure of a slab surface can be achieved by using open-cell or perforated ceiling tiles. This permits air to circulate between the ceiling void and space below for convective surface heat transfer. In addition, open-cell ceilings with a high reflectance may permit a significant amount of radiant heat exchange between the slab above and the space below.

Solid false ceilings will prevent any direct heat exchange between the slab and the space. However, a significant amount of heat exchange may still be possible if the ceiling itself is made of a conductive rather than insulative material.

Surface finishes will insulate the slab from the space below (or above), although thin layers of relatively conductive materials such as plaster shouldn't have a significant effect.

Indirect systems

For indirect systems with air passing through floor voids, cores or air paths, the main surface heat transfer mechanism is convective heat transfer between the air and the store. If convective heat transfer is poor, as is normally the case for airflow in floor voids (typically 2–3 $W \cdot m^{-2} \cdot K^{-1}$) (Barnard, 1994), performance will be limited.

Convective heat transfer coefficients may be increased by using mechanical means to create forced convective heat transfer rather than relying on natural buoyancy forces. High rates of forced convective heat transfer (i.e. 10–15 $W \cdot m^{-2} \cdot K^{-1}$ and upwards) can readily be affected by creating highly turbulent airflow at the surface. This can be achieved by blowing air through hollow cores in slabs or creating air paths through which air can be blown across the surface of a slab. The improvement will ultimately be limited by the thermal characteristics of the concrete.

Forced convection heat transfer coefficients for cores or other air paths may be calculated by using equations given in chapter 3 of CIBSE Guide C: *Reference data* (2007). (It should be noted that these equations are for smooth tubes and therefore represent a worst case, as surface roughness will act to increase turbulence and heat transfer.) Values for the pressure drop for passing the air through the cores or other air paths can be calculated using equations in chapter 4 of CIBSE Guide C: *Reference data* (2007) (which take into account surface roughness).

Thermal admittance ('*Y*-values') can be used to provide a simple measure of thermal performance for different construction types (CIBSE, 2015a). Thermal admittance takes account of both the surface resistance and thermal properties of the element and provides a measure of the dynamic thermal storage performance of an element. This is useful for direct comparison of alternative building element constructions.

Analysis of the performance of thermal mass storage systems should take reasonable account of parameters relating to the storage process, including heat flows in the thermal store and surface heat transfer. Modelling of heat flow in two and three dimensions may also be desirable when analysing geometrically complex building components such as coffered and profiled floor slabs (Jaunzens, 2001).

Night ventilation rates and thermal mass are linked in terms of the cooling provided and should be considered in tandem for design analysis. Increasing night ventilation rates without sufficient thermal mass to store the cooling will be of limited benefit, as will increasing the thermal mass above that required to effectively store the cooling introduced.

(*d*) Conflict with air heating/cooling

For mechanical systems, there is a potential for conflict between heat exchange with the thermal mass and heating/cooling of the air. If air is heated/cooled by a central supply unit and then brought into contact with the thermal mass, heat exchange with the thermal mass may (depending on the relative temperatures) absorb this heating/cooling. Thus the thermal mass will be in conflict with the central supply unit. This could be overcome by providing a bypass to control the thermal link between the supply air and the thermal mass. The bypass could also be used to control when the thermal mass is accessed for storage and discharge. Modulation of the airflow could also be used to vary the rate of storage and discharge.

Where the supply air is cooled, another option is to bring the return air into thermal contact with the thermal mass, e.g. in the ceiling void. The cooled return air provides cooling in the space either by recirculation or by cooling the supply air via a heat recovery device. This avoids any conflict between supply air cooling and the thermal mass. Return temperatures in the ceiling void may be elevated by heat pickup from lights, increasing the cooling effect of the thermal mass.

(*e*) Aesthetics

Exposed soffits should be acceptable aesthetically both in terms of general form and quality of surface finish. Sculpted/profiled/vaulted soffit constructions have been developed to improve the appearance of exposed soffits. Fire protection requirements may also have an impact on the visual appearance. Where there is a desire to conceal part but not all of the slab construction, a partial solution could be adopted. Open ceiling tile solutions could be considered where full concealment is desirable. Although these solutions may not achieve the same level of thermal performance, they may be beneficial in terms of acoustics performance and co-ordination.

(*f*) Acoustics

Exposing a concrete soffit to take advantage of its thermal mass means the absence of a suspended ceiling and hence the loss of acoustic absorbency provided by the ceiling material. This can give rise to increased reverberation time and increased reflected sound across an open-plan space. Counter measures include acoustically absorbent partitions, absorbent banners hung from the ceiling, acoustic plaster, integration of acoustic elements at high level with lighting and profiled slabs to reduce propagation. Sculpted coffers can be designed to focus sound onto acoustic absorbers located in suspended light fittings or back on its source or below carpet level (Bordass et al., 1999)

The effect of acoustic plaster or other finishes on the heat transfer can be significant and should be considered.

For solutions that use partial false ceilings, it may be necessary to adopt measures to avoid flanking transmission between zones. See CIBSE Guide B4 (2016c) for detailed guidance on acoustics and surface finishes.

(*g*) Integration

The absence of a suspended ceiling (and with it the ease of integration of services) can have significant design implications. Where suspended ceilings are provided, modular lighting fittings can easily be integrated. More careful consideration is required where the soffit is exposed to achieve a high level of thermal mass, although it may be possible to integrate the lighting within the coffers. Other options include pendant systems, floor or furniture-mounted uplighters, cornice and slab recessed (CIBSE, 2005). With uplighting, the soffit form is highlighted as an important consideration together with a high surface reflectance of at least 70–80 per cent and a gloss factor of no more than 10 per cent (otherwise lamp images will be visible). Perforations in the light fitting can be used with downlighters to avoid the effect of cavernous coffers.

Routing of conduit and other services should also be considered, as surface mounting may not be acceptable. Solutions include dropping through from the floor above, embedding a conduit network in the slab with access points or routing in hollow cores in slabs.

Partial false ceilings or open ceilings can provide some access to the thermal mass whilst also providing a means of integrating services.

Maximising the use of natural light is important with regard to minimising light energy consumption. Light shelves have been used in a number of buildings to improve the distribution of natural light penetration into a space. The effectiveness of this approach is reliant on reflection from the soffit. As well as a high surface reflectance, the form of the soffit is also important. Plane surfaces are suitable, but 'waffled' surfaces or surfaces with ribs running perpendicular to the flow of natural light will compromise the use of light shelves. Profiling parallel to the flow of natural light can be used to optimise daylight penetration. The design of the soffit should be suitable for integration with possible partitioning layouts.

(*h*) Control strategy

The control of night cooling is important not only in avoiding overheating, but also in avoiding an unreasonable increase in heating demand by cooling unnecessarily (i.e. over-cooling). Inappropriate control strategies can result in significant increases in heating demand (+20 per cent) without appreciable reductions in peak temperatures (Braham et al., 2001).

Strategies are based on a number of criteria including:

— establishing a requirement for cooling (based on zone or slab temperatures)

— cooling availability (i.e. external temperature plus pick-up must be less than the internal temperature)

— avoiding conflict with the heating system (minimum internal set-point)

— scheduled operating periods (to suit occupancy patterns, tariffs)

— disabling heater and heat recovery devices during night cooling

— disabling/enabling mechanical cooling

— avoiding conflict between thermal mass and air heating/cooling

— bypassing/modulation of airflow to control charging and discharging

— damper settings.

Where mechanical cooling is provided, refer to BSRIA TN11/95: *Night cooling control strategies* (Martin and Fletcher, 1995) for detailed guidance on pre-cooling strategies.

(*i*) Construction

The quality of finish required for exposed soffits and the geometrical form will have an influence on whether pre-cast or in-situ construction is to be used for the floor system. One particular issue for pre-cast construction is sealing between units with differential deflection.

The quality of construction of the thermal storage element and surface finishes will have an impact on the thermal storage performance; air gaps under surface finishes can seriously reduce heat transfer. Thermal imaging could be considered as a technique for identifying problem areas with poor heat transfer (Peason and Barnard, 2000) .

For systems where indirect solutions are used, the following should also be considered:

— access to voids, cores and air paths for maintenance purposes

— dust sealing of concrete surfaces within voids cores and air paths.

Slab temperature sensors should be installed at a depth that is representative of the storage capacity of the slab, typically 25–50 mm. Sensors located too close to the surface may be influenced by local effects (e.g. air blowing across the slab, hot plumes rising from equipment). Sensors located too deep into the slab will experience little diurnal swing.

As noted previously, airtightness and conduction losses are particular issues for heating demand in thermally heavyweight buildings. Losses can be stored by the thermal mass resulting in a significant increase in heating demand to overcome this stored cooling. Particular attention should therefore be paid to the sealing and insulation of the building envelope during construction.

3.2.3.5 Ground source systems

3.2.3.5.1 Ground cooling (air)

(*a*) Description

Ground air cooling systems are primarily used for precooling outdoor air in summer. The outdoor air is supplied to the ventilation system via an underground ducting system where the air exchanges heat with the ground (see Figure 3.47). The thermal mass of the ground helps to compensate for seasonal and diurnal temperature variations. The cooling effect in summer is accompanied by an air preheating effect in winter.

The use of ground air cooling is best suited to climates having a large seasonal and diurnal temperature variations. Sensible cooling only of the supply air is provided. The cooling capacity of the system is limited by ground temperatures and by the ratio of the ground coupling area to building area. The system may be used on its own for applications with low levels of gains and where a rise in peak summertime temperatures is permissible. To meet higher cooling loads it may be used in combination with other technologies, in particular those that provide cooling in the space (e.g. cooled ceilings, slab cooling).

In areas of moving ground water, performance may be significantly improved by replenishment of the cooling. However, the presence of ground water involves extensive sealing precautions. The use of ground air cooling is not suited to rocky ground, nor in areas with radon gas.

(*b*) Design

There are a number of key factors that need to be taken into considered during design including:

— size of system

— vertical depth of pipework

— pipework system including header ducts

— location of intake.

The size of the system will be a function of the cooling required and the area available. Smaller systems, e.g. for improving comfort in dwellings, can be built at relatively low cost. In particular, the header ducts can be of a simple design. Systems requiring large header ducts and those immersed in groundwater are considerably more expensive.

Ground temperatures vary as a function of depth and time of year (see Figure 3.48). Pipework should be positioned vertically as deep as possible in the ground without incurring prohibitive excavation costs (i.e. 2–4 m) (Jaunzens, 2001). The system may be located beneath buildings with unheated basements. However, if the basement (or lowest floor) is heated, a significant amount of heat is lost, even if it is well insulated, causing the ground to heat up and performance to drop.

Parameters that need to be considered when designing the pipework system include the following (Jaunzens, 2001).

— *Horizontal spacing of the pipes*: this should be such that the mutual interference between adjacent pipes is not too great (e.g. 1 m).

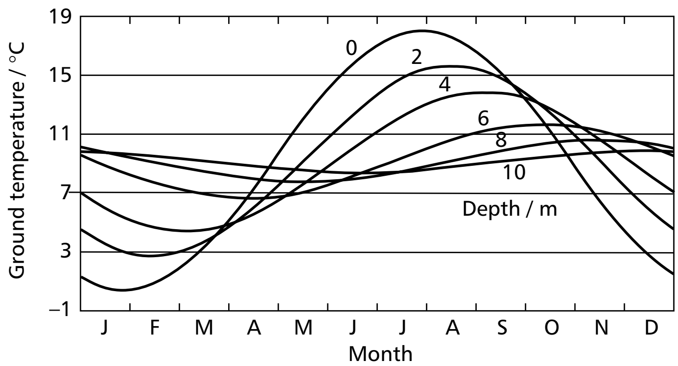


Figure 3.48 Ground temperatures as a function of depth below ground

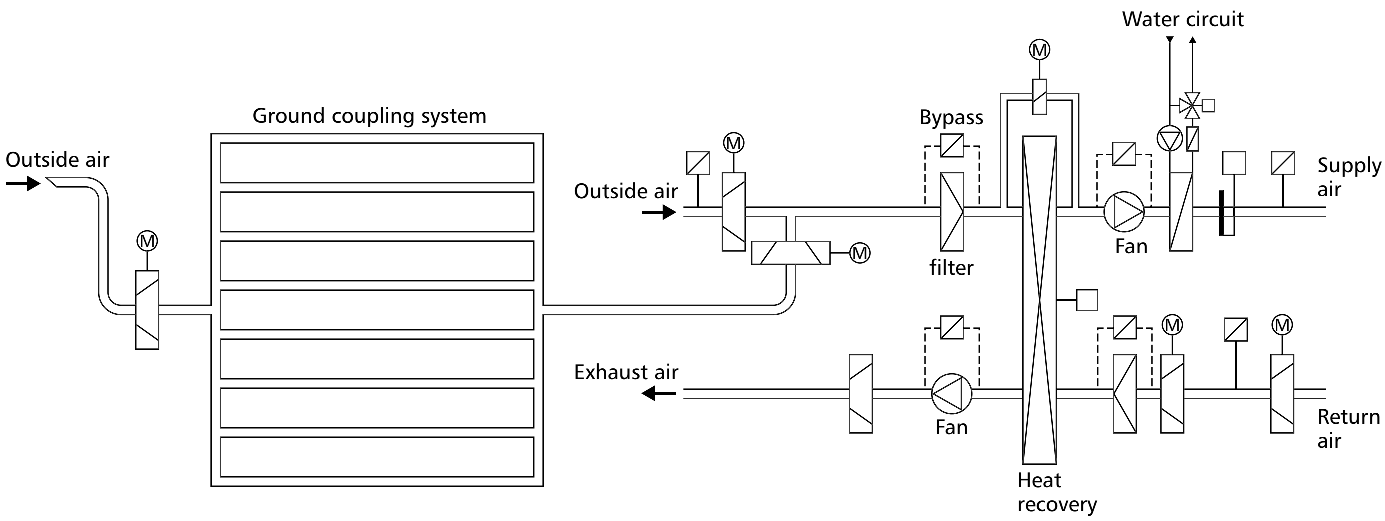


Figure 3.47 Ground air cooling system

— *Design air velocity:* this should be selected to achieve good heat transfer performance without incurring high pressure drops (e.g. 2 m·s^{-1}).

— *Pipe diameter and length*: these should be selected to achieve effective heat exchange (Holman, 1986), typically 80 per cent of the maximum possible (e.g. 200 mm diameter pipes of 20–25 m length, with larger pipes at increased lengths).

— *Soil type*: this has a limited influence on thermal performance (e.g. ±10 per cent), with wet and heavy soils performing better than dry, light soils.

In larger plant, distribution and collection header ducts should be provided. The headers should be adequately sized to ensure that the pressure loss for all flow paths is similar to balance flow rates and for maintenance purposes. For inspection and cleaning, the ducts should be sized to provide crawling access, as a minimum.

The location of the air intake will have an impact on air quality and fouling. Raising the intake above the ground can prevent ingestion of radon gas (which may seep through the ground at any point), reduce the concentration of exhaust fumes from road vehicles and reduce the air intake temperature. To further assist in ensuring low intake temperatures, intake of air should be avoided above parts of the building exposed to strong sunshine or over macadamised surfaces. Fouling can be avoided both by restricting access and by mounting a tight-fitting grille. Fine mesh grilles can be used to prevent entry of birds and rodents etc. Inlet filters should also be considered to limit ingress of dirt and in particular spores to limit possible microbiological contamination (see CIBSE Guide B2, section 2.3.3).

Selection of a suitable operating strategy will depend on the level of load to be met and whether the ground cooling is operating in conjunction with an auxiliary cooling system. Three possible strategies are identified below.

— For low cooling loads, the supply air is passed continuously through the system during occupied periods. Ground regeneration takes place when outdoor temperatures are low.

— For medium cooling loads, the supply air is passed through the system only during occupied periods when cooling is needed to maintain required space conditions, e.g. when the ambient temperature exceeds a preset maximum. Otherwise, the supply air bypasses the system. This will preserve the stored cooling for use during peak conditions. At night, when ambient temperatures are lower, air is passed through the system for ground regeneration.

— When used in conjunction with an auxiliary cooling system to meet higher loads, air is passed though the ground air cooling system continuously. Direct control of the space conditions is achieved by the auxiliary system. The ground air system acts to pre-cool the supply air. Ground regeneration takes place when outdoor temperatures are low.

More detailed design guidance, charts and analysis tools are available for the early design assessment and simulation of ground air cooling systems in BRE IP 6/2000: *Modelling the Performance of Thermal Mass* (Jaunzens, 2001). Thermal design simulation packages that have the facility to model three-dimensional conduction can also be used for assessment purposes.

(*c*) Construction

Ground air cooling system pipes may be plastic, cement or cement fibre. As the location of ground air cooling system pipes makes them very difficult to repair, particular consideration should be given to durability. Thin-walled ribbed pipes or hoses are not recommended. The latter are also more subject to fouling and are very difficult to clean.

Straight pipes are easier to inspect and maintain than curved pipes. Curved pipes should be fitted with a non-corrosive wire with which to draw through cleaning materials. To ensure that condensate and any cleaning water can drain off, ground air cooling system pipes should be inclined at approximately 1 per cent towards the intake (i.e. against the direction of the airflow).

Due to temperature changes, pipes are subject to considerable thermal expansion. The header ducts must be designed to accept thermal expansion. For this, rubber seals may be provided that not only permit axial movement but also protect against groundwater. To prevent long-term lateral movement, the pipes may be cemented in at the centre.

Both distribution and collection header ducts should, as far as possible, be airtight and fitted with drainage and siphon (see Figure 3.49). Drainage will enable condensate, ground water or water remaining from cleaning to escape. This is particularly important for the distribution header duct as it is at a lower level than the collection header duct.

Constructing the header ducts of concrete will add thermal mass to the system. The preheating effect of the distribution header duct in winter will help to protect against icing.

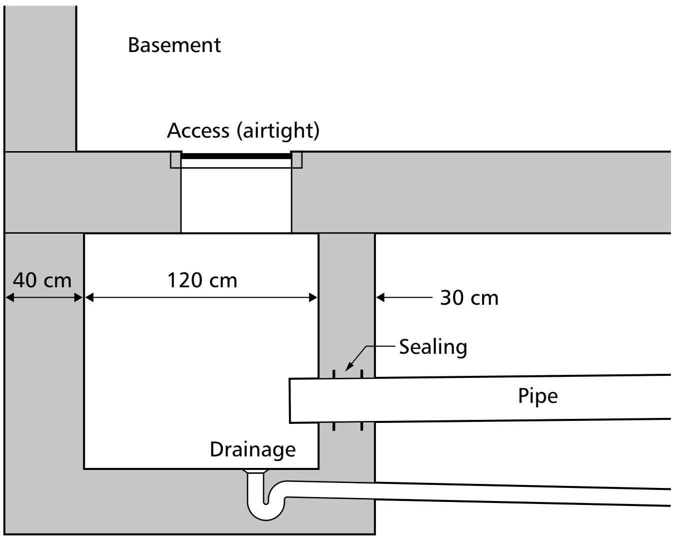


Figure 3.49 Distribution/collection header duct arrangement

3.2.3.5.2 Ground cooling (water)

(*a*) Description (Bunn, 1998; Zimmermann and Andersson, 1999)

In the UK, the annual swing in mean air temperature is around 20 K. The temperature of the ground, however, even at modest depths, is far more stable. At just 2 m below ground level, the swing in temperature can reduce to about 8 K, while at a depth of 50 m the swing is reduced to 0 K.

In addition, at this depth, the ground temperature is approximately equal to the air temperature at that location, that is about 11–13 °C. This stability and ambient temperature makes groundwater a useful source of renewable energy for heating or cooling systems in buildings.

This energy source is usually accessed using a water-to-water heat pump, which provides a means of controlling the temperature of the water delivered to the building and facilitates the most economic sizing of the groundwater collection system. Heating from groundwater almost always requires a heat pump to achieve the necessary delivery temperature. However, useful cooling can be provided by direct connection to the groundwater source. This is known as passive cooling, and is the subject of this section. Ground coupled heating and cooling using heat pumps is discussed above.

Ground water systems are suitable for both retrofit and new-build applications in almost any type of building, including residential. The only proviso is that the geological conditions are suitable and there is sufficient land available on which to install the selected ground water coupling system.

Systems are defined as either open or closed loop. An open-loop system relies on the direct extraction and use of groundwater. A closed-loop system relies on the conductive heat transfer from the surrounding earth or rock into a continuous loop of pipe through which water is circulated.

(*b*) Open-loop systems

These are relatively common and have been incorporated in building designs for many years where there is a readily available supply of accessible natural water. They include not only well systems, but also systems using adjacent lakes, rivers and ponds. The use of sea water has also been recorded. An Environment Agency licence must be obtained for both the abstraction and use of groundwater. The Agency must be assured that no pollutant (other than heat, and even that may be limited as a condition of the licence) will enter the groundwater source. The licence will be for a specific extraction rate.

Although thermally very efficient, open-loop systems tend to suffer from physical blockage from silt and from corrosion due to dissolved salts unless great care is taken in screening, filtering and chemically treating the water. The licence conditions, maintenance and durability issues can significantly increase the overall whole-life operating costs, which has reduced the popularity of open loop systems. Consideration needs to be given to the long-term thermal capacity of the subsurface system, and care must be taken to ensure that the production and re-injection wells are placed suitably far apart to prevent thermal breakthrough occurring in the lifetime of the system. As well as approaching the Environment Agency or equivalent regional bodies, experienced hydrogeologists or thermo-geologists should be consulted particularly for larger systems.

Typical open-loop systems require, following assessment of the geological suitability of the location, a minimum of two vertical boreholes be drilled to a suitable depth to access the aquifer. The system must then be tested to ensure that the water quality is acceptable and that the required and licenced extraction and re-injection rates can be met. Decisions about filtration and materials specification can then be made.

A hydraulic system is then installed, which extracts water, passes it through the primary coils of a heat exchanger and re-injects the water into the aquifer through the re-injection well. Typical groundwater supply temperatures are in the range of 6–10 °C and typical re-injection temperatures are 12–18 °C (although this may be controlled under the extraction licence).

A schematic showing the basic functions of a passive ground water cooling system is shown in Figure 3.50.

Open-loop systems fed by groundwater at 8 °C can typically cool water to 12 °C on the secondary side of the heat exchanger. With a water extraction rate of 25 litre·s^{-1} and a maximum re-injection temperature of 18 °C, this could provide a peak cooling capacity of 900 kW·h. The cooled water on the secondary side of the heat exchanger may be used for a variety purposes as in conventional cooling design, including, for example:

- circulation through an underfloor cooling or chilled ceiling or beam system
- to supply fan coil units.

Underfloor cooling systems may require a higher circulation temperature to minimise the risk of condensation.

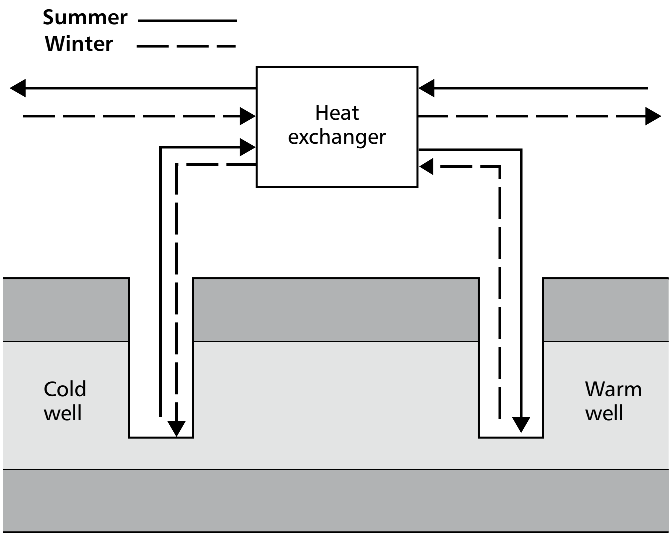


Figure 3.50 Ground water cooling system

The groundwater cooling system in the BRE Environmental Building (see Figure 3.51 below) provides 35 kW of cooling with the borehole temperature picking up 5 K across the primary coils of the heat exchanger. The secondary coils deliver cooled water to underfloor coils which reduce internal temperatures by 2 K at peak loads.

(*c*) Closed-loop systems

These systems are extremely simple, comprising a continuous loop of high-density polyethylene pipe buried in the ground, through which water/glycol solution is circulated. The glycol is circulated by a conventional pump and can be used directly by the cooling distribution system in the building. There are a number of types of closed loop.

- *Vertical boreholes*: these are inserted as U-tubes into small diameter (130 mm) pre-drilled boreholes up to 100–250 m deep. These are backfilled with high-

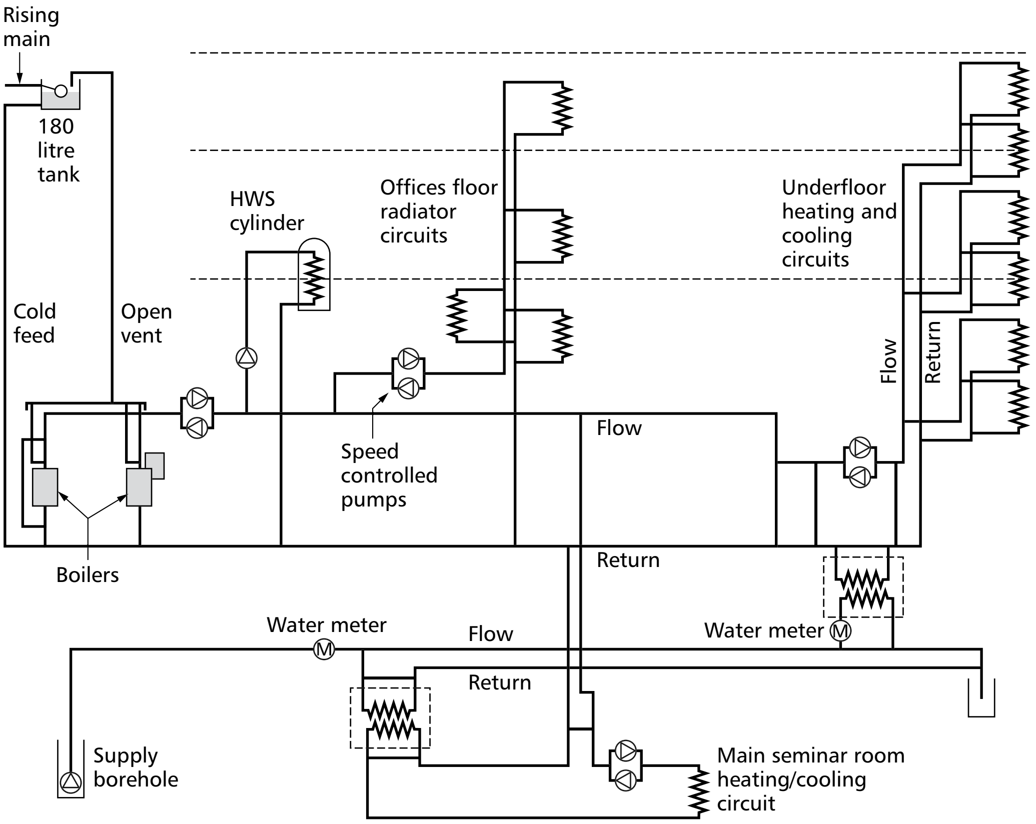


Figure 3.51 Schematic of BRE groundwater system

density grout to seal the bore and prevent cross-contamination of any aquifers the borehole may pass through and to ensure good thermal contact between the pipe wall and the surrounding ground. Vertical boreholes have the highest performance and means of heat rejection, particularly if there is a movement of groundwater across the loop.

— *Thermal piles*: these are U-tubes of polyethylene pipework attached to pile steel reenforcement cages and installed in pile cavity in advance of the pile concrete being poured into place in situ. Thermal piles have comparable performance to vertical boreholes and are an efficient means of heat rejection, particularly if there is a movement of groundwater across the thermal pile.

— *Horizontal loops*: these are laid singly or in pairs in trenches approximately 2 m deep, which are back-filled with fine aggregate. They require a greater plot area than vertical loops but are cheaper to install. The ground temperature is more stable at greater depths, and their performance is affected by how close they are to the surface.

— *Coiled collectors*: with a coiled collector or Slinky™, the length of trench required for a given output will be less than for a trench containing two straight pipes but more collector pipe will be required. They are supplied in the form of a tightly coiled spring, which is released and the resulting looped pipework is either spread horizontally at the bottom of a trench 1 m in width and depth or installed vertically in a 2°m deep narrow (0.25 m) trench. Performance is similar to that of a horizontal loop but may be reduced by pipe overlaps. It is a useful technique for situations where excavation is easy and a large amount of land is available and is a cost effective way of maximising the length of pipe installed and hence the overall energy transfer.

(*d*) Performance

Specific heat transfer rates for vertical indirect collector loops are typically 20–55 $W{\cdot}m^{-1}$ (per metre borehole depth), depending on rock material and water content. Horizontal indirect loops installed in trenches can deliver 10–40 $W{\cdot}m^{-2}$ per square metre of ground area, Depending on the type of ground and duration of heat extraction. Collector design requires specialist software.

More detailed information for heat pump collectors is given in CIBSE TM51: *Ground source heat pumps* (2013b).

(*e*) Critical design factors

Peak cooling loads and the related monthly energy demand profiles will be required before any system sizing can be started. Drilling may present problems if a water-bearing sand layer is encountered and the borehole continually fills with sand. In these circumstances, a cased borehole drilling method will be required, adding to both drilling time and cost. Homogenous rocks such as middle and upper chalk are easy to drill, as are sandstone and limestone. Pebble beds, gravel and clay can be problematic. Site-specific advice may be sought from specialist groundwater cooling consultants. Advice may also be sought from the British Geological Survey. About 50 per cent of the UK landmass is suitable for aquifer-based open-loop technology and virtually 100 per cent is suitable for closed-loop installations.

(*f*) Space requirements

For both open- and closed-loop systems, the main space implications are external to the building and it must be recognised that the ground-loop installation operation itself can occupy a significant part of the total site area. This is often at a time in the normal construction programme when other groundworks are being carried out and site huts etc. are being located.

As far as possible, the horizontal distance between open-loop system boreholes should be at least 100–150 m.

(*g*) Economics

The economic analysis should relate to the area of space served within the building and the relative costs of useful cooling delivered. Passive cooling system installation costs are dominated by the cost of excavation of the boreholes or loop arrays. Operating costs of the circulating costs must be carefully assessed. Sewerage costs will be incurred if it is not possible to discharge water back to the ground.

(*h*) Maintenance

Open-loop systems deliver glycol solution using downhole pumps, which will need periodic inspection and maintenance. Allowance must be made for their withdrawal and reinstallation to/from the borehole. Periodic checks on the chemistry of the circulating fluid, and on any significant increase in reinjection pressures should be made to ensure that there is no significant buildup of deposits in the re-injection well(s). It may be necessary to carry out periodic acid cleaning of the wells to restore flow and re-injection rates.

Closed-loop systems should be periodically inspected to check that there is no loss of ground-loop circulating fluid from the system. A check should be made that the antifreeze concentration is correct and that there is no growth of biological or other contaminants in the system.

3.2.3.6 Legacy systems

3.2.3.6.1 Dual-duct and hot-deck/cold-deck systems

(*a*) Description

Dual-duct systems employ two ducts to circulate separately cooled and heated air to zonal mixing boxes. Zonal temperature sensors ensure that air in the hot and cold ducts are mixed in appropriate proportions to deal with the prevailing load.

Mixing two airstreams to produce an intermediate comfort temperature wastes heating and cooling energy, particularly in constant-volume systems. This may restrict their use to those applications where reclaimed energy can be used. Variable-volume systems offer significantly improved energy efficiency compared with constant-volume systems, although both systems represent a significant energy cost.

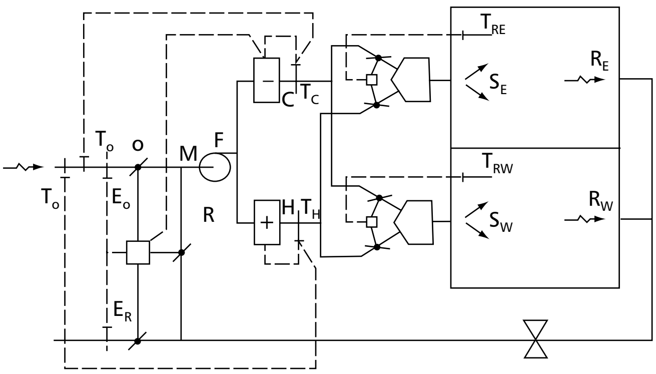


Figure 3.52 Dual-duct system

Dual-duct systems have the ability to deal with heating and cooling loads simultaneously. Room air movement is constant and wet services above ceilings are avoided. However, central plant and distribution systems tend to be larger and more costly than other systems, despite the practice of sizing ductwork for high velocities.

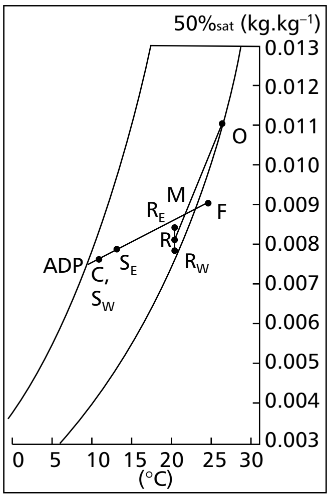
Hot-deck/cold-deck systems are similar in principle to dual-duct systems; the major difference being that zonal mixing occurs at the discharge from the central air handling plant. Hence each zone requires a separate supply from the central plant. This arrangement is best suited to applications involving a small number of zones and where plant can be located centrally. It may also be appropriate for noise-sensitive spaces.

(*b*) Design

Dual-duct constant volume

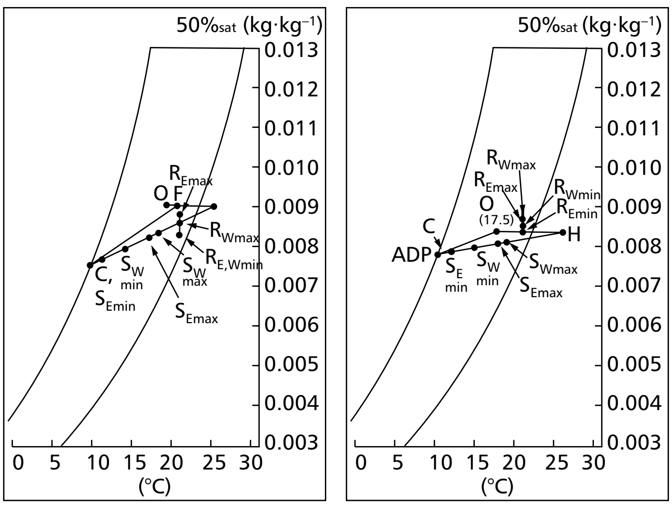
A typical system configuration is shown in Figure 3.52 with the associated psychrometrics in Figures 3.53 and 3.54. Supply temperatures from the AHU should be controlled to provide minimum heating and cooling to satisfy the hottest/coolest zone. Allowance should be made for the reduction in latent cooling due to mixing at part load.

Although the total volume flow handled by the fan remains constant, each duct handles a variable volume. Consequently, the same problems of static pressure fluctuations occur as



t_o = 27 °C

Figure 3.53 Psychrometric chart: dual duct, hot deck/cold deck



(a) t_o = 19 C (b) t_o = 17.5 C

Figure 3.54 Psychrometric chart: dual duct, hot deck/cold deck in VAV systems and require similar remedies at the terminals. Furthermore, with mixing devices operating under part load there is a risk of cross-flow between the two ducts if significant imbalance exists between inlet pressures.

The following methods can be used to maintain system balance:

— Change in duct static pressure resets the set-points of the sensors controlling the hot and cold duct temperatures, hence maintaining constant flow rate in each duct (an unusual solution).

— Static pressure sensors in each duct cause the operation of dampers at the inlet to both hot and cold ducts (suitable for small systems only).

— Employ mixing devices with integral factory-set constant volume regulators (the common solution).

Alternative arrangements and additional features can be employed to deal with specific requirements.

— *Fresh-air pre-heat*: a pre-heater can be incorporated into the fresh-air intake to deal with minimum fresh-air quantities in winter. This avoids the possibility of freezing of the cooling coil due to stratification of fresh and return air through the mixing box and fan.

— *Fresh-air dehumidification*: if the outside air is likely to be very humid at part load, a separate dehumidifying coil can be located in the fresh-air inlet to avoid using very low temperatures at the main cooling coil.

— *Dual-duct reheat*: the cooling coil is located within the central plant so that all the air is cooled and dehumidified, some being reheated in the hot duct, thus providing better humidity control.

— *Dual-duct/dual-fan*: using separate fans for the hot and cold ducts enables the hot duct to handle air recirculated through air-handling luminaires. This assists with winter heating and increases cold duct volume and hence the availability of dry air in summer. Sufficient fresh air must be assured for zones drawing minimum quantity from the cold duct. A bypass between hot and cold ducts will ensure that fans handle constant volumes.

Dual-duct VAV

Alternative arrangements incorporate single or dual supply fans, either with all fans being variable volume or with variable volume cold duct and constant volume hot duct. A cooling coil may also be incorporated into the constant volume system and hence provide the facility to serve some zones with constant volume variable temperature air, some with variable volume cooling and others with a mixture.

The cold duct functions in the same manner as a basic VAV system, providing the facility, at full volume, to deal with maximum cooling load for each zone. The hot duct connection on the mixing box is kept closed until the cooling VAV damper reaches its minimum setting. Any further reduction in cooling loads is dealt with by opening the hot-duct damper. Hot-duct temperature may be programmed against outside air temperature as appropriate.

The cold-duct fan should be regulated under the dictates of a static pressure sensor, in a similar manner to that of a conventional air conditioning system.

Hot-deck/cold-deck systems

As each zone has a separate supply from the central plant, problems of plant imbalance on damper turndown are reduced. Hence, low-velocity distribution is possible, giving reduced fan running costs. However, problems can occur with interaction between separately controlled zones having very different volume flow requirements.

Packaged 'multi-zone' AHUs capable of serving a limited number of zones are available (Figures 3.55 and 3.56) while site-constructed coil/damper arrangements may have as many zonal branches as can be physically incorporated.

Damper quality is an important factor in ensuring satisfactory part-load control and economy of operation. A maximum leakage of 5 per cent when closed should be specified. Precise control action is required in the transmission of the signal from room sensor through control system, actuators and damper linkages.

(*c*) Construction

There are many types of mixing box using various methods of operation. Devices are available both in constant volume form and with sequenced cold duct VAV and mixing (see section 3.2.3 for equipment descriptions).

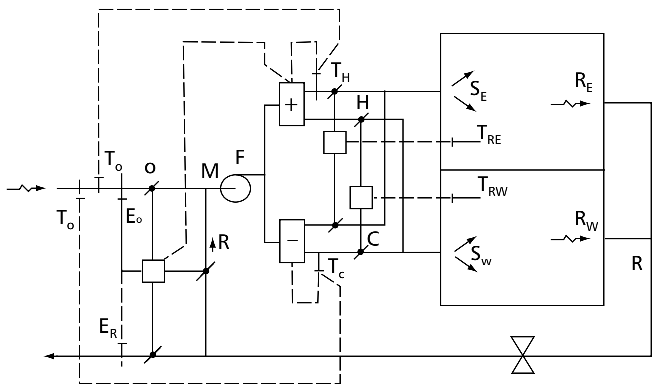


Figure 3.55 Multi-zone hot-deck/cold-deck system

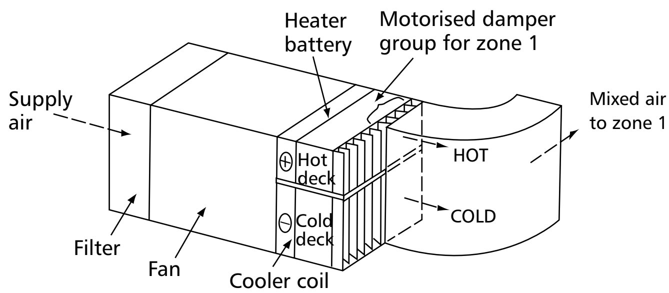


Figure 3.56 Typical packaged multizone arrangement

Basic functions usually performed include:

— mixing air from hot and cold ducts in appropriate proportions to match room load under the dictates of a room air temperature sensor

— mixing air thoroughly to avoid stratification

— attenuating noise generated at mixing dampers

— maintaining constant supply volume against variations in duct pressure.

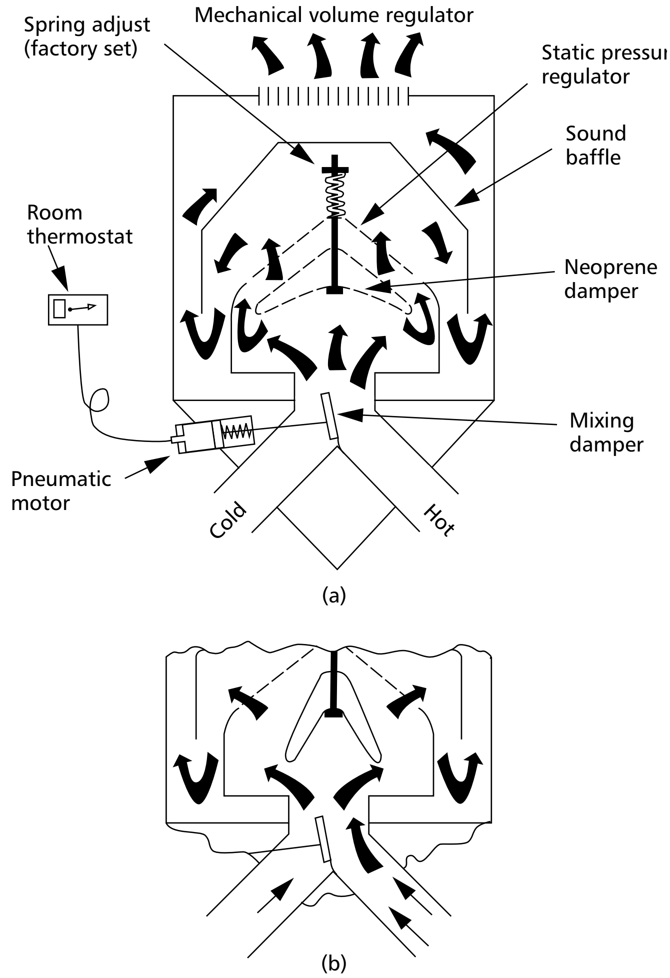


Figure 3.57 Constant volume dual-duct unit with integral static pressure regulator and air terminal devices

Figure 3.57 shows one type of mixing device. Such devices may be individually controlled or several may be slaved from one master device, as with VAV systems. Leakage will always occur through 'closed' dampers. Leakage rates vary from 3–7 per cent of full flow rate for small, well-made devices up to 10–20 per cent for large and site-assembled units. This leakage represents an additional load on the system under peak conditions.

Where mixing devices are provided with integral constant-volume regulators, most types are capable of maintaining a preset volume to within ±5 per cent despite fluctuations of duct static pressure between 250 and 2000 Pa, if necessary. Factory-set volumes need to be checked after installation. The two main types of static pressure regulator are:

— *mechanical*: a spring loaded regulator in the mixed airstream closes as the pressure increases, the mixing dampers operating as a single unit direct from a room sensor

— *pressure actuated*: a room sensor operates the hot-duct damper whilst the cold-duct damper responds to resultant changes in flow sensed by a static pressure differential sensor across an integral resistance.

Stratification can occur if there is inadequate mixing after the terminal and is a particular problem if a multiple outlet mixing device is installed with its outlets stacked vertically.

Noise regeneration at the unit is normally reduced by suitable lining materials and internal baffles. Larger terminals may require separate attenuation.

3.2.3.6.2 Induction units

(*a*) Description

Induction units use the energy in a high-velocity primary air jet to induce room air to flow over a coil and hence promote air circulation within the conditioned space. The benefits provided by induction units include:

— significantly smaller ventilation plant and distribution ductwork than all-air systems

— individual zone control of temperature.

In order to produce the air jet velocity needed to induce airflow, induction systems need to operate at higher pressures than those of low-velocity systems, resulting in fan power and energy penalties.

Induction units are best suited to applications with intermittent medium to high sensible loads but where close humidity control is not required, e.g. offices, hotels, shops and restaurants.

Induction units are normally cased in a vertical configuration for wall mounting, although units designed for overhead installation are available. The vertical units require floor and wall space. Vertical units located under windows or on exterior walls are suitable for buildings with high heating requirements.

(*b*) Design

The various types of induction system are as follows.

— *Two-pipe changeover*: coils are supplied with either chilled or heated water by a common water circuit connected to central heating and cooling plant via three-port changeover valves. This method is appropriate only where the summer/winter transition is easily distinguishable.

— *Two-pipe non-changeover*: coils are supplied with chilled water only via a water circuit (see Figure 3.58). Heating is normally provided by a separate perimeter system or by electric heaters in the induction units. The use of electric re-heaters is not generally recommended for energy efficiency but may be appropriate where heat energy requirements are low (possibly due to high internal gains). Heating the primary air can also be used when heat energy requirements are low, although significant energy wastage can result if zone loads are not similar through induction cooling of heated ventilation air. Primary air temperatures are usually limited to a maximum of 45 °C.

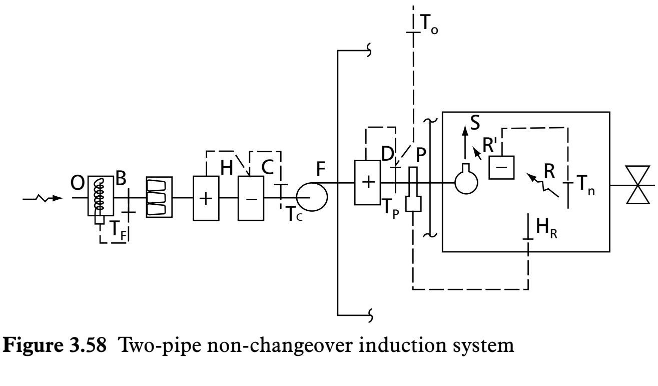


Figure 3.58 Two-pipe non-changeover induction system

— *Four-pipe*: induction units incorporate separate heating and cooling coils, fed by heating and chilled water circuits respectively (CIBSE, 2009). The primary air volume supplied by the central ventilation unit must be adequate to:

- meet fresh-air requirements of the occupants
- provide adequate induction of room air to generate satisfactory air movement
- provide sufficient sensible cooling with induced air without generating unacceptable levels of noise
- deal with the dehumidification load at achievable chilled water temperature
- provide winter humidification, if necessary.

Induction units are rarely used to dehumidify room air, due to the inconvenience of condensate disposal. Therefore, all latent loads must be dealt with by the primary air. Secondary water temperatures must therefore be elevated above the maximum likely dew-point temperature of the room air (see Figure 3.59). The elevated temperatures can improve the efficiency of the central cooling plant and provide more opportunity for 'free cooling' (Figure 3.60).

The central ventilation unit is typically a full fresh-air system with off-coil control of heating and cooling coils, including humidification if required. The ventilation

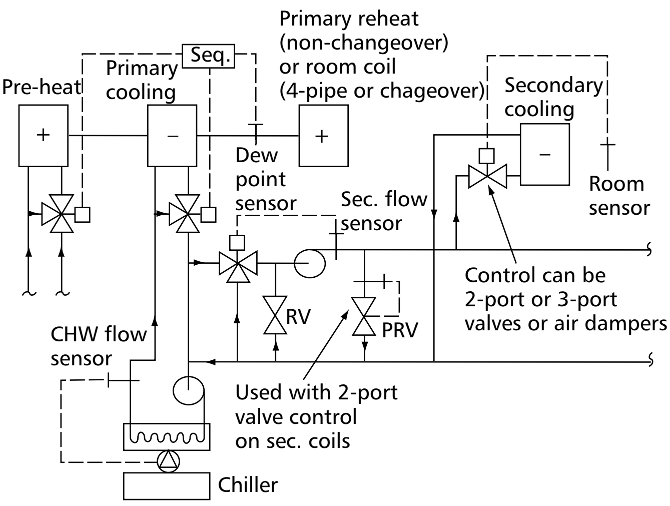


Figure 3.59 Induction system: water control (dry room coils)

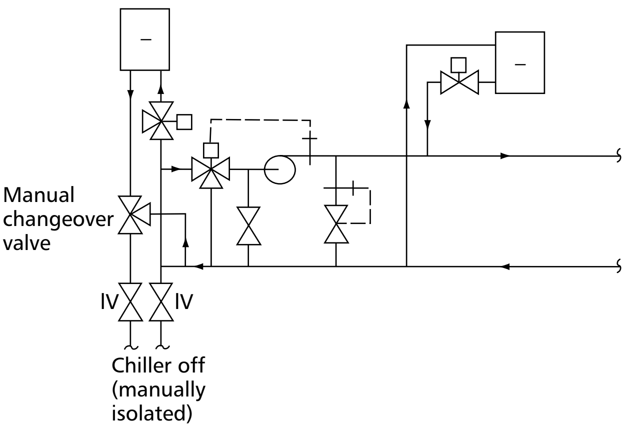


Figure 3.60 Induction system: utilisation of free cooling in primary chilled water

supply air temperature will normally be scheduled against outside air temperature to provide cooling in the summer. Dehumidification should be controlled to minimise the risk of condensation. This may be by limiting the supply air moisture content, or at the dictates of a return-air humidity sensor, or a combination of the two. Ductwork distribution systems are often medium or high pressure.

In winter, the air may be supplied at a neutral temperature or scheduled to provide heating, normally either against outside air temperature or to meet zone requirements. With two-pipe changeover systems, heating may be provided to zones with small cooling loads by increasing the supply air temperature as the outside temperature falls. Changeover to heating can then be delayed until all zones require heating. Humidification may be controlled at the dictates of the supply or return air condition or a combination of the two.

The induction units provide temperature control on a zone-by-zone basis. Induction unit capacity can normally be controlled by coil water flow (waterside) or air bypass (airside). Waterside control can be via four-, three- or two-port coil control valves. Airside control is potentially simpler (one actuator) and will avoid maintenance problems caused by valves blocking but, depending on the configuration, may require slightly larger units and can suffer from problems such as carryover.

Consideration should be given to avoiding conflict between heating and cooling to avoid unnecessary energy waste, particularly where a separate perimeter heating system is provided. One possible approach is to control the heating and cooling in sequence from a common temperature sensor.

Induction units (Figure 3.61) may be used for natural convective heating with the primary plant off. This may assist with early morning pre-heating, which will be costly in terms of energy consumption unless provision is made for recirculation. Access should be provided for maintenance, particularly for cleaning and inspection of the condensate drain pan.

(*c*) Construction

Drain pans should be fitted under each cooling coil to collect moisture from temporary latent loads. Drain pans should be removable for cleaning.

3.2.4 Equipment

3.2.4.1 Introduction

This section sets out critical design issues relating to the specific items of equipment and the key points to be considered in the selection and location of air conditioning equipment.

3.2.4.2 Air intake and discharge points

Each intake and discharge point should be protected from the weather by louvres, cowls or similar devices. Any space behind or under louvres or cowls should be 'tanked' and drained if there is a possibility of penetration by, and accumulation of, rain or snow that could stagnate and give rise to unpleasant odours or contaminants within the

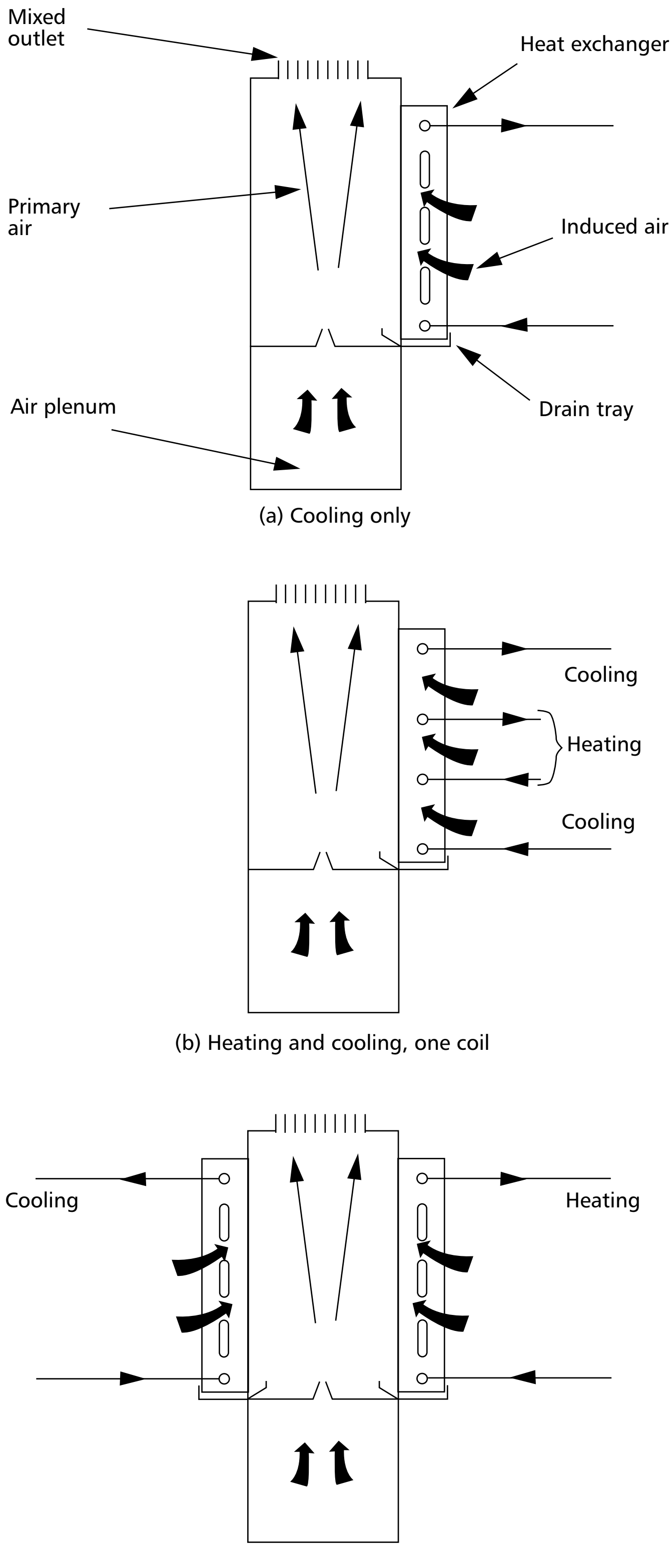


Figure 3.61 Induction units: alternative coil arrangements

building. Bird screens and insect mesh should be used to prevent entry by birds or other large objects.

Intake points should be situated to minimise pollution from a variety of potential sources (existing and planned) including:

— traffic (popularity of diesel engines increases particulate emissions)
— boiler flues and exhausts from standby generators (or combined heat and power engines)
— cooling towers and other heat rejection plant
— vents from plumbing systems, oil storage tanks
— ventilation exhaust outlets from fume cupboards, kitchens, toilets, car parks, print rooms, swimming pools or other sources of emissions
— stagnant water (e.g. on flat roofs)
— roosting ledges for birds (droppings can be a source of biological contamination)
— gardens or areas of vegetation (sources of fungal spores or pollen)
— areas where leaves or other airborne litter might accumulate causing blockages
— radon gas.

Impingement/dust separators should be installed at inlets before filters in dusty/sandy locations (e.g. coastal areas).

Because traffic is generally a ground level pollutant, there is normally a reduction in pollutant concentration with height, so that concentrations are usually lower at roof level. Vehicle loading bays are also likely to be subject to traffic pollution.

Whilst wet cooling towers give rise to the greatest health concern because of the potential risk of *Legionella* bacteria, other heat rejection equipment (e.g. dry coolers or condenser units) can also affect system performance by elevating the temperature of the intake air and increasing the cooling demand on the system.

Locating system discharge and intake points close together facilitates the use of some heat recovery strategies. However, it will also increase the risk of air 'short-circuiting' (i.e. unwanted recirculation). Even extract systems from 'normal' occupied areas will contain pollutants generated by internal sources. These may not represent a health hazard but may still result in an odour nuisance if recirculated. The more remote the intake from the discharge point, the less the risk of air short-circuiting.

Locating the intake and discharge points on different façades can also help to reduce the risk of short-circuiting. However, wind forces on the two fan systems (which will be balanced for openings on the same façade) may affect fan performance and cause flow instabilities, particularly where fan operating pressures are low. The influence of wind pressures can be reduced by:

— positioning openings within a zone of minimal pressure fluctuation
— providing balanced openings that face in two or more opposite directions or an omnidirectional roof-mounted cowl.

Measures that should be considered to minimise re-entry from contaminated sources include:

— discharge exhausts vertically
— group individual exhausts to increase plume rise due to the greater momentum of the combined exhaust airflow rate
— avoid locating exhaust outlets within enclosures or architectural screens that may hold contaminants within the area leading to unwanted recirculation

— locate wall exhausts on the upper third of a façade and intakes on the lower third to take advantage of normal wind separation on a building façade (although consideration should be given to flow recirculation that can occur on a leeward façade and the possible level of traffic pollution)

— avoiding locating inlets and exhausts near edges of walls or roofs due to pressure fluctuations

— placing inlets on roof where wind pressures will not vary greatly with direction to ensure greater stability.

Toxic and hazardous exhaust must not be discharged in a manner that will result in environmental pollution. The local authority environmental health officer should be consulted to ensure that the proposed discharges will be acceptable.

EEC Directive 80/779/EEC (EC, 1980) gives mandatory air quality standards for smoke and sulphur dioxide (see also CIBSE Guide B0 (2016a)). A vertical discharge stack, capable of imparting a high efflux velocity to the exhaust air, may be required. If so, provision must be made for handling any rainwater that enters the stack and avoiding corrosion of the stack materials.

Industrial processes resulting in polluting emissions to air, water or land come under the requirements of the Environmental Protection Act 1990 (HMSO, 1990). Sections 1.6.4 and 1.6.5 of CIBSE Guide A: *Environmental design* (2015a) provide guideline values for pollutants and guidance on air filtration strategies, respectively.

Reference should also be made to CIBSE TM21: *Minimising pollution at air intakes* (1999) for more detailed guidance on pollution sources and assessment methods.

3.2.4.3 Heat recovery devices

3.2.4.3.1 General

This section provides guidance on devices used to recover heat between two separate airstreams. In energy terms alone, recirculation of air is the most efficient form of heat recovery, since it involves little or no energy penalty. However, recirculation is only possible if the return airstream is of sufficient quality for direct reuse and must be managed carefully.

A special case is all-air systems, which generally have much higher ventilation rates, as all the cooling is delivered by the supply air volume. Therefore the ventilation rate is fixed by the cooling load rather than the ventilation (fresh air) needs of the occupants.

Heat recovery devices used in ventilation systems generally provide heat recovery from the exhaust air to supply air in winter, but they can also recover cooling in peak summer conditions. They are also used in specific system configurations such as indirect evaporative cooling (see section 3.2.3.2.2 'Evaporative cooling (direct and indirect)').

Devices used to recover heat from process applications (e.g. dryers, flue stacks, evaporation columns, pasteurisers) may transfer the heat back to the process or to another application. Selection of the heat recovery equipment should be suitable for the process exhaust temperatures and the conditions at which it is expected to operate (pressure, pH). Where the recovered heat is transferred to a ventilation system, modulating control is normally required to prevent overheating in warm weather.

Buildings should be airtight, as infiltration of outside air has a significant impact on the viability of heat recovery (Irving, 1994).

Technical considerations for design and selection of heat recovery devices include:

— heat recovery efficiency (sensible and total)

— airflow arrangement

— face velocity

— fouling (filters should be placed in both supply and exhaust airstreams)

— corrosion (particularly in process applications)

— leakage (cross-contamination of airstreams)

— condensation and freezing

— pressure drop

— construction materials (suitability for temperatures, pressures, and contaminants)

— maintenance (in particular cleaning of surfaces)

— intelligent controls.

The heat recovery efficiency (or effectiveness) of a device is normally defined as follows:

$$\text{Efficiency } (\eta) = \frac{\text{Actual heat transfer}}{\text{Maximum possible heat transfer}}$$

The maximum theoretical efficiency is a function of the exchanger flow configuration; counterflow exchangers have a higher theoretical efficiency than parallel flow exchangers. Practical considerations often favour cross-flow arrangements that lie between the two (ASHRAE, 2012a).

Sensible-heat recovery devices do not transfer moisture (humidity). Latent heat is only transferred when the warmer airstream is cooled below its dew point and condenses. Total-heat recovery devices transfer both sensible heat and moisture between the airstreams. Moisture transfer is desirable in hot, humid climates to reduce the moisture in the supply airstream and conversely in cold, dry climates to increase the moisture in the supply airstream.

Collection pans and drains should be included to collect and dispose of the resulting condensate. In extreme conditions, where the temperature also drops below 0 °C, frosting or icing can occur. This can be prevented by pre-heating the supply air or reducing the effectiveness of the heat exchanger. Alternatively, the heat exchanger may be periodically defrosted. Condensate trays and piping can be treated with self-regulating heating tape to prevent blockages by ice buildup.

Pressure drop across the heat recovery device depends on a number of factors including the exchanger design, airflow rates, working temperatures and connection sizes. These pressure drops should be minimised as they impose a fan energy penalty that will need to be balanced against the value of the recovered energy.

Face velocities are normally limited by the need to avoid excessive pressure drops. Larger devices will have lower pressure drops and higher efficiency but will cost more and require more space. The selection and evaluation of heat recovery devices should include the following parameters:

— capital cost of the heat recovery device and associated equipment (filters etc.)
— capital savings on other plant (e.g. smaller boilers), as recovered heat does not need to be supplied by the heating system
— cost of the energy used to operate the recovery system (e.g. fan, pump, thermal wheel)
— value of the energy recovered by the system (i.e. 100 kW·h of recovered energy typically saves 110–120 kW·h gas at the boilers)
— maintenance requirements
— space requirements of device, filters etc.

Energy analysis may be undertaken using simulation modelling or spreadsheet calculations based on hourly conditions (CIBSE, 2015b), or using graphical approaches such as load-duration curves (CADDET, 1990).

Table 3.12, which is based on information from chapter 9 of ASHRAE Handbook: *HVAC Systems and Equipment* (ASHRAE, 2012c), compares a number of heat recovery devices. These devices are described below. For further information on selection and evaluation of heat recovery devices, See HRS1/2009: *Heat Recovery Systems* (BSRIA, 2009).

3.2.4.3.2 Mixing boxes

A mixing box is a plenum in which recirculated and fresh air are mixed before entering an AHU. It may be part of the ductwork installation, a builder's work chamber or a standard module attached to packaged ventilation plant.

The amount of fresh air and recirculated air entering the mixing box is usually controlled by modulating dampers on both the fresh-air inlet and the extract-air ductwork. The size, stroke and positions of these dampers set the ratio of the two airstreams and hence dictate the resulting mixed air condition. The fresh-air damper must be commissioned to achieve the minimum fresh-air requirement.

Mixing boxes must be designed to provide sufficient mixing under all operating conditions. This is especially the case during winter conditions so freezing outside air does not stratify below warm recirculation air on entering the filters. If in doubt, a frost or pre-heat coil at the air intake should be provided. Dampers should be located and set to promote mixing of the airstreams. Parallel blade dampers may assist with mixing. Air blenders/baffles can also be used to achieve more uniform mixed air conditions.

3.2.4.3.3 Recuperators

Recuperators usually take the form of simple and robust air-to-air plate heat exchangers (see Figure 3.62). Their efficiencies depend on the number and size of the air passages and hence the heat transfer area between the two airstreams. If the passages are large, the heat exchanger may be easily cleaned and will be suitable for heat transfer from particulate-laden exhaust air.

Consideration should be given to the addition of filters if the expected operating conditions will result in rapid fouling or blockage of the heat transfer surfaces.

Modulation control is normally achieved by means of an air bypass. This can be used to reduce fan pressure drops when heat recovery is not required. Little or no air leakage occurs between the airstreams making it suitable for applications where cross-contamination of the airstreams is undesirable. In applications with high differential pressures (> 1000 Pa) exchangers should be selected with robust constructions to avoid plate deformation.

Recuperators normally conduct sensible heat only, but water-permeable materials can be used to transfer moisture.

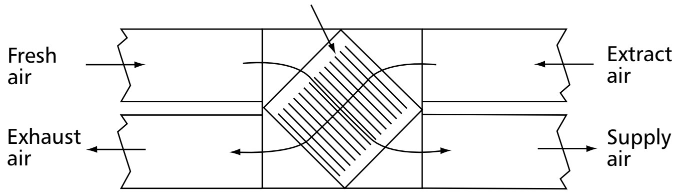


Figure 3.62 Recuperator using a plate heat exchanger

Table 3.12 Comparison of heat recovery devices

Device	Typical heat recovery efficiency / %	Typical face velocity / m·s^{-1}	Cross-leakage / %	Typical pressure drop / Pa	Modulation control	Features
Recuperator	50 to 80 (sensible)	1 to 5	0 to 5	25 to 370	Bypass	No moving parts; easily cleaned
Run-around coil	50 (sensible)	1.5 to 3	0	100 to 500	Pump or bypass valves	Flexibility; exhaust airstream can be separated from supply
Thermal wheel	65 to 90 (total)	2.5 to 5	1 to 10	100 to 170	Wheel speed or bypass	Latent transfer; compact large sizes; cross air contamination possible
Heat pipe	50 to 65 (sensible)	2 to 4	0	100 to 500	Tilt angle down to 10% of maximum	No moving parts except tilt; high cost, few suppliers
Regenerator	85 to 95 (sensible)	1.5 to 3	<1 to 5	70 to 300	Regulating changeover period	Relatively high capital cost, but high efficiency; self-cleaning action from flow reversal

3.2.4.3.4 Run-around coils

Finned air-to-water heat exchangers are installed in the ventilation ducts between which the heat is to be transferred. A water or water/glycol mix (for freeze protection) circuit is used to transfer heat from the warm extract air to the cooler supply airstream (or vice versa in summer) (see Figure 3.63). A pump is required to achieve circulation of the working fluid along with a means of maintaining an acceptable working pressure within the system. An expansion tank is also required to allow for expansion and contraction of the working fluid.

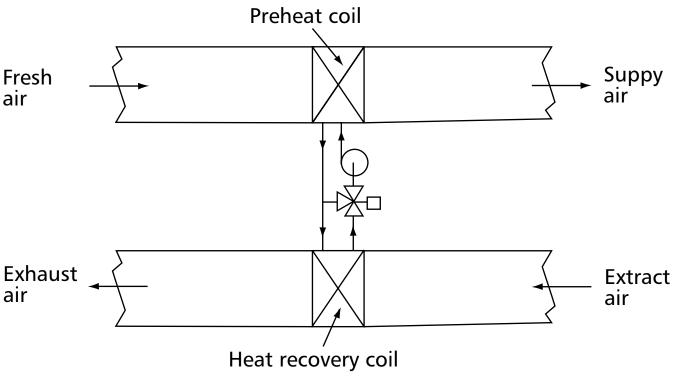


Figure 3.63 Run-around coil arrangement

Overall heat transfer efficiencies are relatively low, as it is a two-stage heat transfer process, and additional pump energy (in addition to the fan energy penalty) and associated maintenance costs need to be taken into account. However, the system uses simple technology and is very flexible in application, as it places no constraints on the relative locations of the two airstreams. The system can also be extended to include multiple energy sources and uses. They are suitable for applications where contaminants in the exhaust airstream prohibit any recirculation.

Modulation control can be achieved by variable pump speed operation and/or valve bypass arrangements on the coils.

3.2.4.3.5 Thermal wheels

A thermal wheel comprises a cylinder, packed with a suitable heat transfer medium that slowly rotates within an airtight casing. The casing bridges the two airstreams between which heat is to be transferred (see Figure 3.64). Thermal wheels are generally quite compact and can achieve high efficiencies when installed in a counter-flow configuration.

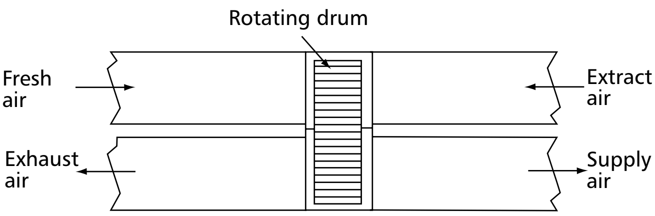


Figure 3.64 Thermal wheel

The heat transfer properties are largely determined by the material contained in the wheel, i.e:

— *corrugated, inorganic, fibrous, hygroscopic material that transfers both sensible and latent heat*: air flows through the channels formed by the corrugations

— *corrugated metal (aluminium, stainless steel or monel)*: latent heat transfer is restricted to that resulting from condensation when the temperature of the heat transfer medium falls below the dew-point temperature of the warm airstream.

Maintenance requirements for thermal wheels need to be taken into account (they can be difficult to clean), as does the additional energy use of the rotational motor drive (although these are usually low-power devices).

Cross-contamination of the airstreams occurs by carryover and leakage. Carryover occurs as air entrained within the wheel is transferred to the other airstream. A purge section can be installed where carryover and cross-contamination is undesirable. Incoming fresh air is used to 'purge' the extract air entrained in the wheel into the exhaust airstream. Increasing the purge rate reduces the capacity of the thermal wheel.

Leakage occurs due to the pressure difference between the two airstreams. This can be minimised by avoiding large pressure differences; providing effective seals and configuring the respective fans in a way that always promotes leakage into the exhaust airstream.

Hygroscopic media may transfer toxic gases or vapours from a contaminated exhaust to a clean air supply, so care is required when selecting the heat transfer media.

Modulation control is commonly achieved either by adjusting the rotational speed of the wheel or by bypassing the supply air. Heat recovery efficiency increases with wheel speed but is ultimately limited by carryover.

3.2.4.3.6 Heat pipes

The heat pipe is a passive heat exchanger of which there are two main types:

— *horizontal*: in which a wick within the tubes transfers liquid by capillary action

— *vertical*: in which heat from the warmer lower duct is transferred to the colder upper duct by means of a phase change in the refrigerant (see Figure 3.65).

Finned tubes are mounted in banks in a similar manner to a cooling coil. Face velocities tend to be low (e.g. 1.5 to 3.0 $m \cdot s^{-1}$) in order to improve efficiency.

Modulation control is normally achieved by changing the slope, or tilt, of the heat pipe.

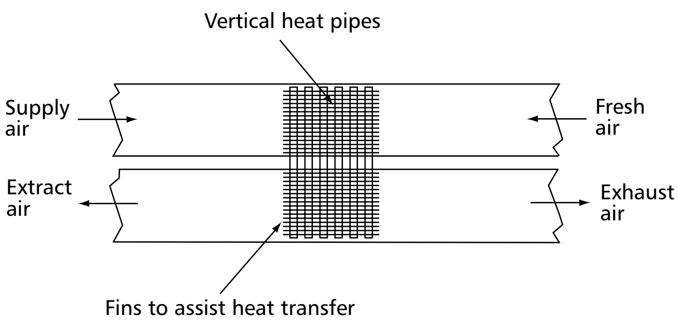


Figure 3.65 Vertical heat pipe arrangement

3.2.4.3.7 Regenerators

A regenerator consists of two accumulators (or a single unit split into two halves) with a damper arrangement to cycle the supply and exhaust airflows between the two (see Figure 3.66).

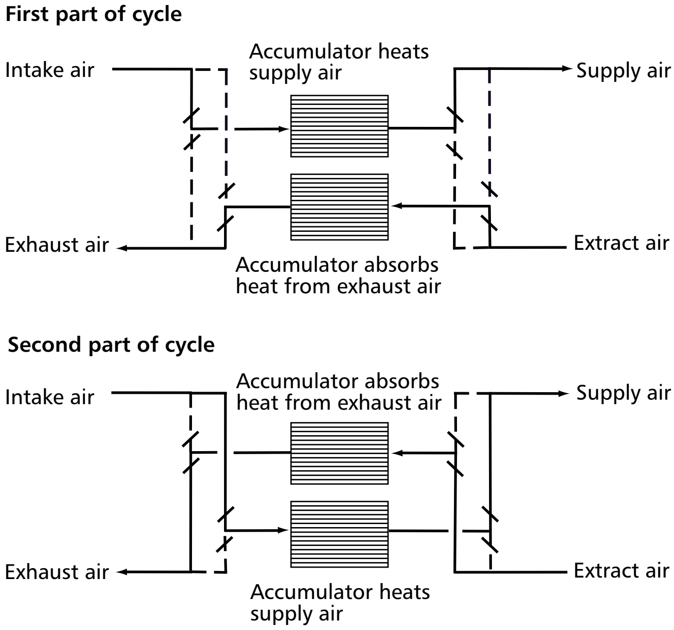


Figure 3.66 Regenerator

In the first part of the cycle, the exhaust air flows through, and heats, one of the accumulators. The dampers then change over so that supply air flows through and absorbs the heat from that accumulator. The second accumulator acts in reverse to match, heating the supply air in the first part of the cycle and absorbing heat from the exhaust air in the second. The changeover period is normally of the order of one minute.

Claimed sensible efficiencies for these systems can be quite high at 85 per cent. Latent efficiencies are normally significantly lower and vary with flow velocity and the accumulator material. Modulation of the heat recovery efficiency can be achieved by regulating the changeover period.

On damper changeover, the exhaust air contained within the damper, accumulator and exhaust ductwork reverses and becomes supply air. The length of exhaust ductwork should be minimised to limit this cross-leakage. Also, the time required for damper changeover should be kept to a minimum by using high torque dampers.

Cross-leakage can range from below 1 per cent on well-designed systems to up to 5 per cent and above. Typical face velocities are 1.5 to 3.0 m·s^{-1}. Reducing the velocity will reduce the pressure drop, but will have only a limited heat transfer benefit as efficiencies are normally high anyway.

3.2.4.3.8 Heat pumps

Heat pumps use the conventional vapour compression cycle to transfer heat from one fluid to another. They may be used in applications where there is a high heat recovery potential but it is not possible to recirculate exhaust air back to the supply, e.g. swimming pools.

See section 3.2.4.7.3 'Direct humidifiers' for further information on heat pumps.

3.2.4.4 Air cleaners and filtration

All fresh-air intake and recirculated airstreams require filtration to remove impurities and other airborne detritus. There are usually specific requirements based on the location and application of the ventilation systems. Refer to CIBSE Guide B2, section 2.3.3 (2016b) for further details.

3.2.4.5 Air heater batteries

3.2.4.5.1 General

A heater battery usually comprises one or more rows of finned tubes, connected to headers and mounted within a steel sheet casing having flanged ends. Tubes in an individual row are usually connected in parallel but sometimes, for water only, may be series-connected as a serpentine coil in a single row. Tubes may be horizontal or vertical except for serpentine coils, which always have horizontal tubes, or steam batteries that always have vertical tubes. Tube rows are usually connected in parallel.

Electric heater batteries consist of resistive heating element(s) regularly spaced across the airflow path. Careful control and interlocks are required to prevent overheating, should the airflow rate reduce or fail.

Gas-fired heating options consist of either direct or indirect burners. They can be used when large quantities of heating are required.

3.2.4.5.2 Materials

Tubes should be of solid drawn copper, expanded into collars formed on the copper or combined with aluminium fins to increase the heat transfer area. Tubes and fins can be manufactured in different materials or pre-coated in a corrosion-resistant finish to suit the application.

Tube wall thickness should not be less than 0.7 mm for LTHW or 0.9 mm for high-temperature hot water (HTHW) or steam. Aluminium fins are usually acceptable, except in corrosive atmospheres, and should not be less than 0.4 mm thick. If copper fins are used they should not be less than 0.3 mm thick.

Because of the variation in applications, specialist advice should be sought on material thickness based on the specific application and tube diameter etc.

Provision should be made to take up movement due to thermal expansion.

Casings and flanges should be of adequate gauge in mild steel, painted with a corrosion-resisting primer. Alternatively, the casings may be in galvanised mild steel with flanges painted in corrosion-resistant primer. Occasionally, both casings and flanges may be galvanised after manufacture.

3.2.4.5.3 Test pressure

Batteries should normally be tested by the manufacturer using compressed air whilst submerged in a water tank and then hydraulically tested on-site in compliance with the Pressure Equipment Directive (EC,1997).

3.2.4.5.4 Heating medium

This is usually LTHW, MTHW (medium-temperature hot water), HTHW or dry steam. Where steam is used for pre-heat coils handling 100 per cent outdoor air, so-called 'non-freeze' heater batteries should be selected. These coils have co-axial steam and condensate tubes that prevent the buildup of condensate, and consequent risk of freezing, in the lower part of the battery.

3.2.4.6 Air cooler batteries

3.2.4.6.1 General

A cooler battery usually consists of one or more rows of horizontal finned tubes connected to headers and mounted within a steel sheet casing having flanged ends. Tubes in individual rows are connected in parallel and rows are usually connected in series, although sometimes they may be interlaced. Piping connections must be made such that the coldest air flows over the coldest battery row, thus approximating contra-flow heat exchange.

Condensate drain trays through the depth of the coil are essential. Intermediate drain trays, condensate sizes and moisture eliminators should be considered by the designer and are dependent on quantity of expected moisture removal and air velocity. Each condensate collection tray must be drained using an appropriately sized connection.

Eliminator plates are necessary if the battery face velocities exceed 2.25 $m \cdot s^{-1}$. Cooler coils should normally be located on the low pressure side of the supply fan to avoid condensate leakage through the casing.

3.2.4.6.2 Materials

Because of the variation in application, specialist advice should be sought on material selection and thickness based on the specific application. Tubes should be of solid drawn copper and expanded into collars formed in aluminium. Tubes and fins can be manufactured in different materials or pre-coated in a corrosion-resistant finish to suit the application.

Alternatively, for more aggressive environments, solid copper tubes should be expanded into collars formed in copper fins, the whole assembly then being electro-tinned. Tube wall thickness should suit the expected working and test pressures, but not be less than 0.7 mm.

Aluminium fins should not be less than 0.4 mm thick and copper fins not less than 0.3 mm thick. Fins should not be more closely spaced than 330 per metre. Facings should be of an adequate gauge of steel, welded or with black mild steel angle flanges, the whole assembly being hot-dipped galvanised after manufacture. A suitable alternative corrosion-resistant construction may be used.

Condensate collection trays should be of not less than 2 mm black mild steel, galvanised after manufacture, and then coated on the inside with bitumenised paint. Suitable alternative corrosion-resistant materials and finishes may be used.

Return bends should be housed within removable covers, allowing sufficient space for the bends to be lagged and vapour-sealed. Alternatively, particularly where a cooler coil is mounted on the high pressure side of a supply fan, return bends should be provided with airtight galvanised steel covers, with adequate provision for condensate drainage back to the main condensate collection tray and drain.

3.2.4.6.3 Sprayed cooler coils

These are generally similar to unsprayed cooler batteries, except that droplet eliminator plates must always be fitted. In addition, a main sump tank is required, which provides a reservoir of water for the spray pump. An array of standpipes and spray nozzles is fitted on the upstream side of the cooler coil which deliver a fine water mist across the coil face. The main sump is made of 3.2 mm black mild steel, galvanised after manufacture and coated internally with bitumenised paint. Cooler coils with aluminium fins must not be used but suitable corrosion resistant materials may be used.

The chemical and microbiological quality of the spray water is paramount to preventing health risks and scaling of the heat transfer surface.

3.2.4.6.4 Test pressure

Batteries should normally be tested by the manufacturer using compressed air whilst submerged in a water tank and then hydraulically tested on-site in compliance with the Pressure Equipment Directive (EC,1997).

3.2.4.6.5 Cooling medium

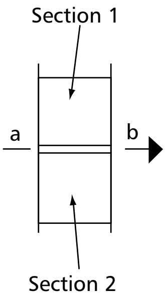
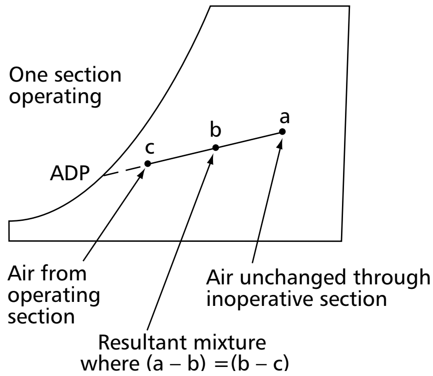
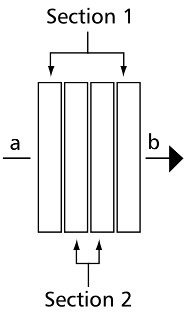
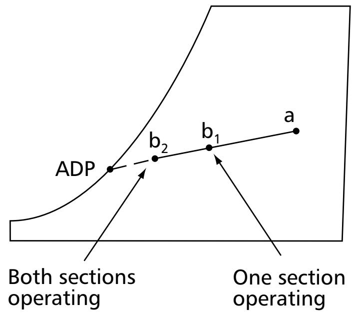
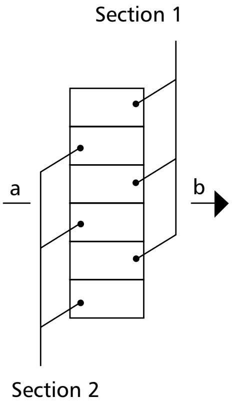
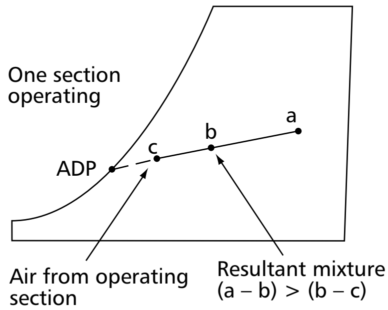
The cooling medium is usually chilled water or occasionally working fluids such as glycol solution or brine are used. Where heat transfer fluids other than water are used, the reaction of the heat transfer fluids with the piping and pumping materials must be considered. All cooling systems require suitable steps to be taken to prevent corrosion and maintain the quality of the working fluid.

3.2.4.6.6 Refrigerant cooling coils

When the cooling coil is a refrigerant evaporator, additional care must be taken with its design, material selection and control because of its interaction with a refrigeration system. A normal vapour-compression refrigeration system using an oil-miscible refrigerant and thermostatic expansion valve has a limited control range.

For a wide control range, it is usually necessary to divide the cooling coil into two or more discrete sections, each with its own thermostatic expansion valve, isolating inlet solenoid valve and, sometimes, its own suction line. By this means, as each section is isolated, the control range of the whole coil is increased as far as the limit of operation of the remaining duty sections.

Table 3.13 Psychrometric effect of coil selection arrangement

Schematic	Remarks	Psychrometric effect
	Effect is that of complete bypassing of half the coil	
	Effect is to reduce contact factor of coil as though it consists of fewer rows	
	Depending on coil construction and number of sections, part of finned surface in an operative section is partially effective in transferring heat to an operative section	

For sectional control, the psychrometric effect of the coil section arrangement must be appreciated, as shown in Table 3.13 above. It is also common practice to connect a separate compressor or condensing unit to each section or to pairs of sections on a multi-section coil in order to increase the overall control range.

When this is done, it is advisable to connect the compressor and coil sections such that each section performs an equal share of the required duty. This avoids a tendency towards frosting due to unequal evaporating temperatures.

Other options include the use of electronic expansion valves and evaporating pressure control by loading and unloading compressors.

Refrigerant cooling coil pressure testing is normally carried out with oxygen-free nitrogen (OFN). Test pressures vary depending on refrigerant and application (refer to BS EN 378-2 (BSI, 2008b) for further guidance.)

3.2.4.7 Humidifiers

3.2.4.7.1 Requirements for humidity control

The demand and requirements for humidity control is considered in CIBSE Guides A (2015a) and H (2009). Reference should be made to BSRIA AG10/94.1: *Efficient humidification in buildings* (Bennett, 1995) and CIBSE KS19: *Humidification* (CIBSE, 2012c).

3.2.4.7.2 System classifications

The variety of equipment types likely to be encountered is shown in Tables 3.14 and 3.15 below. However, the trend is now towards steam and ultrasonic systems due to the potential risks to health resulting from poor water quality. Some older system types are included in the tables because they may be found in existing buildings. The tables distinguish between direct and indirect humidifiers, as follows:

Table 3.15 Storage humidifiers; indirect

Aspect	Spray washers	Capillary washers	Sprayed coils	Pan
Application	Commercial/ industrial	Commercial/ industrial	Commercial/ industrial	Commercial/ industrial
Separation efficiency	70–90%	97%	Up to 95%	Low
Thermal efficiency	Up to 80%	Up to 80%	Up to 95%	Low
Filtration	Low under 20 μm particle size	90% by weight down to 3 μm particle size	Low	Low
Basis of operation	Pump	Pump	Pump	Static water
Saturating method	Fine spray	Surface film	Surface film	Surface film
Use	Humidifying/ dehumidifying	Humidifying/ dehumidifying	Humidifying/ dehumidifying	Humidifying
Advantages	Variable saturation by water control	High efficiency, high filtration, minimum space	High efficiency	Low initial cost

Table 3.14 Non-storage humidifiers; direct and indirect

Aspect	Direct or indirect			Indirect	
	Mechanical disc	Mechanical pressure	Vapour injection	Compressed air	Hydraulic separators
Application	Commercial/ industrial	Commercial/ industrial	Commercial/ industrial	Industrial	Industrial
Separation efficiency	90%	90%	Up to 80%	Variable	Variable
Thermal efficiency	Low	Low	Restricted (humidifying only)	Low	Low
Filtration	Nil	Nil	Nil	Nil	Nil
Basis of operation	Revolving disc	Fan/pump	Steam	Air jet	Water jet
Saturating method	Fine spray	Fine spray	Vapour	Fine spray	Fine spray
Use	Humidifying	Humidifying	Humidifying	Humidifying	Humidifying
Advantages	Fineness of mist	Fineness of mist	Low maintenance cost	Low initial cost	Low initial cost

— *Direct humidifiers*: these have a particular application in industrial fields. They discharge water particles or vapour directly into the space to be treated. The air in the space absorbs the moisture to a degree consistent with the air movement or turbulence and the fineness of the water particles created by the apparatus.

— *Indirect humidifiers*: the addition of moisture to the air takes place within the apparatus itself, the air leaving in a near saturated state. Moisture is presented to the air as a mist or surface film, depending upon the type of apparatus.

If humidification takes place without the addition or removal of heat (i.e. adiabatic), the process relies on evaporation. Sensible heat is taken from the air to provide the latent heat of evaporation necessary to convert the liquid water into a vapour of the same temperature. In doing so the temperature of the air is reduced, although the total heat of the system remains constant. There are three basic types of adiabatic humidifier (Bennett, 1995):

— *Air washers (and evaporative coolers, see below)*: these are usually found in large, central air conditioning systems

— *Wetted media*: these are used in residential and small commercial buildings (not in the UK)

— *Water atomising*: these cover a wide range of applications as a result of their large capacity range.

Through an efficient humidifier, air can be cooled almost to its entering wet bulb temperature and it can then effectively remove sensible heat gains from the building. In practice, internal temperatures may be maintained at or near the external dry bulb temperature.

Evaporative cooling systems are discussed further in section 3.2.3.2.3 'Evaporative cooling (direct and indirect)'.

Isothermal humidification means that the process occurs at a constant temperature. As such, there is no cooling or heating applied to the ventilation process. Strictly, there is a very slight heat input and temperature rise as the water vapour introduced is often at, or near, steam temperature. Isothermal humidifiers can be divided into two categories (Bennett, 1995):

— *steam humidifiers*: including those where the steam is produced remotely and conveyed to the ventilation system

— *vapour generators*: where heat energy is converted to water vapour within the apparatus itself.

3.2.4.7.3 Direct humidifiers

(*a*) Hydraulic separation

Water separators operate direct from the high-pressure mains supply, the water jet impinging on a cylindrical or volute casing, suitable ports liberating the water as spray.

(*b*) Compressed air separation

Where compressed air is available, high-pressure jets can be utilised to produce a fine water spray. Air-atomising systems have larger water openings in the nozzles than water separators and hence are less susceptible to fouling from water impurities.

(*c*) **Mechanical separation**

Mechanical separators operate at constant water pressure. They are often of the spinning-disc type in which water flows as a film over the surface of a rapidly revolving disc until thrown off by centrifugal force onto a toothed ring where it is divided into fine particles. Alternatively, water is injected into a scroll-shaped housing and separated by the action of either a fan or a pump. Some mechanical separators produce fine droplets that are lighter than air, termed aerosols, which are non-wetting.

(*d*) **Vapour injection**

For pre-heating in drying rooms and other applications, direct injection of steam can provide a simple and effective method of increasing the moisture content of air, provided that the rise in wet bulb temperature from the heat in the steam does not cause control problems.

3.2.4.7.4 Indirect humidifiers

(*a*) **Spray washers**

The efficiency of spray washers is governed by:

— the fineness of atomisation achieved by the spray

— the quantity of water sprayed into the chamber in relation to the air capacity

— the length of the unit and, consequently, the time for which the air is in contact with the water mist.

To obtain maximum efficiency, the face velocity is limited to 2.5 $m \cdot s^{-1}$. Efficiencies to be expected are 70 per cent for a single-bank spray washer, 85 per cent for a double-bank spray washer and 95 per cent for a triple-bank spray washer, with a distance between each bank of about 1 m.

Suggested rates of water flow are approximately 5 $litre \cdot s^{-1}$ of water per 3 m^2 of face area of the spray chamber per bank of sprays, which is equivalent to approximately 7 $litre \cdot s^{-1}$ of water per 10 m^3 of air per second. To provide the fine degree of atomisation required, gauge pressures in the region of 200 kPa are required at the spray nozzles.

This type of humidifier is particularly prone to bacteriological growth and other forms of contamination since water storage ponds may remain still for long periods during warm weather.

(*b*) **Capillary type washers**

In principle, capillary-type washers are built up from unit cells, each cell packed with filaments of glass specially orientated to give the minimum resistance to airflow with the highest efficiency.

The cells are sprayed from nozzles at a gauge pressure of 40 kPa, producing coarse droplets of water which, by capillary action, produce a constant film of moisture over each glass filament. The air passing through the cell is broken up into finely divided airstreams providing maximum contact between water and air, resulting in high efficiency of saturation. Most dust particles down to 3 μm in size are also eliminated from the airstream, and it is therefore necessary to provide a constant flush of water through the cells to eliminate the danger of blockage.

Alternatively, an intermittent supply, controlled by time clock, may be used to flush the cells with water at predetermined intervals. The face velocity through the washer chamber is similar to the spray type, i.e. 2.5 $m \cdot s^{-1}$ with a maximum of 2 $m \cdot s^{-1}$ through the cells.

Saturation efficiency of 97 per cent can be achieved with as little as 0.8 litres of water per 10 m^3 of air per second, although a minimum of 4.5 litres per 10 m^3 of air per second is required for flushing purposes. The cells have a maximum water capacity of 11 litres per 10 m^3 of air per second.

Capillary cells are arranged in parallel-flow formation, where the air and water pass through the cell in the same direction, or in a contra-flow arrangement with water and air passing through the cell in opposite directions. Selection is governed by the humidifying or dehumidifying duty required from each cell and also the degree of cleanliness of the air handled.

Prevention of bacteriological and other contamination must also be considered.

(*c*) **Sprayed coils**

Coils fitted into casings and sprayed from low-pressure nozzles provide an efficient means of humidification. The efficiencies obtained are in direct relation to the contact factor of the coil and thus depend on the number of rows provided, the spacing of the fins etc.

The recommended rate of spray is about 0.8 $litre \cdot s^{-1}$ per m^2 of face area with a gauge pressure at the spray nozzles of 50 kPa.

Precautions must be taken to prevent bacteriological and other contamination. Ideally, water circulation should be continuous.

(*d*) **Pan humidifiers**

The simplest form of indirect humidifier is the pan type that consists of a shallow tank in which the water is kept at a constant level by a ball-float valve.

The air passing over the surface of the water picks up moisture and the water may be warmed to increase effectiveness. Efficiencies are low and depend upon the area of water surface presented to a given volume of air. Disadvantages arise from the odours that can result from the static water surface.

Use of this type of humidifier is discouraged because of the high risk of bacteriological contamination.

(*e*) **Mechanical separators**

Mechanical separators of the revolving-disc type can, in addition to their usefulness as direct humidifiers, be mounted into a chamber similar to a spray washer, taking the place of the spray system and pumping set. Water treatment should be considered in hard-water localities, as any free aerosols not absorbed in the plant may be carried through into the conditioned space, evaporating and precipitating salts on surfaces in the form of a white dust.

(*f*) Steam humidifiers

Steam provides a relatively simple and hygienic method of humidification, providing that the heat in the system can be absorbed. For application to air conditioning systems steam can be generated at low or atmospheric pressure from mains steam, an electrode boiler or electrical resistance boiler.

(*g*) Ultrasonic humidifiers

Ultrasonic humidifiers rely on the principle of ultrasonic nebulisation brought about by a rapidly oscillating crystal submerged in water. The crystal, a piezo-electric transducer, converts an electrical signal into a mechanical oscillation. This forms a cavity between the crystal and the water, creating a partial vacuum. At this precise instant, the water is able to boil, creating a low-temperature gas. This is then followed by a positive oscillation, creating a high-pressure wave that is able to expel the pocket of gas through to the surface of the water. Condensation occurs, but the net result is the release of finely atomised water that is readily able to evaporate.

(*h*) Rotating-drum humidifiers

Rotating-drum humidifiers consist of a cylinder or belt that is partially submerged in a water trough. The drum or belt rotates to continuously wet the surfaces. The rate of evaporation is usually controlled by stopping or starting the rotation of the drum. Some humidifiers incorporate a fan.

(*i*) Infrared evaporators

Infrared lamps evaporate water contained in reservoirs or pans. Parabolic reflectors are used to reflect and focus the infrared radiation downward onto the water. Units can be duct mounted or, if equipped with an integral fan, can be positioned in the room to be humidified.

3.2.4.7.5 Excess moisture elimination

Indirect water-type humidifiers normally induce more moisture than that required to saturate the air. To prevent excess moisture entering the ducting system, an eliminator section is generally incorporated in the humidifier. This comprises either a series of vertical plates profiled to cause directional changes of the air or, alternatively, mats of interlaced plastic or metal fibres retained in suitable frames. Depending upon the depth of the coil, an eliminator section is not required with sprayed coil coolers if the face velocity is below 2.25 $m \cdot s^{-1}$.

3.2.4.7.6 Humidifier positioning

Research (Bennett, 1995) shows that humidifiers are often placed where space permits and hence are not necessarily in the location that best suits control or humidification requirements. The preferred position for the humidifier is downstream of the supply fan and clear (i.e. downstream) of any turning vanes or dampers but sufficiently upstream of the space for complete absorption to have occurred. A rapid absorption design (i.e. one that creates greater dispersal across the cross-sectional area of the duct) may be required to avoid the formation of condensation or water droplet impingement if there are nearby obstructions within the duct. An alternative location should be sought if this is not possible. The next best choice is just upstream of the fan, provided that the water has been suitably absorbed by the air. This is to avoid fan failure due to the fan being wetted.

3.2.4.7.7 Materials

Pollution in the air handled, and the nature of water used for humidification purposes, can create chemical conditions that may require the use of protective coatings, plastic materials or other metals in preference to steel. However, some materials provide suitable conditions for growth of bacteria and these should be avoided. A list of such materials is given elsewhere (Bennett, 1995).

3.2.4.7.8 Water supply

(*a*) Scale formation

Treatment of water may be necessary where, for example, available water supplies contain a high degree of temporary hardness or calcium salts in free suspension. The local area water authority should be contacted to identify the water quality and the manufacturer's or supplier's advice subsequently sought on water-treatment requirements. Any precipitation that does take place can be dealt with by the use of special inhibitors or dispersant treatments.

(*b*) Health hazards arising from humidification

Expert advice must be sought to ensure that all humidification systems are safe in their design, operation, and subsequent upkeep. Designers are advised to be aware of the latest guidance, in particular that produced by the Water Regulations Advisory Scheme, Health and Safety Executive, CIBSE Guide G: *Public health engineering* (2014) and CIBSE TM13: *Minimising the risk of Legionnaires' disease* (2013a). HSC Approved Code of Practice and guidance L8: *Legionnaires' disease: The control of Legionella bacteria in water systems* (HSE, 2013) applies to any humidifier or air washer where a spray of water droplets is produced and the water temperature is likely to exceed 20 °C, as infection is caused by inhaling airborne droplets and the formation of legionella bacteria is promoted within water temperatures in the range of 20–45 °C.

To avoid risks, it is suggested that designers specify equipment that does not create a spray, i.e. steam or evaporative humidifiers (HSE, 2013). Where humidifiers that create a spray are used, the risk should be controlled by ensuring that the equipment, and the water supplied to it, is kept clean. This involves regular cleaning and disinfection, continuous water circulation and the drainage of tanks and headers when not in use. Note that water treatment chemicals are not recommended for use in humidifiers and air washers when buildings are occupied. Using water direct from the mains supply, rather than recirculated or stored water, will reduce microbiological contamination. However, local authority approval will be needed under The Water Supply (Water Fittings) Regulations 1999 (TSO, 1999d).

3.2.4.7.9 Energy use

Steam-injection systems, whether drawn from a steam distribution system or from local electric boilers, give rise to very significant energy consumption, increasing with the closeness of control required, see Figure 3.67 (CIBSE, 2012b). Where steam is available on-site, it is sensible to make use of it and directly inject steam to provide humidification. Local electrically powered humidification units generally have independent controls and can cause significant increases in electricity consumption. Ultrasonic systems are becoming more popular and use up to 90 per cent less energy than electrode boiler systems. However, electricity is required for the water purification plant, and reheating may be needed if the air temperature is reduced by the evaporative cooling effect.

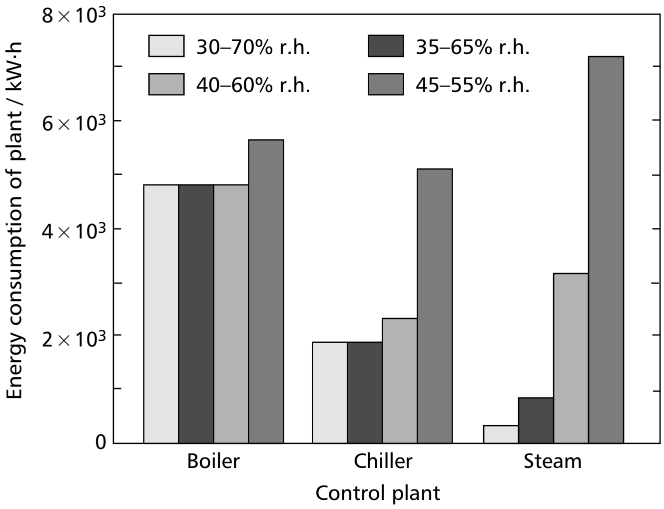


Figure 3.67 Effect of humidity control on efficiency

3.2.4.8 Fans

Fans consume a large proportion of the total energy used in mechanically ventilated and air conditioned buildings. A high priority should therefore be given to achieving and maintaining energy efficiency of the system to reduce pressure drops, e.g. the AHU, as this is normally responsible for the majority of the losses. Equipment selection should favour the most efficient fan types and try to ensure that the fans will be operating at their peak efficiency.

Volume control should always be incorporated to meet varying levels of ventilation demand or to achieve the exact design volume. This may be at the dictates of temperature, pressure or air-quality sensors. Volume control may be achieved by a number of means including:

— on/off, multispeed or variable-speed fan operation

— varying the blade pitch for axial fans.

Main dampers should not be used, as they impose an additional resistance on the fan so incur a further energy penalty.

The various fan types and components and fan performance are discussed in detail in CIBSE Guide B2 (2016b).

3.2.4.9 Airflow control units

3.2.4.9.1 General

When multiple areas that require air conditioning have differing heat gain patterns with respect to time, these can be serviced from a central plant in which either the temperature or volume (or both) of the air supplied to each area is varied to meet the particular cooling (or heating) requirements of the area. Such temperature or volume control may be carried out in the common ductwork serving a number of rooms or zones, or it may be carried out in the terminal units supplying conditioned air into individual rooms.

3.2.4.9.2 Control of volume

Volume control may be achieved in the following ways:

— *Volume-control damper*: normally of the butterfly or multi-leaf type and capable of controlling the volume, provided the pressure drop across the damper does not exceed about 40 Pa. If the pressure drop is higher, there will be a tendency to generate excessive noise. Normally the damper is supplied as a separate component for direct installation in the ductwork and not as part of a terminal unit. Final adjustment is carried out manually on-site. The damper should be ‘locked’ and marked when the correct position is established.

— *Pressure-regulating valve*: an assembly consisting of one or two rows of shaped blades, the size of which changes when volume adjustment is required. Because of the particular blade shape, the device gives volume adjustment up to pressure drops of about 630 Pa without generating excessive noise. The majority of dampers are set on-site, but they can be controlled from a static pressure sensing element. Such units are generally supplied as a separate component for direct installation in the ductwork and not as part of a terminal unit.

— *Mechanical volume controller*: a device that is self-actuating and capable of automatically maintaining a constant preset volume through it, provided that the pressure drop across it is within the required working range (above a minimum of about 120 Pa and below a maximum of about 250 Pa). As the supply air pressure increases, most devices of this type tend to close progressively by means of a flexible curtain or solid damper; a multi-orifice plate fixed across the complete airway of the unit. As such, a unit achieves volume reduction by reducing the area of the airflow path. There is a tendency to generate noise, particularly when working at high air pressures. For this reason, the volume controller is generally supplied in an acoustically treated terminal unit.

  These terminal units are available in different sizes, with each size capable of airflow regulation in a specified working range. Each unit can be factory set to deliver a specific air volume or, more often, is manually adjusted on-site to deliver the required airflow rate.

3.2.4.9.3 Control of temperature

This may be achieved by:

— *blending*: two separate airstreams, one warm and one cool, are supplied to a zone and mixed in a terminal unit to produce a supply air temperature that offsets the zone-cooling or heating loads

— *reheat*: controlled reheat of a pre-conditioned, low temperature air supply by means of hot water, steam or electric coils may be used to increase the local supply air temperature, which will satisfy the zone(s) requirement.

Table 3.16 Types of air terminal device

Type		Application	Location	Core velocity / $m \cdot s^{-1}$		Description and remarks
				Quiet	Commercially quiet	
1	Perforated or stamped lattice	Supply, extract, transfer	Ceiling, sidewall floor	Up to 4	Up to 6	Simple form of grille with small free area. Alternatively can be used as supply diffuser with high air entrainment allowing large quantities to be diffused. For low-level 'laminar flow' panels to give displacement ventilation, a velocity of 0.25 $m \cdot s^{-1}$ is used.
2	Aerofoil blades, one row adjustable	Supply, extract	Ceiling, sidewall desk top	7	10	Frequently used grille with large free area. Directional control in one plane only for supply applications.
3	Aerofoil blades, two rows adjustable	Supply	Sidewall	7	10	As type 2 but with directional control in two planes.
4	Fixed blade	Supply, extract		6	9	Robust grille with limited free area. Some directional control possible using profiled blades.
5	Non-vision	Extract	Sidewall transfer	7	10	Low free area. Designed to prevent through vision.
6	'Egg crate'	Extract	Ceiling, sidewall	7	10	Generally largest free area grille available.
7	Fixed geometry diffusers	Supply, extract	Ceiling, floor, desk top	7	10	Radial discharge diffusers offer good air entrainment allowing diffusion of large air quantities. Square or rectangular diffusers can provide one-, two- or three-way diffusion. Angled blades can be used to apply twisting motion to supply air.
8	Adjustable diffusers	Supply	Ceiling	4	6	As type 7 but offers horizontal or vertical discharge. Can be thermostatically controlled.
9	Slot and discharge, linear diffusers	Supply, extract	Ceiling, side wall, desktop, under window	6	9	Offers vertical or horizontal single or multiple slots. Care must be taken with design of plenum box. Desk top units may incorporate induction of room air.
10	Air handling luminaires	Supply, extract	Ceiling	7	10	As type 9 but single slot only. Normally used in conjunction with extract through luminaires.
11	Ventilated ceiling nozzle	Supply, extract		—	—	Void above ceiling is pressurised to introduce air at low velocity through many single holes or through porous panels. Air entrainment is restricted and natural air currents may affect room air distribution.
12	Nozzles, drum and punkah louvres	Supply	Ceiling, side wall, under window, seat back	—	—	Adjustable type can be rotating drum or swivelling ball, with or without jet for long throws and personal air supply or 'spot' cooling. Fixed multiple nozzles are used for high-induction applications. Velocities depend on throw, noise and induction requirements.

3.2.4.9.4 Air terminal devices

Air can be supplied into a space in a number of ways (HEVAC, 2013); the principal division being between diffusers and perpendicular jets. The resulting airflow patterns from both types of terminal device are strongly dependent upon the presence or absence of the Coanda effect, which determines the effectiveness of the mixing between the conditioned supply air and the bulk air within the space (see section 3.2.2.2.3 'Air distribution').

Table 3.16 below summarises the types of air terminal devices and provides information on the typical face velocities (based on any local control devices being fully open) and noise levels achieved by each type of terminal device.

Diffusers may be radial, part radial or linear and normally utilise the Coanda effect and/or swirl to reduce excessive room air velocities and promote mixing. These usually require manual adjustment and setting to achieve the correct volume and airflow pattern.

A perpendicular jet may be formed by discharging air through grilles, louvres, nozzles or any opening that allows perpendicular airflow. The direction and spread adjustment can be provided using blades and/or swivel adjustment, which require manual setting to achieve the desired airflow pattern.

Supply air terminals can be incorporated into any room surface, for example ceiling (flat or sculptured), floor, wall (high or low level), desk top, seat back or under seats. However the positioning of the air terminals must be carefully considered to ensure they do not instigate occupant complaints relating to noise, draughts or temperature.

3.3 Refrigeration

3.3.1 Introduction

3.3.1.1 General

This section gives guidance on the selection of refrigeration and heat rejection systems and equipment currently available for the built environment. The designer should consider carefully the use of alternative free-cooling and low-energy techniques in the interest of minimising the overall global warming impact arising from the manufacture and operation of refrigeration and heat rejection equipment. Some refrigerants are also potentially environmentally harmful. Reasons for choosing such alternatives should be recorded and substantiated by the designer, see CIBSE Guide F: *Energy efficiency in buildings* (2012b) and Energy Efficiency Best Practice Programme publications* (EEBPP, 1999a/b, 2000a–e).

This section is a development of chapter 4 of CIBSE Guide B (CIBSE, 2001–2). It has been comprehensively revised to take account of developments in the intervening years, in particular to incorporate guidance on health and safety issues and new regulations, and refrigerant information has been updated. More detailed information on low-energy cooling and ventilation strategies can be found in section 3.2 and CIBSE Guide H: *Building control systems* (2009).

This chapter includes sections on:

- design criteria
- system types
- heat rejection and cooling water equipment
- heat pumps
- components
- controls
- commissioning, operation and energy management.

Whilst the process for each application and design will be unique, the route to final selection of a system will follow a common fundamental path and format involving problem definition, idea generation, analysis and selection of the preferred solution. As an aid to this iterative process of system selection, a flowchart is given in Figure 3.68 below.

It should be noted that the guidelines given in this section are to be used by practising engineers who hold a basic knowledge of the fundamentals of refrigeration and heat rejection. As such, mathematical derivations of formulae are not given. References are given where appropriate to enable further detailed investigations of the systems covered.

3.3.1.2 Energy efficiency and environmental issues

Governments worldwide are taking steps to improve energy efficiency in buildings to save money and reduce greenhouse gas emissions. It is CIBSE policy that the energy efficiency of refrigeration and heat rejection systems should be optimised (energy efficiency and environmental references are provided in section 3.3.2).

Refrigeration and heat pump systems often use refrigerant fluids (greenhouse gases) that have a high direct global warming potential. These refrigerants are harmful to the environment if released to the atmosphere. Refrigerant may be released accidentally during installation, maintenance, repair or decommissioning procedures or through leaks from the system. Refrigerant losses from a refrigeration system are referred to as 'direct emissions' (see section 3.3.2.4.9 'Greenhouse gases').

Most refrigeration and heat rejection plant is electrically driven and therefore contributes to power station greenhouse gas emissions. Heat-driven refrigeration plant, such as absorption chillers, will also contribute to primary energy consumption and greenhouse gas emissions, unless driven by waste heat. Refrigeration system power consumption effect is typically referred to as 'indirect emissions'.

Total equivalent warming impact (TEWI) is a way of assessing the overall impact of refrigeration systems from the direct refrigerant-related emissions (e.g. leakage) and indirect fuel-related emissions (power consumption) and is discussed further in section 3.3.2.4.9 'Greenhouse gases'.

It should be noted that lifetime power consumption is normally by far the largest contributor to carbon dioxide emissions.

In the UK, buildings (non-dwellings) account for around 20 per cent of the energy consumption and greenhouse gas emissions and are therefore a key target for action to improve energy efficiency and reduce carbon emissions. The need for refrigeration and heat rejection is increasing in response to a warmer climate and through higher comfort expectations and more sophisticated building usage such as greater use of information and communication technology (ICT). Studies under the Government's Energy Efficiency Best Practice programme (discontinued) suggest that energy consumption related to air conditioning is likely to increase in future.

The chosen refrigeration and heat rejection strategy will influence, and will be influenced by, energy efficiency and environmental issues and any specific targets. The design team should ensure that agreement is reached with the client on any specific energy and environmental targets at an early stage in the design process. Checks should be

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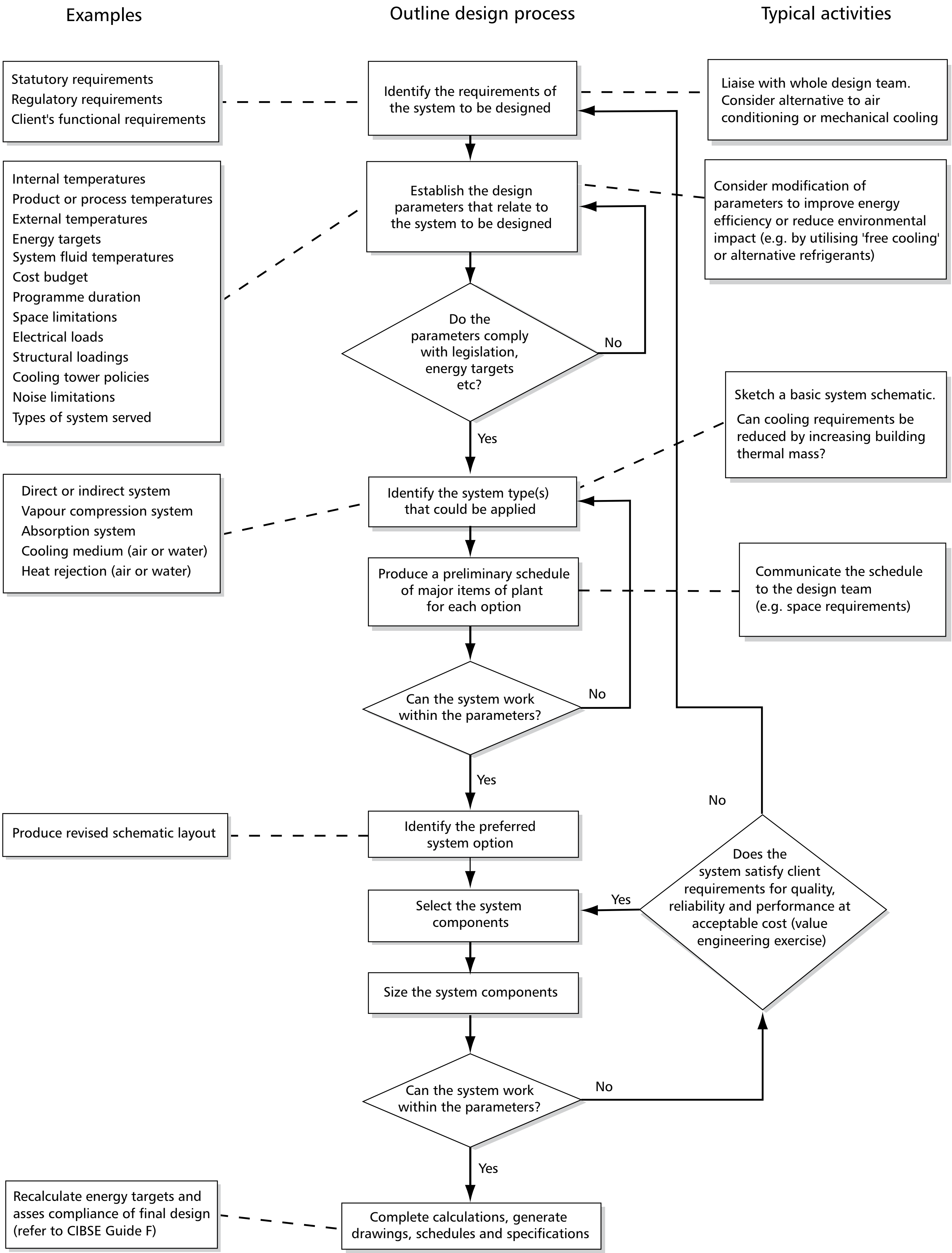


Figure 3.68 'First pass' flow chart for design process

carried out continuously by the design team to ensure that the implications of any changes made during design, construction or subsequent installation are understood and mutually acceptable.

Specific energy consumption targets may be difficult to achieve where, for example, there is an overriding requirement for close control of temperature and humidity. In these cases the design team may need to agree with the client a relaxation of specific energy consumption targets where it can be proven that the need for tight control is an unavoidable requirement of the building use. Specific guidance on achieving energy targets is also given in CIBSE Guide F: *Energy efficiency in buildings* (2012b). It should be noted that tight control of temperature and humidity are not needed to achieve human comfort but could be needed for the preservation of artefacts or the production or storage of some products or materials.

Refrigerant leakage can also have an adverse impact on energy efficiency, as operating a vapour-compression system with too much or too little refrigerant can cause a significant reduction in the cooling performance and energy efficiency of the system (see section 3.3.3.4.4 'Refrigerant leakage').

Refrigeration and heat rejection systems may also use other potentially environmentally harmful substances including antifreeze and water treatment chemicals required to minimise microbiological contamination and corrosion (see section 3.3.4.2.3 'Water treatment').

3.3.1.3 Strategic design decisions

3.3.1.3.1 Introduction

This section addresses the general requirements for the application of refrigeration and heat rejection in buildings. It reviews the factors to be considered in deciding the appropriate design strategy for the building and the client, and highlights points relevant to specific requirements. The design process must be based on a clear understanding of client and end-user needs and expectations and must be followed by correct installation and effective commissioning, handover and building management.

For the purpose of this chapter, refrigeration is defined as the process of removing heat, and heat rejection is defined as the discharge of heat to waste or atmosphere or to a system permitting reclaim or recovery.

Key factors to be considered in determining a refrigeration and heat rejection strategy are summarised as follows:

- end-user requirements
- energy efficiency and environmental issues
- interaction with building fabric, services and facilities
- choice of refrigeration and heat rejection strategy
- associated systems
- whole-life costs
- procurement issues
- commissioning
- operation
- maintenance
- future needs.

An appreciation of the above issues is an essential part of the briefing process.

3.3.1.3.2 End-user requirements

The key end-user requirements that need to be clarified before a refrigeration and heat rejection strategy can be chosen are summarised in Table 3.17 below.

Ideally, where the issues highlighted in section 3.3.1.3.1 'Introduction' have not been covered by a standard specification document, the design team should expect to agree requirements with the client at the onset of the project to optimise the choice of refrigeration and heat rejection strategy and the system itself. If the client is unable to advise on the precise needs, the design team must, as a minimum, make the client aware of any limitations of the chosen design. Requirements may subsequently be adjusted over the course of the project to meet financial constraints or changing business needs. The design team must also advise the client on the impact of any such changes on the final plant performance and life-cycle costs.

The designer should review the need for cooling in relation to the end-user requirements (Table 3.17) and the key factors listed in section 3.3.1.3.1 'Introduction'. Refrigeration and heat rejection systems should only be specified where there is a clear and proven need to meet cooling requirements that cannot be met by simpler and less energy-intensive means.

3.3.1.3.3 Interaction with building fabric, services and facilities

The cooling requirements to be met by a refrigeration and heat rejection system will include the ventilation, air conditioning and other internal cooling loads within the building. These loads will be based on estimates of:

- internal gains determined by the occupants, e.g. occupancy itself, lighting, small power loads and any business-related process
- internal gains determined by the fabric, e.g. insulation, glazing, thermal mass.

Although the architect is, traditionally, associated with fabric-related decisions, the building services engineer must be involved at an early stage and advise on their implications for the building services and the requirements for cooling and heat rejection. The services engineer must therefore be involved in the decision-making process as far is as practical and at as early a stage in the process as possible.

Building Regulations Approved Document L2A (NBS, 2013a) requires that buildings with air conditioning and mechanical ventilation are designed and constructed such that the form and fabric of the building do not result in a requirement for excessive installed capacity.

As a minimum, the building services engineer should be able to enter into a dialogue with the architect on building-fabric-related issues that will impact on the cooling requirements of the ventilation and air conditioning systems, including:

- location
- pollution
- orientation
- form

Table 3.17 Establishing end-user requirements

Issue	Requirement/comments
Client brief	To be developed in the context of the other issues.
Building occupants' activities/processes	Understanding of the business process(es) to be undertaken in the building and their specific cooling requirements including any requirement for tight temperature and humidity control.
Energy/environmental targets	Compatibility with statutory requirements (e.g. Building Regulations) and client company environmental policy (e.g. refrigerant policy, BREEAM certification) (see section 3.3.1.3 'Interaction with building fabric, services and facilities').
	Anticipate future statutory energy/environmental targets or requirements.
Integrated design	Integration with building fabric, services and facilities. Requires co-ordinated approach with the client, architect and other professionals from the outline design (see section 3.3.1.3 'Choice of refrigeration and heat rejection strategy').
Investment criteria	Constraints imposed by 'letability' requirements.
Whole-life costs	Understanding of the client's priorities towards capital costs and issues of whole-life costs (see section 3.3.2.2).
	Has the client been involved in discussions of acceptable design risk?
	Importance of part-load performance.
Provision of controls	Required basis of control, e.g. temperature, humidity.
	Required closeness of control.
	Ability and willingness of the occupants to understand and operate controls: controls for unit air conditioners may be in the occupied space but controls for central chiller plant may be hidden from the user and only accessible to facilities or engineering staff.
	Ability and willingness of the building operator to maintain controls.
Reliability	The business process(es) to be undertaken in the building may demand specific levels of reliability of the refrigeration and heat rejection systems (dealer floors and call centres may represent very high value operations to the owner and information technology (IT)/telecommunications centres may be 'mission critical' operations that require completely separate backup cooling and power supply systems) (see section 3.3.1.3, 'Reliability')
Maintenance requirements	Understanding of the client's ability to carry out, or resource, maintenance (see section 3.3.2.6). Client willingness for maintenance to be carried out in occupied space (e.g. unit and multi-split air conditioning systems).
	Any requirement for 'standard' or 'familiar' components.
Aesthetic and noise considerations	The need for system concealment (visible plant on the roof or at ground level).
	Restriction on placement of cooling towers.
	Restriction on location of noisy plant (e.g. proximity to conference rooms and neighbouring buildings).
	Restrictions imposed by local authorities, building listing etc. (particularly related to plant on roofs or mounted on walls).
Security	Restrictions on size and location of any openings.
Space allocation	Restrictions on space allocated for refrigeration and heat rejection equipment may have a significant effect on the choice of plant, its energy efficiency and the ability to maintain it adequately and safely.
Procurement issues	Time constraints.
	Programming constraints, particularly for refurbishment projects.
Future needs	Adaptability, i.e. the need to cope with future change of use.
	Flexibility, i.e. the need to cope with future changes in work practices within the current building.
	Acceptable design margins: it is important to distinguish, in collaboration with the client, between design that is adequate for current cooling requirements and design that makes sensible agreed allowances for future changes in cooling requirements.

— insulation
— infiltration
— shading
— window choice
— glazing
— thermal mass.

Specific guidance on these issues is given in CIBSE Guide A: *Environmental design* (2015a) and in section 3.2 of this Guide. Some of these issues now have statutory requirements. For example, Approved Document L2A (NBS, 2013a) requires that non-domestic buildings should be reasonably airtight, and that buildings of greater than 1000 m^2 are to be pressure tested in accordance with CIBSE TM23: *Testing buildings for air leakage* (2000a).

The design strategy for the windows and glazing will impact on the provision of daylight, which will in turn interact with the cooling load created by the use of electric lighting. More specific guidance is given in SLL Lighting Guide LG10: *Daylighting — a guide for designers* (SLL, 2015). Heat gains from the lighting should be minimised through:

— selection of appropriate light levels and differentiating between circulation spaces and workstations

— selection of efficient luminaires

— installation of appropriate controls to minimise unnecessary electric light usage

— use of ventilated luminaires to minimise heat gains to the occupied space.

Small power loads arising from IT and office equipment are an increasingly significant component of internal heat gains. It is important that a realistic estimate is made of the anticipated diversity in use of such equipment. The designer should also encourage the client to reduce small power heat gains through:

— the selection of low-energy equipment and the use of power cut-off mechanisms

— the location of shared equipment, e.g. photocopiers, printers and vending machines, in spaces that can be readily cooled (e.g. through the use of opening windows or simple extract ventilation).

The choice of cooling distribution system and terminal units can also affect the requirement for cooling, as well as the size and efficiency of the cooling or heat rejection system. For example, the use of chilled ceilings or beams can allow secondary chilled-water temperatures of 14 °C or higher, which makes the use of simpler heat rejection systems or other free-cooling strategies practical for a greater proportion of the year than is possible with conventional chilled-water-based systems. By treating cooling loads that require low temperatures (e.g. dehumidification systems) separately, the refrigeration efficiency can also be raised. Further guidance on these and other cooling distribution systems and terminals is given in section 3.2 of this Guide.

At an early stage, the designer must agree with the architect and structural engineer any specific requirements relating to the refrigeration and heat rejection systems and the safety of those installing and maintaining it. These include ensuring that there is:

— sufficient space for the plant itself, and for installation and subsequent maintenance procedures: manufacturers' literature for plant and equipment should be consulted for the space requirements around plant for procedures such as the withdrawal of heat exchange coils, compressors etc.

— sufficient access for replacement and damaged parts to be brought into and out of the plant room: adequately sized doorways, access stairs or demountable structural openings may need to be provided; in addition it may be advantageous for lifting beams to be built into the structure in order to move equipment easily into and out of plant rooms

— adequate structural strength to support heavy items of plant such as cooling towers, chillers and water tanks: these items may be located on the building roof, which may need to be specially strengthened.

3.3.1.3.4 Choice of refrigeration and heat rejection strategy

The selection of an appropriate strategy should take into account all of the strategic design decisions discussed in previous sections. It is important that the requirement for cooling is minimised, as this should reduce the energy consumption of the building, minimise maintenance costs (e.g. specialist refrigeration maintenance and water treatment costs) and, in many cases, reduce the life-cycle costs of the building.

The requirement for cooling can also be minimised by selecting an appropriate building ventilation strategy that maximises the use of ambient air for ventilation and cooling instead of providing a full air conditioning system. Figure 3.69 below shows a decision flow chart that may assist this selection process.

The choice of refrigeration or heat rejection strategy should take account of the following guidelines:

— The need for refrigeration can be minimised by using simpler heat rejection or free-cooling strategies during cool weather, such as cooling towers, dry air coolers or refrigerant migration (thermosyphon) chillers (see section 3.3.3.2.4).

— The decision to use simple heat rejection or free cooling in cool weather, in addition to a separate refrigeration system for use in hot weather, requires careful assessment of life-cycle costs.

— Free-cooling systems can be particularly appropriate for buildings where cooling requirements are high and unrelated to ambient temperature, e.g. computer suites and telecommunication switching centres.

Where the need for a refrigeration system is unavoidable, the overall global warming impact of the system (due to energy use and refrigerant emissions) should be minimised through the following guidelines:

— Ensure good refrigeration efficiency through the selection of an efficient machine and by minimising the refrigeration 'lift' (the difference between the temperature of the cooling medium, usually air or chilled water, and that of the heat sink, usually ambient air).

— Where multiple refrigeration machines are installed, machine sizing should be related to the cooling demand profiles in preference to installing a number of equal-sized machines. Good control provisions in such cases are essential (see section 3.3.7).

— Where a vapour compression system is used, minimise the direct global warming effect from possible refrigerant emissions by selecting a refrigerant with a low global warming potential and by selecting a machine with a low specific refrigerant charge (kg of refrigerant per kW cooling capacity) (see section 3.3.4).

Where there is a source of waste or reclaim heat of suitable temperatures, e.g. where there is a case for using a CHP system, consider the use of heat-driven absorption cycle chillers.

3.3.1.3.5 Reliability

The reliability, security of supply, maintenance and backup of the refrigeration or heat rejection system is a major design consideration, the importance of which will depend upon the end user's business operation.

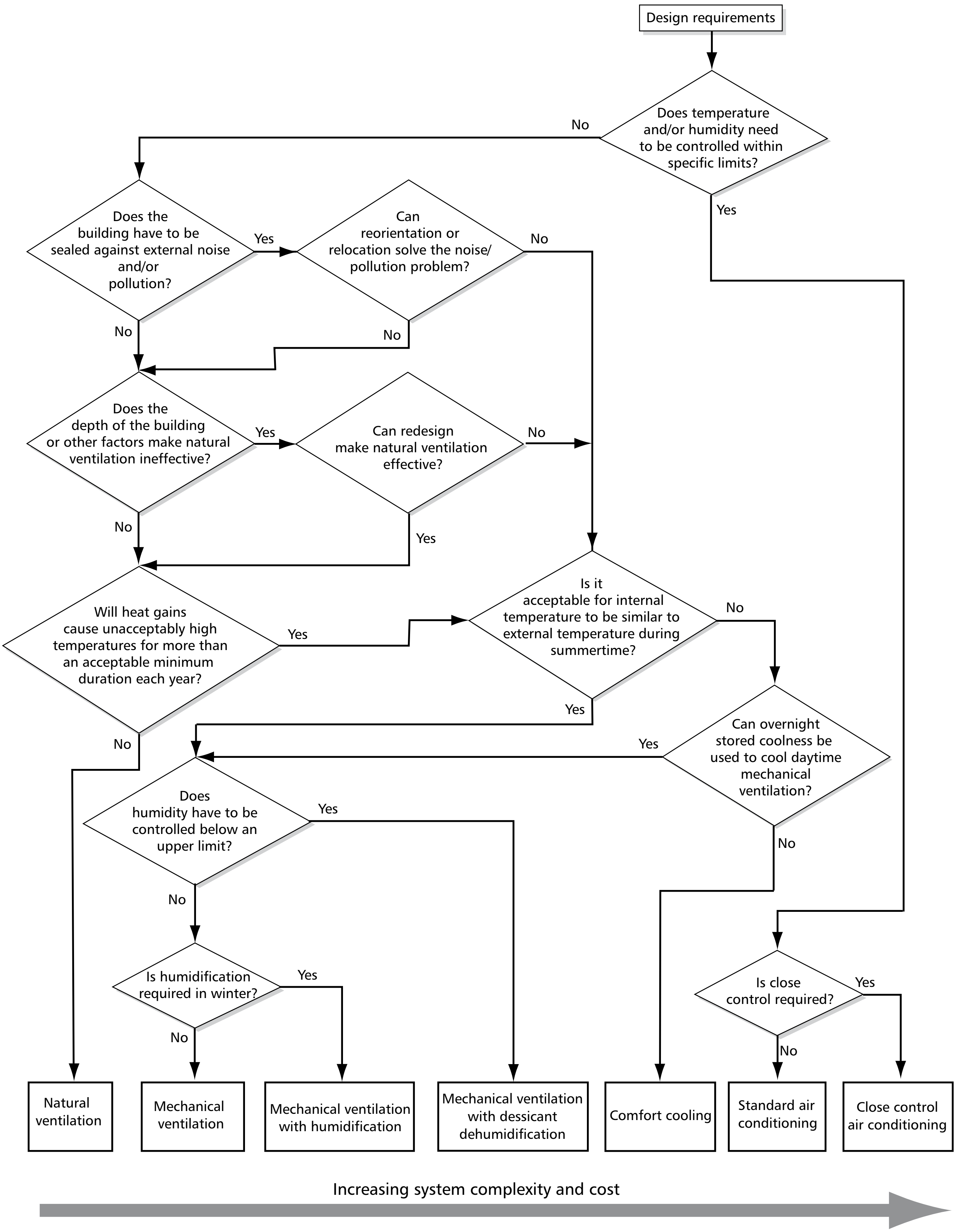


Figure 3.69 Decision flow chart

A distinction must be made between 'mission critical' operations such as IT data and telecommunications centres and standard office comfort cooling applications. The financial consequences of the loss of cooling to a dealer floor may be considerable. Resilient IT cooling systems may require up to 8 kW/m^2 of cooling and may be unable to operate for more than 5–10 minutes after a failure of the cooling system.

It is a requirement of Approved Document L2A (NBS, 2013a) that refrigeration equipment and fans and pumps used to condition general office space are sized to have no more capacity than is necessary for the task. This excludes the capacity of any 'offline' standby equipment.

For standard comfort air conditioning, the loss of some or even all cooling capacity may not be a serious problem. However, 'mission critical' systems may require two independent chiller and chilled-water distribution systems and controls with a backup-generator-fed electricity supply to guarantee the availability of cooling.

Large air conditioned buildings with central chiller plant are often designed with multiple chillers. Multiple chillers offer operational flexibility, some standby capacity and less disruptive maintenance. The chillers can be sized to handle a base load and increments of a variable load and, with a suitable sequencing control strategy, may achieve better energy efficiency than a single chiller installation. It is usual in these situations to provide 'run' and 'standby' pumps. Section 3.3.6.6 provides guidance on piping and control arrangements for multiple chiller installations.

The designer should be aware that the time taken to achieve normal temperatures is affected by the pull-down time (the time taken for the system to achieve normal operating temperatures), which can be relatively long for some large chilled-water systems. Designers should also be aware that some chillers have a long start-up time, which may affect the time necessary to bring on standby plant. The designer should obtain this information from the manufacturer.

It is recommended that the designer should consider undertaking a risk analysis assessment of the system to determine the level of reliability and backup or redundancy that should be provided by the design. It should indicate the failure scenarios, which would dictate client action such as training, keeping spares, provision and safekeeping of original equipment manufacturer's operation and maintenance (O&M) manuals. Carrying out a risk assessment in the early design stages will help the designer to reduce or identify the risk. Further guidance may be found in Tozer (2001) and the introduction to BS EN 378-1: *Specification for refrigerating systems and heat pumps. Safety and environmental requirements* (BSI, 2008a).

3.3.2 Design criteria

3.3.2.1 Introduction

This section outlines the general limiting requirements, including relevant regulations, that need to be considered by the designer when selecting refrigeration and heat rejection plant and equipment with regard to:

— the provision of a safe, comfortable and healthy working internal environment with due consideration of the external environment and relevant regulations:
  - safety
  - noise
  - vibration
  - pollution
  - building regulations
  - commissioning
  - maintenance and decommissioning

— the building fabric:
  - building structure, layout and available plant space
  - aesthetics

— the specific requirements of the individual building, plant and equipment installed therein.

Associated systems may include chilled-water and condenser-cooling distribution pipework and pumps, fans and pumps in heat rejection systems and ventilation air distribution fans. The fans and pumps in a typical air conditioned office building may consume between two and three times the electrical energy consumed by the chillers. It is therefore important that the design and energy efficiency of these associated systems is given as much attention as the main refrigeration and heat rejection systems.

Approved Document L2A (NBS, 2013a) requires that fans and pumps are reasonably efficient and are appropriately sized.

Most air-conditioned buildings experience varying cooling loads depending on the season and time of day. Significant savings in chilled-water pumping energy and fan energy can be made by using variable speed drives to vary the flow rate with the load. Actual energy savings are around 40 per cent at 80 per cent flow and 80 per cent energy savings at 50 per cent flow (the actual savings are always lower than the theoretical, cube law, power savings predicted by the fan and pump laws). It is also important not to oversize fans and pumps, especially when manual-flow-regulating valves and dampers are used, as these simply increase system pressure. Although the use of variable speed drives can avoid the energy penalty associated with manual-flow-regulating valves, excessive oversizing should still be avoided because of the capital cost implications of oversized components.

Water treatment to prevent scaling, microbiological contamination and corrosion is recommended for chilled-water distribution systems and is a legal requirement for any evaporative heat rejection system. The capital and running costs of water treatment may be significant and must be taken into account in calculating life-cycle costs to assess the economics of alternative refrigeration and heat rejection systems.

3.3.2.2 Costs

3.3.2.2.1 Whole-life costs

It is important to consider the whole-life cost of all of the available refrigeration plant options to ensure that the end user can make an informed decision on how to proceed.

The whole-life cost analysis should factor in:

— capital cost

— installation cost

— commissioning costs

— running costs (power, water, chemical)

— maintenance costs

— disposal.

While capital costs is an important consideration, the actual cost of running the plant across its full life can be a major consideration to the end user and should be factored into the final basis of selection.

It is important that the designer ascertains the capability and willingness of the client to maintain an efficient but more complicated system. For example, a simple system that requires only one service visit per year may be more suitable for some clients than a highly efficient but complicated system that requires three-monthly service visits.

Minimising the environmental impact of building cooling systems is inextricably linked to life-cycle costs. For example, systems with the least whole-life costs are likely to be those with the lowest energy consumption and therefore have the least impact on global warming.

An assessment of whole-life costs should take account of the availability of enhanced capital allowances (ECAs). The Government's Enhanced Capital Allowance scheme is devised to encourage businesses to invest in low-carbon technologies. It enables businesses to claim 100 per cent first year capital allowances on investments in energy-saving technologies and products. The list, which includes a range of refrigeration components and systems, including variable speed drives, is subject to constant update and can be consulted at www.eca.gov.uk.

3.3.2.2.2 Procurement issues

When specifying components and plant, the designer should take account of the need for, and likely availability of, replacement parts and spares. Refrigeration and heat rejection plant is commonly the most expensive single item of plant and it is important that spares and replacement parts are readily available at reasonable cost for the anticipated lifetime of the equipment.

The designer should consider the available refrigerant options. The choice of refrigerant is an important factor in the overall plant selection process and is factored into environmental assessment models including BREEAM. For additional information refer to Institute of Refrigeration Guidance Note 18: *Refrigerant selection and system design — the role of HFCs* (IoR, 2009a). The designer should keep up-to-date with current and developing regulations and standards.

Some components and systems are already subject to specific health and safety related regulations. These include cooling towers, refrigerants and most pressure systems (which covers most vapour-compression refrigeration systems). Health and safety related regulations are constantly being updated and their scope widened. In many cases, the implications can include increased cost of ownership. The designer should keep up to date with current and developing regulations and standards and ensure that the most up-to-date information is used when carrying out life-cycle cost studies and when specifying equipment to procure.

3.3.2.2.3 Future needs

The future needs of the client should be discussed and agreed at the initial design stage. Future needs may relate to potential changes or additions to the cooling load of business processes in the building (e.g. telecommunications and IT equipment), occupancy densities or the floor area covered by an air conditioning system.

Future needs may be allowed for by the provision of sufficient additional space for the installation of separate refrigeration and heat rejection plant. Alternatively, provision may extend to the installation of the additional plant, pipework, controls etc. at the initial construction phase, sized to cater for any foreseeable requirements.

Initial oversizing of an individual refrigeration plant is possible subject to consideration of the system design for part-load operation. It is important to assess if installing additional capacity at the outset will maintain efficiency and reliable operation.

Another alternative is to provide additional refrigeration capacity by an additional chiller in a multiple chiller installation. The excess capacity could then be left offline until required. Careful design of the piping and pumping arrangements, including provision of inverter-driven pumps, should reduce the impact on pumping power and flow control (see section 3.3.3.5.1(b)).

As part of the consideration of future needs, the designer should be aware of new regulatory requirements that have been published but are yet to come into force.

3.3.2.3 Energy efficiency assessment

Traditionally, the energy efficiency of a refrigeration system is defined as the coefficient of performance (COP):

$$\text{COP} = \frac{\text{Refrigeration effect (kW)}}{\text{Power input of compressor (kW)}}$$

The quoted COP usually offers an assessment of compressor performance at full load. The overall performance of a refrigeration system relies on the interaction of all of the system components.

In practice, the coefficient of system performance (COSP) is more useful for comparing different systems. COSP includes the power consumed by ancillary equipment associated with the refrigeration system, including condenser fan motors, condenser water pumps, electrical controls and cooling tower fans and pumps. COSP does not include the power consumed by the chilled-water pumps or ventilation pumps.

$$\text{COSP} = \frac{\text{Refrigeration effect (kW)}}{\begin{array}{c}\text{Power input of compressor and}\\ \text{ancillary motors and controls (kW)}\end{array}}$$

COSP is also limited in that it only offers an assessment of performance at full-load conditions.

The majority of refrigeration equipment used in building services applications comprises packaged units either for stand-alone operation or integration with heat rejection plant on site.

'Energy-efficiency ratio' (EER) is an alternative term for energy efficiency of chillers. Within Europe this term has an identical definition to CoSP above.

The European EER definition should not be confused with the United States EER definition set out in ANSI/AHRI 550/590 (AHRI, 2011), where EER is defined as:

$$\text{EER} = \frac{\text{Refrigeration effect (kBtu/h)}}{\text{Compressor power input (kW)}}$$

CoP is dimensionless, but the United States units of EER are $\text{kBtu}\cdot\text{h}^{-1}/\text{kW}$, and the adoption of the term EER in Europe instead of CoSP can cause confusion.

BS EN 14511: *Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling* (BSI, 2013) defines test conditions for cooling equipment including chiller installations as applicable to building services.

This standard includes an allowance for ancillary power for packaged equipment, condenser fans for air-cooled condensers and chilled and condenser water pumps as applicable to each type of chiller. It has been adopted by Eurovent and therefore by most manufacturers.

The standard is used as the basis for test data for chillers using EER values for operation at capacity steps at specific air or water on temperatures as set out in Table 3.18.

Table 3.18 Part-load efficiency weightings for chillers

Capacity	Air on (air cooled)	Condenser water on (water cooled)	Part L 'SEER' weightings	Eurovent 'ESEER' weightings
100%	35 °C	30 °C	0.12	0.03
75%	30 °C	26 °C	0.32	0.33
50%	25 °C	22 °C	0.36	0.41
25%	20 °C	18 °C	0.2	0.23

Each capacity step is assigned a weighting (as shown in the table) that is used to calculate the seasonal energy efficiency ratio (SEER and European ESEER) that can be used in life-cycle cost analysis when comparing system and product options.

Care should be taken to ensure that the correct values are used in the relevant building energy assessment models.

The designer should seek to maximise the chiller or cooling system SEER. This can either be inherent in the selection of efficient equipment or a result of designing systems to minimise the pressure (temperature) differential between the evaporating temperature and the condensing temperature. This is usually effected by maximising the size and efficiency of the heat absorption and rejection heat exchangers. A reduction in condensing temperature of 1 K or an increase in the evaporating temperature of 1 K reduces energy use by 2–4 per cent. Guidance on optimising CoP and CoSP is given by CIBSE Guide F: *Energy efficiency in buildings* (2012b).

3.3.2.4 Safety and environmental design considerations

3.3.2.4.1 General safety

The designer has a responsibility to ensure that the design of the refrigeration and heat rejection system as a whole takes into account all the necessary provisions for safe o&m, as well as the necessary monitoring, warning and automatic protection features to ensure that the system remains safe during normal operation and during times of component failure. Reference should be made to BS EN 378: *Specification for refrigerating systems and heat pumps. Safety and environmental requirements* (BSI, 2008a–d) and in particular BS EN 378-2 (BSI, 2008b) and BS EN 378-3 (BSI, 2008c).

3.3.2.4.2 UK health and safety legislation

The designer should take account of the requirements of the Health and Safety at Work etc. Act 1974 (HMSO, 1974) and all related regulations. UK health and safety regulations with specific requirements for refrigeration and heat rejection systems include:

— The Management of Health and Safety at Work Regulations 1999 (TSO, 1999a), which require employers to assess the risks to the health and safety of their employees and to take appropriate measures to prevent or control those risks

— The Pressure Systems Safety Regulations 2000 (TSO, 2000a) require that all refrigeration systems with an input power greater than 25 kW are maintained properly and are subject to a written scheme for regular safety inspections

— The Pressure Equipment Regulations 1999 (TSO, 1999b) concern the design, manufacture and supply of equipment. They cover all pressure equipment including refrigeration compressors, pipework, heat exchangers and safety devices. They require that equipment is manufactured and installed to meet certain requirements and is subject to conformity assessment procedures

Control of Substances Hazardous to Health Regulations 1999 (TSO, 1999c) (COSHH) require employers to ensure that exposure of their employees to substances hazardous to health is either prevented or, where this is not reasonably practicable, adequately controlled. COSHH would normally cover exposure to refrigerants in plant rooms and the control of *Legionella* in water systems including cooling towers and evaporative condensers.

The Construction (Design and Management) Regulations 2007 (TSO, 2007b) (CDM) require designers to prepare a health and safety file for the client on how to manage the safety risks when the plant is maintained, repaired, renovated or decommissioned

The Notification of Cooling Towers and Evaporative Condensers Regulations 1992 (HMSO, 1994) requires that the local authority is notified of all cooling towers and evaporative condensers.

Specific guidance on meeting the requirements of these regulations for vapour compression refrigeration systems is given in the Institute of Refrigeration Safety Codes (IoR, 2009c/d, 2013, 2015a). The Codes also give guidance on health and safety risk assessments for refrigeration systems. Guidance on compliance with the regulations with respect to the risk of exposure to legionella is given in the HSE's Approved Code of Practice and guidance L8: *Legionnaires' disease: The control of Legionella bacteria in water systems* (HSE, 2013).

3.3.2.4.3 Refrigerant safety

Refrigerants, their mixtures and combination with oils, water or other materials present in the refrigerating system can present risks to both people and property. Depending on the refrigerant used, the following risks can be caused by the escape of refrigerant from refrigeration systems:

- fire
- explosion
- toxicity
- caustic effects
- freezing of skin
- asphyxiation
- panic.

Other risks related to refrigerants include bursting or explosion due to over pressure or failure of some part of the refrigeration system. These risks can be caused by poor system design, maintenance or operation and, in the worst case, can lead can lead to significant property damage and danger to people.

The risks associated with the escape of refrigerant and the risks of systems bursting or exploding due to over pressure of refrigerant or equipment failure should be minimised by complying with relevant regulations, codes and standards, some of which have been detailed above. In addition, it is CIBSE policy that the requirements of BS EN 378 (BSI, 2008a–d) should be complied with. The Institute of Refrigeration Safety Codes (IoR, 2009c/d, 2013, 2015a) provide specific guidance on the requirements of BS EN 378.

Safety requirements specific to particular refrigerants are given in section 3.3.3.4.2 'Refrigerant safety'.

3.3.2.4.4 Legionella

Any system that contains water at between 20 °C and 45 °C is at risk of supporting colonies of *Legionella*. If the system has the means of creating and disseminating breathable water droplets or aerosols, it is at risk of causing exposure to *Legionella*, the cause of a potentially fatal disease in humans. Cooling and heat rejection systems that incorporate a cooling tower or evaporative condenser are thus at particular risk of supporting the bacteria that could cause *Legionella* infection.

Relevant regulations (see section 3.3.2.4.2) that must be complied with include:

- The Management of Health and Safety at Work Regulations 1999 (TSO, 1999a)
- Control of Substances Hazardous to Health Regulations 1999 (COSHH) (TSO, 1999c)
- The Notification of Cooling Towers and Evaporative Condensers Regulations 1992 (HMSO, 1992).

Practical guidance on complying with these regulations is given by the HSE's Approved Code of Practice and guidance L8: *Legionnaires' disease: The control of Legionella bacteria in water systems* (HSE 2013) and CIBSE TM13: *Minimising the risk of Legionnaires' disease* (2013a).

The regulations impose specific legal duties on employers and building owners/operators to identify and assess all potential sources of *Legionella*, prepare a scheme for preventing or controlling the risk, implement, manage and monitor precautions and keep records of the precautions.

Designers and installers have legal responsibilities to minimise the risk of *Legionella* infection through the design, construction and commissioning of cooling and heat rejection systems. These include the design and construction of cooling towers and evaporative condensers, their location, water treatment systems and water distribution system design (see section 3.3.4).

Other forms of heat rejection equipment, such as dry air coolers, normally have no risk of causing legionella infection. However, if rainwater is allowed to collect in idle equipment in warm conditions, there is a risk of *Legionella* multiplying and being distributed in an aerosol when fans are restarted. Care should be taken when designing, installing and maintaining systems to ensure that water drains away freely. This includes making sure that components are properly levelled, that drains are free falling and kept clear and that there is adequate access for cleaning.

3.3.2.4.5 Operation and maintenance (O&M)

The design of the system and selection of components should allow the system to be operated and maintained safely. The designer should also ensure that the O&M requirements are within the capabilities and resources of the intended owner and operator.

The Construction (Design and Management) Regulations 2007 (TSO, 2007b) require that designers consider the need to design in a way that avoids risks to health and safety or reduces these risks as far as is practicable so that the project can be constructed and maintained safely. This would include, for example, provision of safe access to cooling towers and evaporative condensers to allow their regular inspection and cleaning, and sufficient space and lighting around refrigeration plant to allow regular refrigerant leak checks. The regulations require the designer to provide the client with a health and safety file on how to manage the safety risks when the plant is maintained, repaired or decommissioned. This document is in addition to the O&M manuals and to the building log book recommended in Building Regulations Approved Document L2A (NBS, 2013a) .

O&M procedures should comply with relevant UK health and safety regulations, which are summarised in section 3.3.8.3 and BS EN 378 (BSI, 2008a–d). In particular the Pressure Systems Safety Regulations 2000 (TSO, 2000a) require that systems are maintained properly and that all refrigeration systems with an input power greater than 25 kW are subject to a written scheme for regular safety inspections. The Institute of Refrigeration's Safety Codes

(IoR, 2009c/d, 2013, 2015a) provide guidance on complying with relevant UK regulations and BS EN 378, as well as practical guidance on what constitutes a suitable maintenance and inspection schedule.

The COSHH Regulations (TSO, 1999c) include the risk of exposure to refrigerants in plant rooms and elsewhere in buildings. Plant rooms should be adequately ventilated and provided with refrigerant gas detectors linked to alarms at the occupational exposure limit concentrations. Specific requirements are given in BS EN 378 (BSI, 2008a–d) and are summarised in section 3.3.2.4.

3.3.2.4.6 Noise

Refrigeration and heat rejection plant produces noise pollution in the form of:

— *mechanical vibration*: which can be transmitted through the building structure and generate noise in occupied rooms

— *airborne noise*: which can be a nuisance to the occupants of the building, neighbouring buildings and operatives inside plant rooms.

Dealing with noise pollution is an important aspect of design and requirements for its control are given in CIBSE Guide B4 (2016c).

Designers and employers should be aware that they have specific requirements under the Noise at Work Regulations 1989 (HMSO, 1989) relating to the exposure of employees to noise in the workplace. This is especially relevant to noise levels in refrigeration system plant rooms and requires a risk assessment and the implementation of measures to protect people from hazardous noise levels.

Designers should also be aware of the requirement to establish the permissible noise emissions from plant that can be derived from acoustic limits set out in planning permission requirements. (*Note*: acoustic limits can vary over time in accordance with local authority requirements.)

There are several approaches to preventing noise problems and these are outlined as follows:

— Selection of quieter plant: this is the simplest option but may not be practicable or economic. The size, speed and design of the fans on air-cooled chillers and other heat rejection equipment can also have a significant impact on noise.

— Location of plant: noisy plant such as refrigeration compressors should be located in appropriately designed and constructed plant rooms or well away from occupied areas.

— Plant rooms containing very noisy equipment should be constructed with high mass floor, walls and roof, and particular care should be taken to seal any potential noise leakage paths.

— Scheduled maintenance programmes should be adhered to with particular care to ensure that moving parts are adequately lubricated and that worn or loose parts are replaced or tightened.

— Airborne noise from plant such as chillers and cooling towers can be partially blocked using barriers, with or without sound absorbing material, placed between the noise producing plant and the occupied areas.

— Airborne noise from cooling towers can also be reduced by fitting an acoustically lined vent cowl.

— Where barriers are not sufficient, plant can be enclosed inside an acoustic enclosure with silencers at the fan outlets and air intakes.

— Noise resulting from vibration can be reduced by placing plant on suitable anti-vibration mounts.

Where plant is enclosed in structures such as an acoustic enclosure, the designer must ensure that adequate access space is provided around the plant to allow for maintenance procedures to be carried out safely. Space should also be provided to ensure that the plant receives adequate airflow for cooling and heat rejection.

Enclosing plant or fitting silencers to air inlets and outlets may have a detrimental effect on energy efficiency, and this should be taken into account.

The method of control of plant can affect noise levels. For instance, where fans are speed controlled to suit the heat rejection load and ambient temperature, they are likely to operate for much of the time at reduced speed and hence produce less noise than at the design load. The designer and contractor should consult the manufacturer or supplier at an early stage so that all available options may be considered.

Particular care is needed when installing small refrigerant condensers on the outside walls of buildings as is common with split or multi-split air conditioning systems. Close proximity to windows and neighbouring buildings should be avoided. The start-up noise from compressors should be considered, as well as the general noise when operating compressors and fans.

Where structure-borne noise is a particular issue, such as in residential-type buildings, absorption chillers should be considered instead of vapour compression plant. However, energy-efficiency issues should be taken into account (see section 3.3.3.7).

The relatively low efficiency of absorption chillers also means that significantly larger heat rejection systems or cooling towers are required, taking up more space and possibly increasing noise levels outside the building compared with vapour-compression chillers.

3.3.2.4.7 Pollution

Many refrigerants, oils and other chemicals used in refrigeration systems may cause pollution to the environment. The system designer and the equipment specifier should be aware of any environmentally damaging substances or materials used in the refrigeration and heat rejection equipment.

Hydrochlorofluorcarbon (HCFC) leakage is a concern with respect to ozone depleting potential (ODP) and global warming potential (GWP) for existing systems and hydrofluorocarbon (HFC) leakage is a concern with respect to GWP.

The requirements of the following regulations must be met.

— EC Regulation No 2037/2000 (EC, 2000a) on ozone depleting substances: as well as phasing out and controlling use of chlorofluorocarbons (CFCs) and HCFC refrigerants, this regulation also includes legal requirements for the minimisation and avoidance of refrigerant emissions and leakage (see section 3.3.3.4.4).

— Environmental Protection Act 1990 (HMSO, 1990): Section 33 of the Act states that it is illegal to 'treat, keep or dispose of controlled waste in a manner likely to cause pollution to the environment or harm human health'. Most refrigerants and oils come under the category of controlled waste. Section 34 places a duty of care on all those who handle controlled waste to ensure that it is legally and safety dealt with. This includes preventing its escape.

— The Fluorinated Greenhouse Gases Regulations 2015 (F-Gas Regulations) (TSO, 2015) place a responsibility on the operator to detect and prevent leakage of HFC refrigerants.

It is CIBSE policy that the requirements of BS EN 378 (BSI, 2008a–d) are complied with as well as the above statutory regulations.

The UK is party to a number of international agreements including the Montreal Protocol (UNEP, 2000) and the Kyoto Protocol (UN, 1998). The Montreal Protocol is implemented through EC Regulation 2037/2000 (EC, 2000a). The Kyoto Protocol addresses the issue of the emission of man-made greenhouse gases including many refrigerants (see section 3.3.1.2).

Whilst the designer should consider the impact of the choice of substances used within a system, consideration should also be given to ways in which the severity of pollution could be reduced in the event of a leak. This can be by means of a simple refrigerant-gas-detection system that raises an alarm. Alternatively, it could involve the use of a complex system that initiates automatic shutdown in the event of a leak, thus minimising the volume of material leaked.

3.3.2.4.8 Ozone-depleting substances

CFCs are no longer in common use in the UK.

HCFCs have been widely used as refrigerants but can no longer be used in new refrigeration systems or added to existing systems.

The Montreal Protocol (UNEP, 2000) is an international treaty to protect the stratospheric ozone layer through controls on the consumption and production of ozone-depleting substances, including refrigerants. The requirements of the current version of the Montreal Protocol are detailed in Table 3.19.

Within the European Union, the Montreal Protocol is enforced through EC Regulation No 2037/2000 on ozone depleting substances (EC, 2000a) (as amended by Regulations 2038/2000 (EC, 2000b) and 2039/2000 (EC, 2000c)). This regulation is law throughout the European Union and is implemented in the UK by the Environmental Protection (Controls on Ozone-Depleting Substances) Regulations 2002 (TSO, 2002b). The regulation requires a faster phase-out of HCFCs than the Montreal Protocol and has other requirements related to prevention of leakage and the use of recycled CFCs and HCFCs.

3.3.2.4.9 Greenhouse gases

Under the Kyoto Protocol (UN, 1998), many of the developed countries agreed to targets to reduce their emissions of greenhouse gases (carbon dioxide, methane, nitrous oxide, HFCs, perfluorocarbons and sulphur hexafluoride). Individual countries are entitled to set greenhouse gas reduction gas targets and reference should be made to current national requirements.

For refrigeration systems, the issue of greenhouse gas emissions is complex because of the direct impact of the emission of greenhouse gas refrigerants and the indirect impact of the use of energy, with its associated carbon dioxide emissions, to operate the refrigeration system.

The total equivalent warming impact (TEWI) is a way of assessing the overall impact of refrigeration systems from the direct and indirect emissions. Designers should seek to minimise TEWI through the selection of an appropriate refrigeration machine and refrigerant and by optimising the selection of components and system design for the best

Table 3.19 The Montreal Protocol

Product	Date and restriction imposed			
	Developed countries		Developing countries	
CFCs	1/1/1996	Consumption banned	1/1/2010	Target for phase out
HCFCs	1/1/1996	Freeze consumption at 2.8% of 1989 CFC consumption plus 1989 HCFC consumption (in ODP tonnes)	1/1/2016	Freeze consumption at 2015 level
			1/1/2040	Consumption banned
	1/1/2004	Consumption limited to 65% of 1996 level		
	1/1/2010	Consumption limited to 35% of 1996 level		
	1/1/2015	Consumption limited to 10% of 1996 level		
	1/1/2020	Consumption limited to 0.5% of 1996 level		
	1/1/2030	Consumption banned		

Note: 'Consumption' means production plus imports minus exports (all values are in ODP tonnes). Recycled and reused product is excluded from the above controls and some exceptions are possible for limited 'essential uses'.

energy efficiency (indirect greenhouse gas emissions will normally significantly exceed direct greenhouse gas emissions over the life of the refrigeration plant).

The direct global warming impact of a gas is measured by the relative global warming potential of 1 kg of the gas relative to 1 kg of carbon dioxide. It is usual to base this on a 100-year time horizon, although in reality many greenhouse gases have atmospheric lifetimes longer than this. Global warming potentials for common refrigerants are shown in Table 3.20.

Table 3.20 Global warming potentials of common refrigerants

Refrigerant	Global warming potential 100-year time horizon (GWP100)
R134a (HFC-134a)	1300
R404A (HFC-404A)	3800
R407C (HFC407C)	1600
R410A (HFC-410A)	1900
R290 (propane)	3
R600a (isobutane)	3
R717 (ammonia)	0

Source: BS EN 378-1 (BSI, 2008a) (values vary slightly depending on source of data)

For further information on other refrigerants refer to BS EN 378-1 (BSI, 2008a), *TEWI Guidelines for Calculation by BRA* (IoR, 2006) and to manufacturers' literature.

Guidance on refrigerant leakage rates is available from the Institute of Refrigeration (see REAL Zero Skills guides (IoR, 2015b).

3.3.2.4.10 Other pollutants

Other refrigerants and substances used in refrigeration and heat rejection systems are potentially harmful to the environment and to human health and are therefore subject to the requirements of the Environmental Protection Act 1990 (HMSO, 1990) and should be disposed of as controlled waste or in some cases as special waste. These pollutants include the following:

— *Ammonia*: used as a refrigerant in some vapour-compression and absorption chillers. Ammonia is highly toxic to people, aquatic organisms and fish so should never be discharged to surface water courses. However, ammonia is lighter than air and safe to discharge to the atmosphere as long as it is in a safe location away from people and buildings.

— *Lithium bromide and water*: used in many absorption chillers and is corrosive.

— *Corrosion inhibitors*: used specifically in absorption chillers (which may also contain alcohol) but may also be used in water-based cooling systems.

— *Water treatment chemicals*: cooling towers and evaporative condensers use special water treatment chemicals including biocides and antifreeze—these may not be accepted into public sewers.

— *Oil recovered from systems*.

3.3.2.5 Building regulations

The designer should ensure compliance with building regulations as set out in section 3.1 of this Guide.

3.3.2.6 Commissioning and maintenance

3.3.2.6.1 Commissioning

In order to ensure that the design is commissionable, the designer should consider how the system should be commissioned at an early stage in the design process and ensure that the necessary components and facilities are provided to allow commissioning to be properly carried out. The designer should seek the assistance of a commissioning engineer where there is insufficient in-house experience. The designer must make sure that a clear description of how the system is intended to operate and the design parameters are clearly stated and recorded either in the design specification or the system design drawings. The requirement for subsequent re-commissioning during the lifetime of the system should also be considered and this should take into account the resources available to the client.

It is particularly important for the success and feasibility of commissioning that the refrigeration-system cooling capacity, controls and safety devices permit stable operation over the specified range of cooling load conditions. Oversizing should also be avoided and standby arrangements should be consistent with the design risk.

Further details on commissioning are set out in section 3.3.8.1.

3.3.2.6.2 Design for maintenance and decommissioning

It is the designer's responsibility under the Construction (Design and Management) Regulations 2007 (TSO, 2007b) to ensure that future maintenance of the plant can be carried out safely. Attention is drawn to the people working on refrigeration plant and their responsibilities within the provisions of the Health and Safety at Work etc. Act 1974 (HMSO, 1974). In particular, people who are responsible for maintenance should be 'competent' (i.e. have the necessary training and knowledge for their task to achieve 'competence'). One way of demonstrating competence is by registration with an accredited registration organisation such as the Air Conditioning and Refrigeration Industry Board (ACRIB) and thereby obtaining a refrigerant handling certificate.

European and UK legislation and standards must be observed. The Institute of Refrigeration codes for safety in the design, construction and installation, commissioning, inspection, maintenance and decommissioning of vapour compression refrigerating systems (Institute of Refrigeration (2009c/d, 2013, 2015a), amplify the requirements of BS EN 378-4 (BSI, 2008d) and take account of relevant UK regulations.

The designer should be aware that some systems require very different or specific maintenance compared with standard equipment, for example:

— Evaporative cooling towers and condensers require meticulous water treatment and maintenance, which some users and/or operators may be incapable of providing (or unwilling). It is possible to subcontract maintenance of these systems to specialist maintenance companies or to the equipment manufacturer.

— Chillers using ammonia as a refrigerant should only be maintained by specialist contractors or the manufacturer.

— The Pressure Systems Safety Regulations 2000 (TSO, 2000a) specify that all vapour compression refrigeration equipment with more than 25 kW compressor input power requires regular periodic inspection by a competent person in accordance with a written scheme of inspection.

The designer must be aware that certain components may fail and will need to be replaced during the life of the plant and the designer must also consider the removal of plant on decommissioning at end of life.

3.3.2.7 Building fabric design considerations

3.3.2.7.1 Building structure and layout

The design and specification of refrigeration and heat rejection systems should take account of specific building structure and layout requirements, including the following.

— *Acceptable structural loads*: roof and floor structures must be strong enough for the specified refrigeration and heat rejection plant. Inadequate strength will result in a need for strengthening or may limit the type or size of plant.

— *Available space inside the building*: space for plant rooms inside a building may be restricted due to commercial requirements to maximise the lettable area.

— *Available roof space*: competing space requirements from other equipment (such as mobile phone masts and other telecommunications equipment) may restrict the space for heat rejection plant. This may cause heat rejection plant to be installed too close together, which could reduce cooling capacity, efficiency and reliability in hot weather.

— *Building layout constraints*: these may make access to plant areas difficult for maintenance or for replacement of large items of plant such as chillers. Knock-out panels may have to be incorporated into roofs and walls to allow access for future plant replacement.

These building structural and layout requirements should be taken into account at an early stage in the design process, as they will affect the ability of the designer to meet the legal requirements of the Construction (Design and Management) Regulations 2007 (TSO, 2007b). Access for both installation and maintenance of the refrigeration and heat rejection plant is affected by the building layout, as well as the building structural design, and will have an impact on the type of system and size of plant that can be installed.

More energy efficient plant may be larger and heavier than less efficient alternatives and will increase the impact of the above requirements. For example, a more energy efficient chiller may have a larger condenser than a less efficient chiller, and will require more roof space and have a greater weight.

3.3.2.7.2 Aesthetics

Local authority planning consent and architectural considerations may require that external plant meets certain visual requirements. This may call for concealment or screening of plant on building roofs. For example, this might affect cooling towers, chillers and other heat rejection equipment. These conditions may be in addition to any requirements for noise reduction. Listed buildings will have additional requirements.

The designer and contractor should liaise with the architect at an early stage in the project to ensure that any planning, listed building and architectural requirements are known at the outline design stage.

Any requirement for screening or concealment of plant should be assessed for the effect on plant airflow requirements and space for commissioning and maintenance.

3.3.3 System types

System selection should be based on the requirements of the refrigeration and heat rejection strategy determined from the guidance given in section 3.3.1, taking into account the general limiting requirements, including relevant regulations, detailed in section 3.1.

It is assumed that the requirement for some form of building cooling system has been determined, taking into account the guiding principles of strategic design outlined in section 3.1. The designer should select a system to satisfy this cooling demand whilst avoiding or minimising the requirement for refrigeration, taking into account economic and environmental considerations. For example, a simpler heat rejection system may meet part or all of the cooling load without the need to operate a refrigeration system, an example of 'free cooling'.

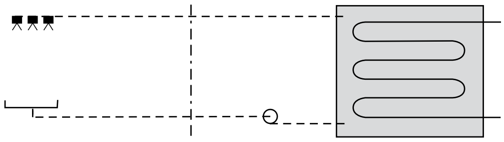
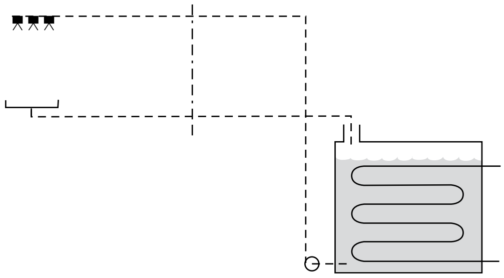
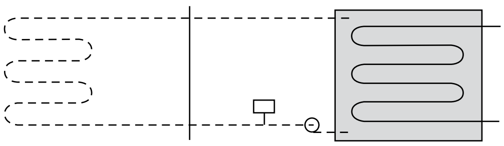
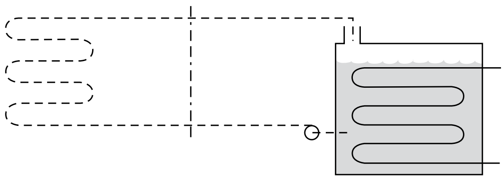
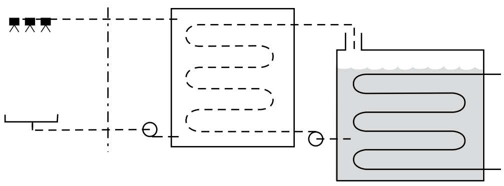
Section 3.3.3 considers systems, including methods of free cooling, which are currently available to implement the strategies set out in section 3.3.1 and meet the regulatory and other requirements defined in section 3.3.1. Details of the constituent items of systems are given in section 3.3.6.

The level of information provided here is not intended to give step-by-step guidance, but to provide a summary of key issues and performance targets that need to be addressed during design. The guidance contained in this section should be read in conjunction with CIBSE Guide A: *Environmental design* (CIBSE, 2015a) and CIBSE Guide F: *Energy efficiency in buildings* (CIBSE, 2012b).

3.3.3.1 Choice of systems

Guidance on selecting cooling systems is given in Table 3.21, which summarises and classifies the main types of systems according to BS EN 378-1 (BSI, 2008a). This is

Table 3.21 Types of cooling system

System type	Description	Area to be cooled	Refrigerating or heat rejection system	Comments
(*a*) Direct system (direct expansion or 'DX' system)	The refrigeration system evaporator is in direct communication with the space to be cooled.			Most efficient cooling system but risk of refrigerant leaks in the occupied areas of the building. May have a relatively high refrigerant charge.
(*b*) Indirect open system	A refrigeration or heat rejection system cools a heat transfer medium which is in direct communication with the space to be cooled.			May use an air washer or spray coil to cool air. Hygiene risks mean that these systems are not widely used.
(*c*) Indirect vented open system	Similar to (*b*) but with an open or vented tank.			
(*d*) Indirect closed system	A refrigeration or heat rejection system cools a heat transfer medium which passes through a closed circuit in direct communication with the space to be cooled.			Widely used with chilled water as heat transfer medium. Lower energy efficiency than (*a*), (*b*) or (*c*) due to additional heat transfer process. Safer than (*a*) because it keeps refrigerant-containing parts out of occupied areas.
(*e*) Indirect vented closed system	Similar to (*d*) but with an open or vented tank.			
(*f*) Double indirect system (open or closed)	A combination of (*b*) and (*d*) where the cooled heat transfer medium passes through a second heat exchanger.			Two additional heat transfer processes make it the least efficient system. Highest safety where toxic and/or flammable refrigerants are used, but little justification for its use.

Pipework containing refrigerant ——
Pipework containing heat transfer medium - - - -

intended as an aid to understanding the types of equipment referred to later in section 3.3.4.

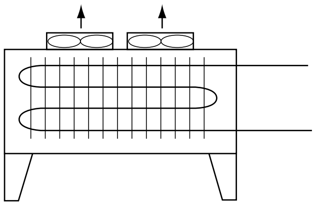
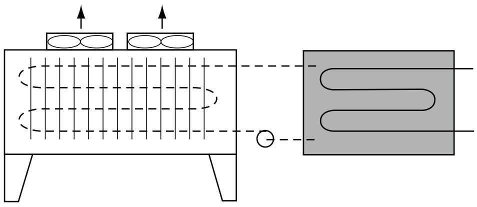
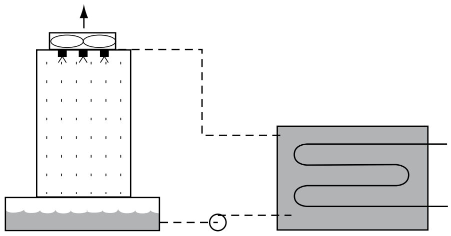
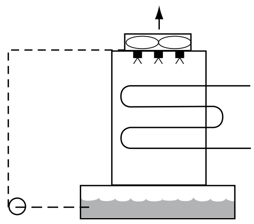
Type (*a*) systems have the disadvantage that components that contain refrigerant are installed in the space or room being cooled. Distributed versions of (*a*) may contain large quantities of refrigerant. Most conventional chilled-water-based systems are based on systems (*d*) or (*e*) and use terminal units such as fan coil units, induction units or chilled beams and panels. With these systems the components that contain refrigerant are outside the space being cooled and this avoids the risk of refrigerant leaks into occupied areas.

Table 3.22 below summarises the main types of heat rejection system and provides guidance on selection and cross-references to further information in section 3.3.4.

3.3.3.2 Free cooling

The aim of 'free cooling' is to minimise or eliminate the need to provide and operate a refrigeration system. Most buildings or processes that require cooling throughout the year have the potential to use free cooling during cool weather.

Table 3.22 Types of heat rejection system (see also section 3.3.4)

System type	Description	Heat rejection system	Comments and section reference
(*a*) Air cooled condenser	Fans induce air flow over finned tubing in which refrigerant condenses.		Convenient and common for chillers up to a few 100 kW. Free of hygiene risks and do not require water piping. Can be adapted to provide free cooling with thermosyphon systems. See section 3.3.4.1.
(*b*) Dry air cooler	Similar to (*a*) but aqueous solution or water is passed through the tubes instead of refrigerant.		Less efficient than (*a*) because glycol an additional heat transfer process, and pumps, are required to reject heat from refrigeration plant. May cool water sufficiently in winter to avoid need to operate a refrigeration plant, i.e. 'free cooling'. See section 3.3.4.3.
(*c*) Cooling tower	Water is sprayed over a packing material. Airflow over the packing evaporates some of the water causing the water to be cooled.		More efficient than (*a*) or (*b*) because less air is required and water is cooled to a few degrees above the wet bulb temp. May cool water sufficiently to avoid the need to operate a refrigeration plant, i.e. 'free cooling'. High maintenance requirement. See section 3.3.4.2.
(*d*) Evaporative condenser	Water is sprayed over tubing in which refrigerant condenses. Airflow across the tubing evaporates some of the water causing the water and the tubes to be cooled.		Can be an efficient method of rejecting heat from a refrigeration plant. Similar maintenance requirements as (c). Can be adapted to provide free cooling with thermosyphon systems. See section 3.3.4.4.

Pipework containing refrigerant ———— Pipework containing heat transfer medium - - - - - - - -

— Free cooling can minimise or eliminate the need to operate a refrigeration system.

— Free-cooling opportunity can be increased through selection of an appropriate cooling system.

— Air and chilled water transport energy consumption may be quite high.

Free cooling usually requires the transport of air or water as a cooling medium and may also require the use of additional fans and pumps for heat rejection at a dry air cooler or cooling tower. The designer should ensure that the overall energy used by a free-cooling system is less than would be consumed if a refrigeration system were operated. Free-cooling systems generally involve moving relatively large amounts of tempered or ambient air and cool water compared with the smaller amounts of cold air or water distributed when a refrigeration system is operated.

The designer can maximise the opportunity for free cooling by selecting a cooling system that requires air or chilled water at a relatively high temperature. For example, chilled ceilings and beams generally require chilled water at between 13 °C and 18 °C, although higher temperatures than these can be used at part load or if comfort conditions are slightly relaxed. Displacement ventilation systems typically supply air at 18 °C to the conditioned space, which increases the proportion of the year that full fresh air can be used and reduces refrigeration requirements in hot weather. Further guidance on these systems is given in section 3.2 of this Guide.

In some systems, the potential for free cooling may be increased by separating the latent and sensible loads. Free cooling may satisfy the sensible cooling load at quite high ambient temperatures, while a separate refrigeration system is provided for dehumidification, as this function usually requires a lower coolant temperature.

Free-cooling systems may be classified as follows:

— environmental free cooling

— chilled water free cooling

— condenser water/chilled water heat recovery

— refrigerant migration chillers (thermosyphon chillers).

3.3.3.2.1 Environmental free cooling

Environmental cooling may be used directly for cooling a building or building-related process (i.e. free cooling), or if it is not cold enough then it may be used as a heat sink for heat rejected from a refrigeration plant. Sources of environmental cooling include:

— ambient air
— ground water
— rivers or lakes
— seawater.

Fresh-air free cooling using ambient air is the simplest form of free cooling. It relies on delivering a sufficient quantity of fresh air to meet all or part of the cooling load when the ambient air temperature is sufficiently low. The fan energy may be quite high and this needs to be assessed in relation to the refrigeration plant energy savings.

Fresh-air free cooling is unlikely to reduce the peak cooling load or size of chiller required in hot weather, because the maximum cooling requirement usually coincides with maximum outside temperature, unless some form of night cooling and stored cooling is utilised. Guidance on night cooling and stored cooling in the building fabric is given in section 3.2.3.4.4. The effectiveness of fresh-air free cooling is also improved by enthalpy control, especially when humidity control is required (see section 3.3.2.1).

Some air conditioner units are also designed to make use of free air cooling. Such units incorporate a damper that can be automatically adjusted to allow 100 per cent outside air free cooling, mechanical cooling with full recirculation or incremental free cooling plus mechanical cooling. They are controlled such that when the return air temperature is greater than the cooling set-point and the outside air temperature is less than the return air temperature (by, say, 2 K), the evaporator fans will continue to run and the damper will modulate between 0 and 100 per cent in order to utilise ambient air to maintain the space conditions.

(*a*) Ground-water cooling

Ground-water cooling at between 11 °C and 13 °C is available in many areas of the UK all year round and is practical for free cooling buildings in conjunction with chilled ceilings and beams or displacement ventilation systems. Chilled ceilings and beams typically require chilled water at between 14 °C and 17 °C, and displacement ventilation uses air at around 18 °C, so they are therefore suitable for use with ground-water cooling without the need for refrigeration plant, except perhaps for dehumidification purposes. Further guidance on ground-water cooling and chilled ceilings and beams and displacement ventilation is given in section 3.2.2.

Closed-loop ground-source systems can also offer a limited amount of passive cooling when used with chilled beam applications. The ground-loop circulating fluid is passed to a heat exchanger instead of the heat pump/chiller. Heat is rejected to the ground loop until the ground temperature is unable to deliver additional passive cooling, whereupon the heat pump/chiller is activated to provide cooling.

(*b*) River, lake and seawater cooling

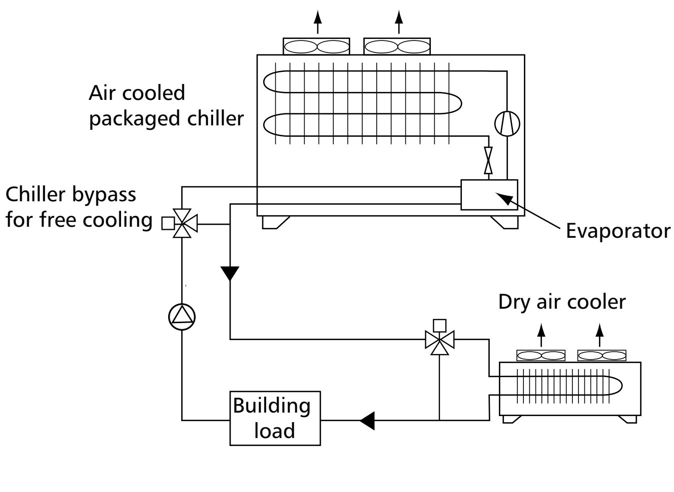
Opportunities for free cooling with river and lake water are fairly limited in the UK due to relatively high water temperatures during much of the year. For this reason, these sources of cooling are more suitable for rejecting heat from refrigeration plant condensers. However, compared with the use of cooling towers and evaporative condensers, river and lake cooling may offer little advantage.

Seawater may be used for rejecting heat from refrigeration plant and is used in the UK for some large, power station cooling systems. Rejection of condenser heat to seawater is fairly common in coastal cities such as Hong Kong. Special provision is needed to prevent corrosion and fouling of heat exchangers.

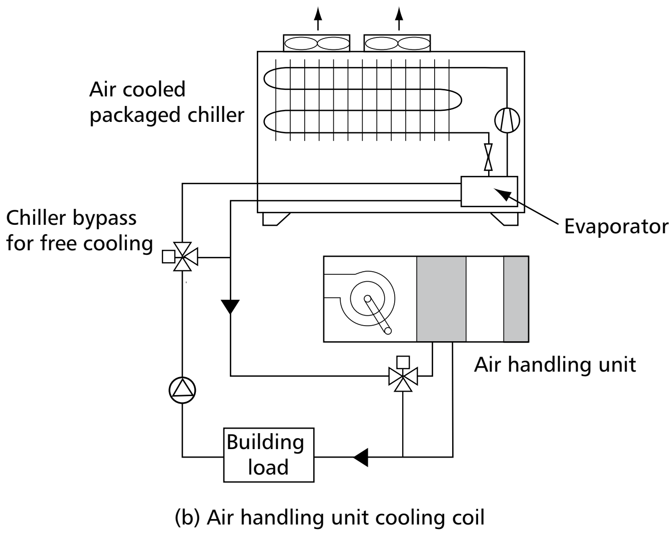
Further guidance on these forms of environmental cooling are given in section 3.3.4.5.

3.3.3.2.2 Chilled-water free cooling

If a cooling load exists when the ambient air temperature is low then cooling of chilled water can be provided by circulating the chilled water through separate dry air coolers or the main air supply cooling coil or dehumidification coil (see Figure 3.70). Automatic control



(a) Dry air cooler



(b) Air handling unit cooling coil

Figure 3.70 Chilled-water free cooling valves bypass the refrigeration system (if one is installed). Precautions must be taken to prevent freezing in cold weather, which may require the use of a water-glycol mixture. This will alter the viscosity and other physical parameters including specific heat capacity and pressure drop through the system. Appropriate allowances must be made for these effects when selecting components including the pumps and dry air coolers. Higher pressure drop will also increase pumping energy consumption. Packaged air-cooled chillers are also available with integral free-cooling coils and diverter valves (see Figure 3.71).

Cooling towers may also be used for chilled-water free cooling, although maintenance requirements are much higher than other methods. Section 3.3.4.2 gives specific information on cooling towers and section 3.3.2.4.4 gives guidance on minimising the risk of *Legionella*. The most thermodynamically efficient system is a direct system (sometimes known as a 'strainer cycle') in which the chilled water is directly circulated through the cooling tower (see Figure 3.72). Cooling towers continually wash dirt and other pollutants from the atmosphere, and very effective filtration is vital to prevent excessive dirt contamination or blockage of the building cooling system. There is also a high risk of corrosion. In systems that use the same cooling towers for rejecting heat from refrigeration system in warm weather, special consideration is needed for the hydraulic balance between the two systems.

Indirect cooling tower based systems use a plate heat exchanger to separate the cooling tower water from the building cooling system (see Figure 3.73). A temperature difference of around 2 K to 3 K across the plate heat exchanger, depending on its selection, will reduce the availability of free cooling compared with a direct system. The advantage, however, is that the building cooling system is kept clean and the two circuits are hydraulically independent. The volume of water that requires regular chemical biocide treatment will also be lower, which will reduce the overall running costs. Further guidance on free-cooling systems is given in BSRIA BG8/2004: *Free cooling systems* (De Saulles, 2004).

3.3.3.2.3 Condenser-water/chilled-water heat recovery

A disadvantage of the free-cooling systems described above in section 3.3.3.2.2 'Chilled-water free cooling' is that free-cooling availability is limited to those periods when the return water temperature from the cooling tower or dry air cooler is lower than the required chilled water supply temperature. However, by locating the free-cooling plate heat exchanger in series with the chilled water return, the warmest water in the building cooling system is in contact with the water from the cooling tower or dry air cooler (Tozer, 2003). This extends the availability of free cooling compared with the systems shown in section 3.3.3.2.2 'Chilled-water free cooling'. Such a system is shown schematically in Figure 3.74. The system is able to operate in a mixed operating mode with the chillers able to top up the cooling to meet the building load. Free cooling can be exploited whenever the cooling tower or dry air cooler water in the heat exchanger is cooler than the building

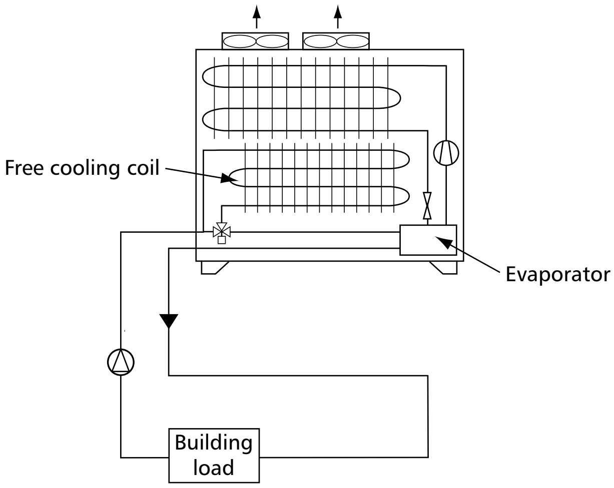


Figure 3.71 Chilled-water free cooling using an integral free-cooling coil in a packaged chiller

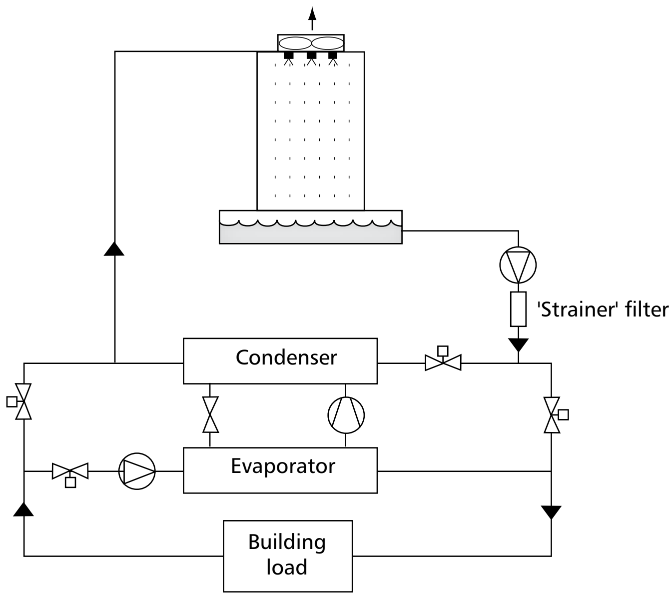


Figure 3.72 Direct cooling-tower-based free-cooling system

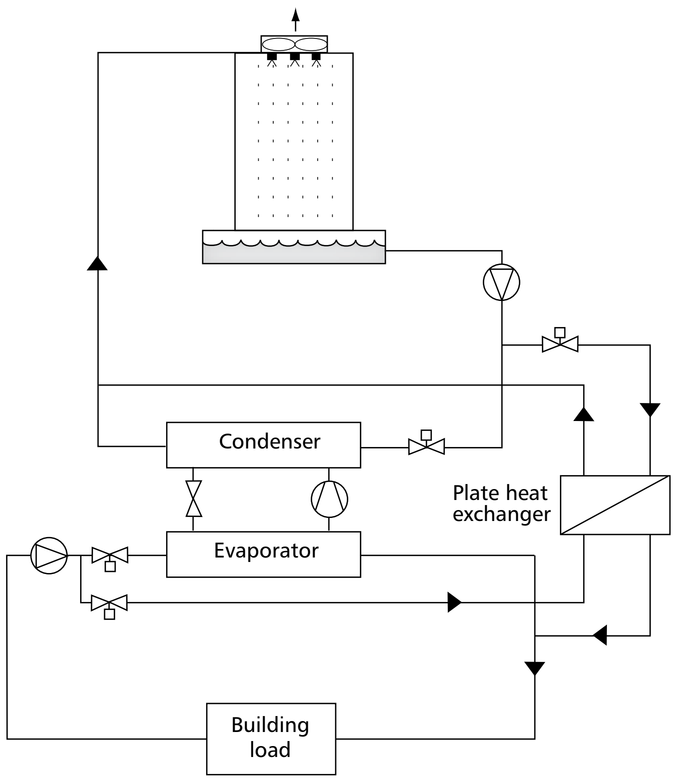


Figure 3.73 Indirect cooling-tower-based free-cooling system

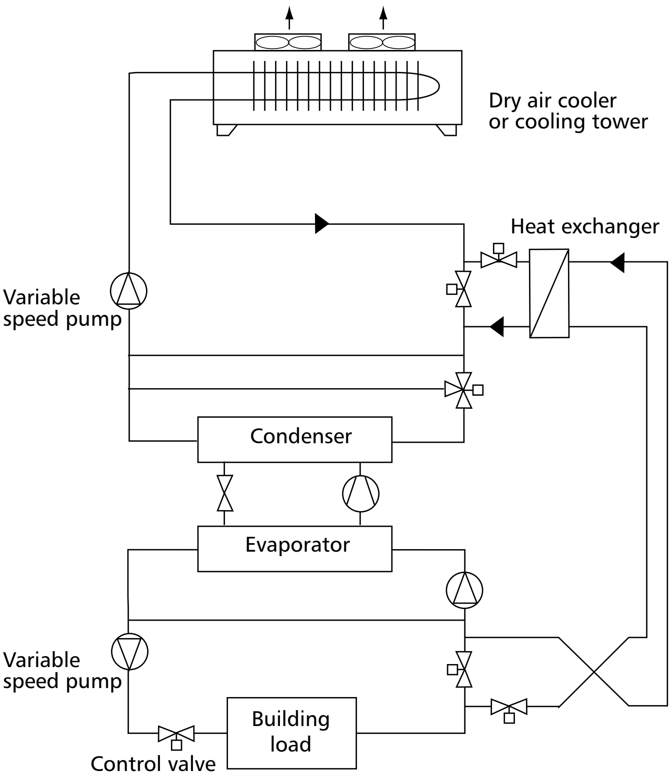


Figure 3.74 Series condenser water/chilled water free-cooling system

return water temperature. However, at some point the increased pumping energy from circulating water through the plate heat exchanger will be greater than the chiller energy savings and the system should then be switched to chiller-only operating mode.

The opportunity of free cooling with series condenser water/chilled water free cooling systems is dependent on the chilled-water return temperature. With conventional constant flow rate chilled-water cooling systems, employing three-port valve control of cooling terminals and coils, the chilled water return temperature falls at part load, which reduces free cooling opportunity. However, in systems with variable-flow chilled water distribution systems, employing two-port valve control of cooling terminals and coils, the return water temperature actually rises at part load. Such systems can therefore be used in conjunction with series condenser-water/chilled-water systems to maximise free-cooling opportunities.

3.3.3.2.4 Refrigerant migration chillers (thermosyphon chillers)

Refrigerant migration or 'thermosyphon' chillers achieve free cooling through bypassing the chiller compressor and expansion valve when the condenser temperature is lower than the evaporator temperature (Blackhurst, 1999). Refrigerant flows to the air-cooled or evaporative condenser (choosing to use an evaporative condenser will increase the period when thermosyphon cooling can be achieved) due to the pressure difference produced by the temperature difference between the chilled water and the ambient air. Condensed refrigerant then returns to the evaporator through gravity circulation. This requires the condenser to be elevated above the chiller and for the refrigerant pipes to be carefully designed to minimise pressure drop (the thermosyphon process can be assisted by provision of a refrigerant pump to guarantee refrigerant circulation). The compressor is operated when insufficient cooling is provided by thermosyphon free cooling.

Ideally, to maximise the opportunity for thermosyphon free cooling, the system should comprise several individual chillers, including any standby chillers, connected in series on the chilled-water side. This ensures that even when one or two compressors need to be operated, the other chillers in the system can continue to operate in free-cooling mode. Figure 3.75 shows a typical thermosyphon chiller system. It should be noted that the lead chiller is always last in line to receive return chilled water. No attempt is made to equalise compressor run hours. Annual system CoSP (see section 3.3.2.3) has been predicted to be 10.5 for a thermosyphon chiller system utilising air-cooled condensers installed in a data-processing centre with a year-round cooling load (Dunsdon, 2001). The use of evaporative condensers would result in a higher annual CoSP.

The duration of the thermosyphon operating mode and the system CoSP can be maximised by using specially treated heat exchanger surfaces on the refrigerant side of the evaporator to improve the heat transfer coefficient. This allows the initiation of refrigerant boiling at very low temperature differences and maximises the time that thermosyphon cooling is available.

For further guidance see section 3.2 of this Guide and BSRIA BG8/2004: *Free cooling systems* (De Saulles, 2004).

3.3.3.3 Vapour-compression refrigeration

The vapour compression refrigeration cycle employs a volatile refrigerant fluid that vaporises or evaporates in a heat exchanger, cooling the surroundings through the absorption of heat. The vapour is then restored to the liquid phase by mechanical compression. The mechanical vapour-compression refrigeration cycle is currently the dominant technique for refrigeration and air conditioning applications.

Figure 3.76 below shows the basic components in a vapour compression circuit and Figure 3.77 illustrates the complete refrigeration cycle on a pressure–enthalpy diagram. The cycle shown is simplified and in particular ignores the effect of pressure drops in pipes and heat exchangers.

The stages in the cycle are as follows:

— *Stage 1 to 2*: low-pressure liquid refrigerant in the evaporator absorbs heat from the medium being cooled (usually water or air) at constant pressure while evaporating to become dry saturated vapour.

— *Stage 2 to 3*: the refrigerant vapour absorbs more heat while in the evaporator and the pipework joining the evaporator to the compressor, to become a superheated vapour.

— *Stage 3 to 4*: the superheated vapour is compressed, increasing its temperature and pressure.

— *Stage 4 to 5*: the hot superheated vapour enters the condenser where the first part of the process is de-superheating.

— *Stage 5 to 6*: the hot vapour is condensed back to a saturated liquid at constant temperature and pressure through being cooled by a coolant (usually air or water);

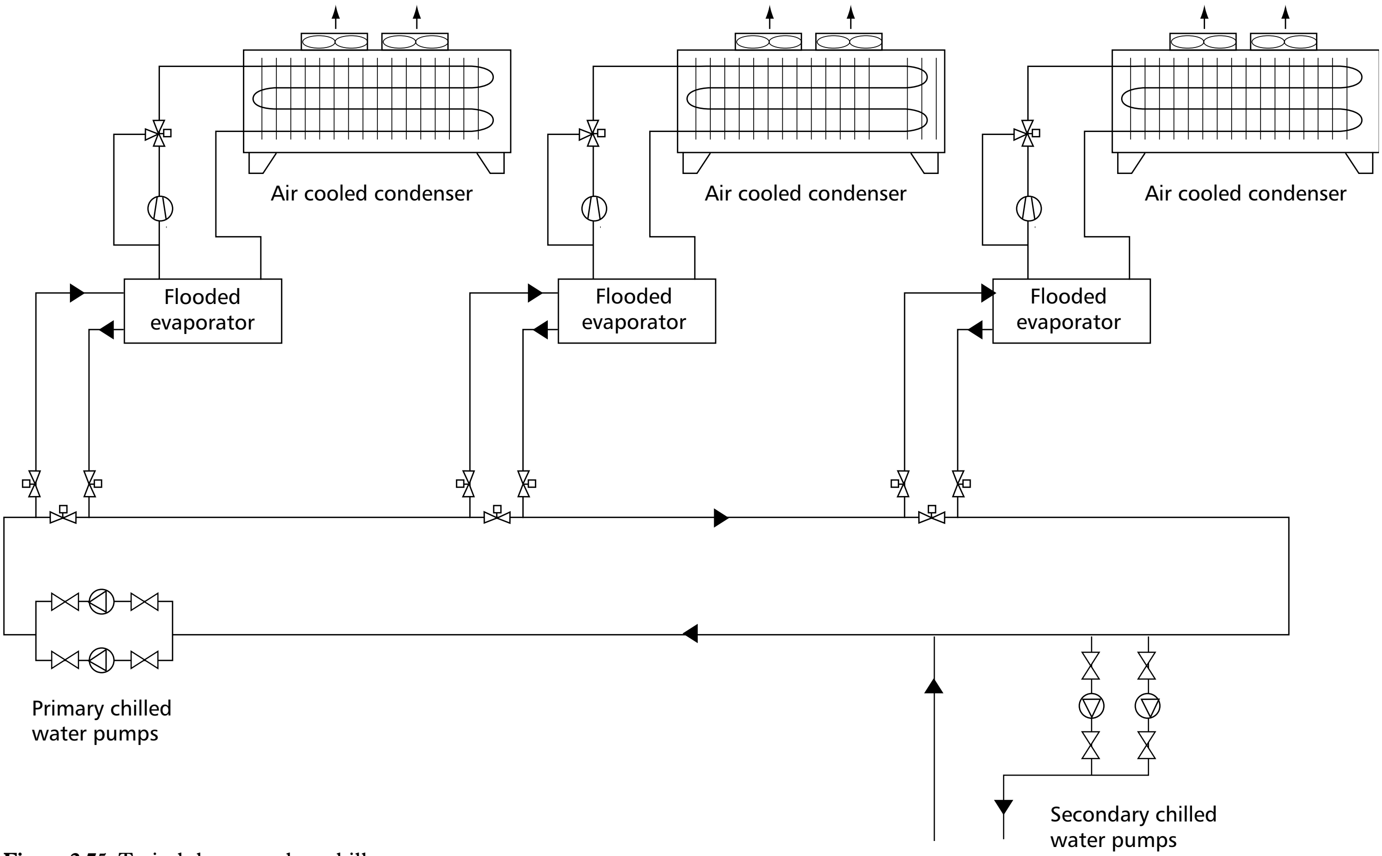


Figure 3.75 Typical thermosyphon chiller system

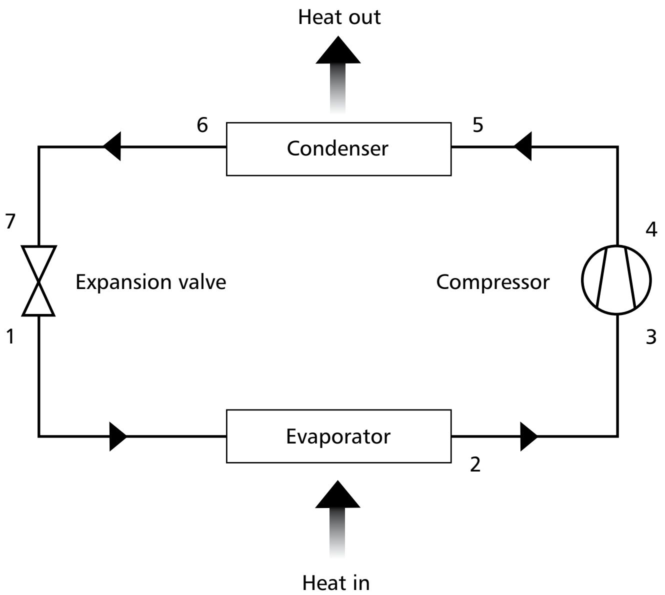


Figure 3.76 Vapour-compression cycle: principal system components

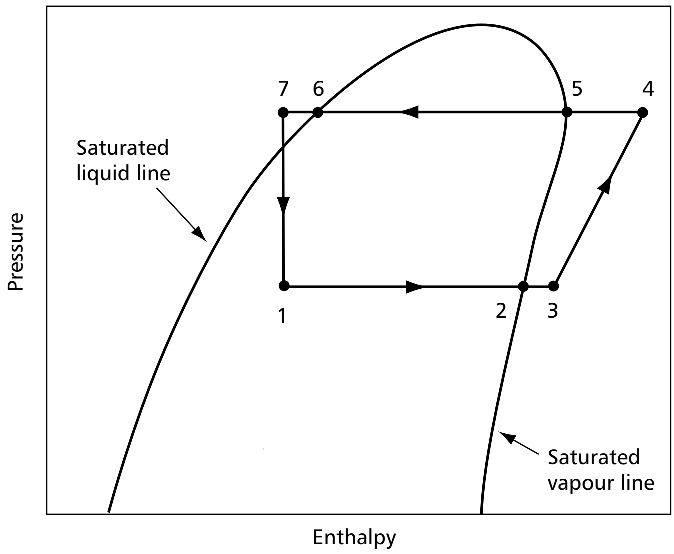


Figure 3.77 Vapour-compression cycle: simple pressure–enthalpy diagram

— *Stage 6 to 7*: further cooling may take place to sub-cool the liquid before it enters the expansion valve (this may occur in the condenser, a second heat exchanger or in the pipework connecting the condenser with the expansion valve).

— *Stage 7 to 1*: the high-pressure sub-cooled liquid passes through an expansion device causing a reduction in its temperature and pressure at constant enthalpy.

Where a vapour compression system is intended to provide useful heating (from the heat rejected at the condenser) it is usually known as a heat pump. Heat pumps may also provide cooling. Figure 3.35 (page 3-35) shows a reversible (refrigerant changeover) air-to-air heat pump that may be used for either heating or cooling. Water-to-water heat pumps may employ a water changeover valve arrangement instead.

Vapour compression plant may use one of a variety of compressor types depending partly on the cooling capacity required. Information on compressors and other system components is given in section 3.3.6.

The designer or specifier should take into account a number of factors when specifying a mechanical vapour-compression plant, including:

— refrigerant type

— refrigerant charge
— safety requirements
— environmental requirements
— cooling efficiency
— heating efficiency (heat pumps)
— seasonal efficiency.

3.3.3.4 Refrigerants

3.3.3.4.1 Refrigerant selection

A wide range of fluids may be used as refrigerant in vapour-compression systems. The basic requirements are that the fluid evaporates at the required cooling temperature at a reasonable pressure and condenses at the temperature of a readily available cooling medium. However, commercial vapour-compression refrigeration systems should be safe, practical and economic and generally the refrigerant should possess as many of the properties listed in Table 3.23 as possible.

No single refrigerant satisfies all of the desired refrigerant properties and a range of refrigerants are commercially available for standard refrigeration and air conditioning applications. Some key properties of these refrigerants are summarised in Appendix 3.A3 and pressure–enthalpy charts are shown in Appendix 3.A4. The designer should be aware of the key properties of different refrigerants and clarify with the equipment supplier or manufacturer the most suitable refrigerant selection for a particular application taking into account the following factors:

— *Equipment size*: capacity required per machine, since the capacity of a compressor is affected by the type of refrigerant.

— *Type of compressor*: reciprocating, centrifugal, screw or scroll.

— *Operating temperature range*: air conditioning, process cooling.

— *Economics*: running costs, first cost of the equipment and refrigerant, cost of servicing, refrigerant handling requirements and eventual cost of refrigerant disposal.

— *Environmental and safety factors*: acceptability to the client (for example in relation to environmental policies and green labelling and certification), future refrigerant availability (Government environmental regulation), safety requirements of relevant codes and standards (see section 3.3.2.4).

The following refrigerant types are in common use in vapour compression refrigeration systems (although not all are available for use in new plant):

Table 3.23 Key refrigerant selection criteria

Refrigerant property	Selection criteria
Low toxicity	A desirable property especially for systems that may be installed in occupied parts of buildings, such as split and multi-split air conditioners. See section 3.3.3.8
Zero ozone-depletion potential	Ozone-depleting substances are no longer acceptable as refrigerants. See section 3.3.2.4.
Low global-warming potential	Substances with high global-warming potential are likely to be restricted or phased out by some governments, and some corporate environmental policies already restrict the use of refrigerants with high global-warming potential. See section 3.3.2.4
Non-flammable	A desirable property, especially for systems that contain large quantities of refrigerant and are located in occupied parts of buildings. See section 3.3.2.4.
Chemically stable and compatible with conventional materials and compressor lubricants	Substances without these properties are unlikely to be used as commercial refrigerants.
Suitable pressure/temperature	Excessively high operating pressures require the use of strong components, pipework relationship and heat exchangers, which increases the cost and weight of systems and increases the likelihood and rate of leakage. Operating pressures below atmospheric pressure increase the risk of contamination and ingress of air. High-pressure ratios reduce compressor and system energy efficiency.
High latent heat	This determines the mass flow of refrigerant that has to be circulated and, although it is a highly desirable property, it can be offset by other properties.
High critical temperature	The critical temperature (the temperature above which a substance behaves like a permanent gas and cannot be liquefied) should normally be well above the required heat rejection temperature. Also, as the critical temperature is approached, the latent heat of vapourisation decreases, which tends to reduce the efficiency of the system.
Low vapour-specific heat ratio	The specific heat ratio determines the index of compression and hence the temperature rise during compression. Low indexes of compression give low discharge temperatures, which are desirable to minimise breakdown of refrigerant and lubricant.
Low temperature glide*	This can either improve or reduce heat transfer coefficients in the evaporator and (for blends) condenser, depending on their design. High glide can cause handling difficulties and cause preferential leakage or fractionalisation (increasing the cost of maintenance). Refrigerants with a high glide are generally unsuitable for use in flooded evaporators due to large concentration changes in the evaporator leading to reductions in performance. High glide causes a reduction in the refrigerant temperature at the evaporator inlet for a given chilled-water supply temperature. This may cause a risk of icing in systems supplying water at 6 °C or below and may require an anti-freeze additive.

* Non-azeotropic blends exhibit a change ('glide') in temperature during the evaporation and condensation process. The temperature of the evaporating refrigerant rises along the evaporator, and the temperature of the condensing vapour decreases along the condenser. The extent of the temperature glide is mainly dependent on the boiling points and proportions of the individual constituents.

— Fluorinated refrigerants:

- *Hydrofluorocarbons* (HFCs): contain only hydrogen, fluorine and carbon. Widely used in commercial and industrial systems but future limits will apply (European F-Gas regulation).

- *Hydrochlorofluorocarbons* (HCFCs): contain only hydrogen, chlorine, fluorine and carbon. The use of virgin refrigerants of this type has been banned since 2010 and recycled or reclaimed refrigerant since 2015 (European ozone depletion regulations).

- *Chlorofluorocarbons* (CFCs): contain chlorine, fluorine and carbon. These have been banned since the 1990s.

- *Hydrofluoroolefins* (HFOs): contain fluorine and carbon. They have a low GWP and limited flammability. These refrigerants are being developed as blends for replacing HFCs.

— Natural refrigerants:

- *Ammonia* (NH_3): contains nitrogen and hydrogen. Mainly used in large outdoor systems. Its use is limited by its toxicity and flammability.

- *Hydrocarbons* (HCs): contain hydrogen and carbon. Mainly used in small, self-contained systems. Its use is limited by its toxicity and flammability.

- *Carbon dioxide* (CO_2): contains carbon and oxygen at very high pressures. Systems have been developed or are under development

These alternative refrigerants are considered in detail below.

(*a*) *Hydrofluorocarbons* (HFCs)

HFCs have many of the desirable refrigerant properties above and are widely used in new systems and as replacements for CFCs and HCFCs and in some existing systems, on account of their good thermodynamic properties and zero ozone-depletion potentials. R134a is a close match to the previously used CFC R12, but for air conditioning systems that would previously have used R22, blended products are widely used, the most common of which is R407C.

The R400 series denotes a blended product. Some blends have quite high glide temperatures, which can reduce capacity and efficiency and potentially increase the cost of maintenance.

Some HFCs have global warming potentials similar to CFCs and HCFCs and as a result there is political pressure in Europe to minimise or phase out the use of HFCs. Section 3.3.2.4 gives information on European policy on the use of HFCs. HFCs are now subject to management and control under the F-Gas Regulation (EU, 2014).

HFCs are not soluble or miscible with conventional mineral-based oils. Many systems rely on the solubility and miscibility of the oil and refrigerant for effective oil transport in the system and return of oil to the compressor. Most equipment manufacturers specify synthetic polyolester oil for use with HFC refrigerants. Polyolesters are more hygroscopic than traditional lubricants, which means that greater care is needed in transport, storage and charging to avoid excessive moisture contamination.

The following HFCs are commercially available and used in refrigeration and air conditioning systems.

— R134a (HFC-134a) is a non-flammable pure fluid, similar in refrigeration capacity and operating pressures to R12 (see Appendix 3.A3). It has replaced R12 in many applications including car air conditioning, centrifugal chillers and domestic refrigerators and freezers. Many manufacturers also specify it for large screw-compressor-based chillers. Its operating pressures are relatively low, which makes it attractive for very large chillers (due to the potential materials and weight savings) but this is countered by a relatively low refrigeration capacity compared with R22 and its HFC-based alternatives. This means that compressors and pipework are approximately 30 per cent larger to cater for the higher volumetric flow. This is in part counteracted by improvements in system design.

— R404A is a blend of HFC-125, HFC-143a and HFC-134a. Its use is mainly as a substitute for R502 in commercial refrigeration systems (for example cold stores and supermarket refrigeration systems), because its relatively low discharge temperature makes it a good choice for low temperature systems. However, its relatively low critical temperature makes its use for air conditioning systems less suitable than other HFCs. It also has a relatively high global-warming potential compared with other HFCs (see Appendix 3.A3).

— R407C is a blend of HFC-32, HFC-125 and HFC-134a. Its operating pressures and refrigeration capacity are similar but not identical to those for R22 and it is therefore quite often used as a replacement for R22. However, R407C is a blend with a relatively high temperature glide (around 6 K), which will lead to reduced capacity and efficiency (on a like-for-like basis) and may increase the cost of maintenance and refrigerant handling, and it is not recommended for systems with flooded evaporators.

— R410A is a blend of HFC-32 and HFC-125 but has a low temperature glide (<1 K), which makes refrigerant handling and system maintenance easier than for R407C. R410A operates at relatively high pressures and stronger pipe and other components are required than for R134a and R407C. However, R410A has a relatively high refrigeration capacity, making it very suitable for VRF and heat pump systems where small size pipes can be used with low pressure drop. R410A is less favourable for large systems on account of the stronger compressor casing and pipework compared with other refrigerants. R410A is also less suitable for systems with high condensing temperatures such as is necessary in hot climates.

There are numerous refrigerants that have been developed as drop-in replacements for CFCs and HCFCs. Low GWP HFC blends such as R407F are substitutes for R404A applied as an interim solution. However, most if not all will result in reduced capacity and/or efficiency and will potentially require oil flushing and replacement and some system modifications. Guidance from the refrigerant manufacturer should be sought in advance of using these refrigerants.

(*b*) Hydrochlorofluorocarbons (HCFCs)

Some HCFCs have most of the desirable refrigerant properties above except for their ozone-depletion and global-warming potentials. Although HCFCs have lower ozone-depletion potentials than CFCs, the supply and use of HCFCs has been phased out (see section 3.3.2.4.8). However, HCFCs are expected to continue to be available in non-EU and developing countries for some time. HCFCs such as R22 (HCFC-22) will also continue to be used in many existing air conditioning and refrigeration systems until the products or systems are replaced. The most common HCFC used in air conditioning was R22, and if a system containing R22 requires maintenance, consideration should be given to changing the refrigerant to an HFC. R123 (HCFC-123) has been used in some centrifugal chillers as an interim replacement for R11 (CFC-11) although concerns about its toxicity have limited its use.

(*c*) Chlorofluorocarbons (CFCs)

CFCs have many of the desirable refrigerant properties except for their ozone-depletion and global-warming potentials. Because CFCs have high ozone depletion-potentials, their use has now been phased out except in developing countries (see section 3.3.2.4). However, some existing refrigeration systems may still contain CFCs, particularly R12 (CFC-12) in domestic refrigerators and some heat pumps and R11 (CFC-11) in some large centrifugal chillers. At the end of use of these products or systems it is a legal requirement for the CFCs refrigerant to be recovered and destroyed by an environmentally acceptable technology.

(*d*) Hydrofluoroolefins (HFOs)

HFOs have a short atmospheric life, and consequently have extremely low global warming impact which enables them to be considered as long term refrigerants under the F-Gas regulation. However they are mildly flammable, and expected to fall into a new safety category A2L with the release of the next edition of BS EN 378. The cost is likely to be high at first but should stabilise with more production. R1234yf was initially developed as a replacement for R134a in automobile or mobile air conditioning (MAC) systems. With similar pressure characteristics both R1234yf and R1234ze are suitable alternatives to R134a and are being applied in centrifugal and screw chillers. Blended with HFCs in such a way as to restrict the GWP to comply with the F-Gas limits, new refrigerant mixtures suitable for long term HFC replacements in other applications are becoming available.

(*e*) Ammonia

Ammonia, R717, is an effective, tried-and-tested natural refrigerant without any harmful ozone depletion or global warming effects. However, ammonia is highly toxic and mildly flammable, which means that special safety precautions are necessary (see section 3.3.2.4.2 'UK health and safety legislation'). Ammonia is widely used in industrial refrigeration systems and is increasingly being used in chillers for air conditioning in the USA and Europe including the UK. It is not suitable for use in direct systems (for example split and multi-split air conditioners) due to the risk of leakage into the occupied space and its non-compatibility with standard copper pipes and copper-based alloys. Ammonia chillers generally use steel or stainless steel pipework and valves.

(*f*) Hydrocarbons (HCs)

HCs such as propane (R290) and isobutene (R600a) are natural refrigerants without any harmful ozone-depletion or global-warming effects and with good refrigeration properties. They are compatible with standard materials and components. However, because HCs are highly flammable, specific safety precautions are necessary (see section 3.3.3.4.2 'Refrigerant safety'). Generally, systems with small refrigerant charges are most suitable for HC refrigerants. Large-capacity packaged products sited in well ventilated installations such as rooftops with restricted access are acceptable. Many European domestic refrigerator and freezer manufactures have switched entirely from R12 or R134a to R600a (isobutane). HCs are also increasingly used in small commercial display cabinets and vending machines.

(*g*) Carbon dioxide

There have been some significant development in the use of carbon dioxide, R744, as a refrigerant over the past 10 years.

Applications are divided into:

(1) direct refrigerant applications

(2) volatile secondary pumping applications.

Direct refrigerant systems are further divided into:

— 1(a) sub-critical: where the condensing medium can achieve condensing temperatures below the critical point 31.3 °C

— 1(b) transcritical: where the condensing medium cannot achieve condensing temperatures below the critical point 31.3 °C, where a gas cooler is required.

System type 1(a) can act as a stand-alone system.

System types 1(b) and 2 require a means to condense the carbon dioxide vapour. This is often by the second stage of a cascade system, where the evaporator acts as a carbon dioxide condenser or water cooled when connected to a chilled water or glycol system.

System types 1(a) and 1(b) have been successfully applied to supermarket and retail distribution centre cooling. System type 1(b) and 2 have been employed in specialist data-centre-cooling and in desk-cooling systems.

3.3.3.4.2 Refrigerant safety

The designer has a responsibility to ensure that the design of the refrigeration system takes into account all the necessary provisions for safe installation, commissioning operation, maintenance and decommissioning of the system. The requirements of relevant UK health and safety related legislation are outlined in section 3.3.2.4. The selection of the system type and refrigerant should minimise hazards to persons, property and the environment. Specific requirements are detailed in BS EN 378 (BSI, 2008a–d), and theses are amplified in the Institute of Refrigeration Safety Codes (IoR, 2009c/d, 2013, 2015a). Specific requirements for commissioning systems are given in CIBSE Commissioning Code R: *Refrigeration systems* (CIBSE, 2002).

The purpose of BS EN 378 (BSI, 2008a–d) is to minimise possible hazards to persons, property and the environment from refrigerating systems and refrigerants. It stipulates specific requirements for different refrigerants, types of refrigeration systems, locations and type of building.

BS EN 378 classifies refrigerants into groups according to their influence on health and safety and these groupings are maintained in the Institute of Refrigeration safety codes (IoR, 2009c/d, 2013, 2015a). Building occupancies are classified according to the safety of the occupants, who may be directly affected in case of abnormal operation of the refrigerating system (such as catastrophic leakage). Refrigerating systems are classified according to the method of cooling the space and heat rejection and broadly follows the classification given below.

(*a*) Occupancy categories

Building occupancy categories are classified as:

— *General occupancy, A*: rooms, parts of buildings, or buildings where:

- people may sleep
- people are restricted in their movement
- an uncontrolled number of people are present or to which any person has access without being personally acquainted with the necessary safety precautions.

Examples of these are hospitals, courts or prisons, theatres, supermarkets, schools, lecture halls, public transport termini, hotels, dwellings, restaurants.

— *Supervised occupancy, B*: rooms or parts of buildings where only a limited number of people may be assembled, some being necessarily acquainted with the general safety precautions of the establishment. Examples of these are business or professional offices, laboratories and places for general manufacturing and where people work.

— *Authorised occupancy, C*: rooms, parts of buildings, buildings where only authorised persons have access, who are acquainted with the general and special safety precautions of the establishment and where manufacturing, processing or storage of material or products takes place. Examples of these are manufacturing facilities, e.g. for chemicals, food, beverage, ice cream, refineries, cold stores, dairies, abattoirs, non-public areas in supermarkets.

Where there is the possibility of more than one category of occupancy, the more stringent requirements apply. If the occupancies are isolated, e.g. by sealed partitions, floors and ceilings then the requirements of the individual category apply.

(*b*) Refrigerant safety groups

Refrigerant safety groups are classified according to their flammability and toxicity (see Table 3.24). Detailed information on derivation of the limits is given in BS EN 378-1: 2008 + A2: 2012 (BSI, 2008/2012).

Table 3.24 Refrigerant safety group summary

Flammability	Toxicity	
	Lower	Higher
No flame propagation	A1	B1
Lower flammability	A2	B2
Higher flammability	A3	B3

Note: introduction of a new A2L Low flammability category is anticipated. The latest version of BS EN 378 should be consulted.

(*c*) Maximum charge of refrigerant

BS EN 378-1 (BSI, 2008a) limits the hazards from refrigerants by stipulating the maximum charge of refrigerant for given occupancy categories and refrigerant safety groups. Table 3.25 summarises maximum refrigerant charge and other restrictions for chillers (indirect closed systems) and direct (DX) systems. The latest version of BS EN 378 should always be consulted for full details. Maximum refrigerant charge is related to the 'practical limit' or maximum allowable short-term refrigerant concentration should the entire charge be released into the space or the room occupied by the system (does not apply to systems located out of doors). Table 3.26 shows the practical limit for a range of common refrigerants and the corresponding maximum refrigerant charge.

(*d*) Specific requirements for ammonia

Ammonia is highly toxic through direct contact and inhalation of concentrations above 1000 ppm. However, it has a highly pungent smell (detectable by nose down to about 10 ppm), which makes voluntary exposure highly unlikely. At these lower concentrations, ammonia has no known long-term or accumulative health effects, although its pungency can induce panic and alarm. Ammonia is flammable at concentrations between 16 per cent and 27 per cent by volume, although in practice such high concentrations are only likely in the event of the most flagrant contravention of safety guidelines.

It is essential that relevant health and safety regulations, safety standards and codes and other industry guidance documents are complied with. These include:

— relevant UK regulations (see section 3.3.2.4.2 'UK health and safety legislation')

— BS EN 378 (most recent edition)

— *Ammonia Refrigeration Systems Code of Practice* (Institute of Refrigeration, 2009d)

— Institute of Refrigeration Technical Bulletin TB35: *The Basics of Ammonia Refrigeration* (IoR, 2012)

— Dangerous Substances and Explosive Atmospheres Regulations 2002 (TSO, 2002c).

In addition to meeting the specific requirements of these standards and codes, it is highly recommended that the following additional guidelines are followed:

— Ammonia systems installed for air conditioning in buildings occupied by humans should be installed either in a special plant room within the building or inside a special enclosure, which may be on the building roof. The advantages of such a plant room or enclosure are that any spilled or leaked liquid ammonia can be contained and that external gas discharge rates can be controlled.

Table 3.25 Summary of BS EN 378-1: 2008 + A2: 2012 (BSI, 2008/2012) requirements for maximum refrigerant charge (the most recent version of BS EN 378 should be consulted for full details)

Refrigerant safety group	System type	Occupancy category A		Occupancy category B		Occupancy category C	
		Max. charge (per system)	Restrictions	Max. charge (per system)	Restrictions	Max. charge (per system)	Restrictions
A1	DX	Practical limit × room volume	No restriction if all refrigerant-containing parts are in open air or unoccupied machinery room	No limit	If located below ground or on an upper floor without emergency exits then maximum charge is practical limit × room volume	No limit	If located below ground or on an upper floor without emergency exits then maximum charge is practical limit × room volume
	Chillers	No limit	Restrictions apply unless in open air or unoccupied machinery room	No limit	Restrictions apply unless in open air air or unoccupied machinery room	No limit	Restrictions apply unless in open air air or unoccupied machinery room
B2	DX	Practical limit × room volume (2.5 kg limit)	Must be a sealed absorption system unless all refrigerant-containing parts are in open air or unoccupied machinery room	10 kg	25 kg if compressor and receiver are in open air or unoccupied machinery room. No restriction if machinery room has no direct communication to occupied space.	10 kg	If number of people is < 1 per 10 m^2 and are there emergency exits, then 50 kg. If compressor and receiver are in open air or unoccupied space, then 25 kg or no restriction if < 1 person/10 m^2. If all refrigerant-containing parts are in unoccupied space or open air then no restriction.
	Chillers	No limit	Must be in open air or unoccupied machinery room† with an exit to open air and no direct access to category A or B. Limits apply if any refrigerant-containing parts are in occupied areas.	No limit	Must be in open air or unoccupied machinery room with no direct access to occupied rooms	No limit	Limits apply if any refrigerant-containing parts are in occupied areas
A3	DX	Practical limit × room volume (1.5 kg limit)	Restrictions apply; refer to BS EN 378	Practical limit × room volume (2.5 kg limit)	Restrictions apply; refer to BS EN 378	10 kg	Restrictions apply; refer to BS EN 378
	Chillers	5 kg	Must be in open air or unoccupied machinery room and additional restrictions apply; refer to BS EN 378	10 kg	Must be in open air unoccupied machinery room and additional restrictions apply; refer to BS EN 378	No limit	Must be in open air unoccupied machinery room and additional restrictions apply; refer to BS EN 378

— The total quantity of ammonia used should be minimised through appropriate design such as the use of multiple chillers for large systems and the use of compact heat exchangers such as plate heat exchangers.

— Relief valves should discharge in a safe place away from any building, such as a vertical pipe on the building roof.

— An acceptable way of disposing of large quantities of spilt or leaked ammonia is through controlled atmospheric dispersion, ideally through a high fan-assisted stack away from people and other buildings.

— The practice of spraying water onto pools of liquid ammonia is highly hazardous and should be avoided.

Table 3.26 Maximum refrigerant charge derived from practical limits

Refrigerant	Practical limit (/ $kg \cdot m^{-3}$)	Maximum refrigerant charge (/ kg) for a direct system serving or installed in a 100 m^3 room
R12	0.50	50
R22	0.30	30
R123	0.10	10
R134a	0.25	25
R407C	0.31	31
R410A	0.44	44
R717 (ammonia)	0.00035	0.035
R290 (propane)	0.008	0.8
R600a (isobutane)	0.008	0.8

Further information on minimising ammonia hazards, including predicting gas concentrations in dispersing ammonia plumes is given in BRE Information Paper IP18/00: *Ammonia Refrigerant in Buildings: Minimising the Hazards* (Butler and Hall, 2000).

(*e*) Specific requirements for HCs

HC refrigerants are highly flammable. However, they have very good refrigeration properties and may be used in a wide range of refrigeration equipment.

It is essential that relevant health and safety regulations, safety standards and codes and other industry guidance documents are complied with. These include:

— relevant UK regulations (see section 3.3.2.4.2 'UK health and safety legislation')

— BS EN 378 (BSI, 2008a–d)

— *A2 & A3 Refrigerants (flammable including hydrocarbons and HFOs) Code of Practice* (IoR, 2009c)

— *Guidelines for the Use of Hydrocarbon Refrigerants in Static Refrigeration and Air Conditioning Systems* (ACRIB, 2001).

The above standards and codes are designed to minimise hazards associated with the use of flammable HC refrigerants and bring the degrees of risk in line with other types of refrigerant. It is also essential that safe working practices are adhered to during maintenance and servicing, as refrigerant grades of HCs are odourless. In particular, the following precautions are essential during maintenance and servicing.

— HC vapour detectors should be used during maintenance and servicing to alert the technician to the presence of potentially flammable atmospheres.

— No source of ignition should be present and special precautions should be taken prior to any brazing or welding.

— Any person who is involved with working on or breaking into a refrigeration circuit should hold a current certificate from an industry-accredited authority that certifies their competence to handle refrigerants (including HCs) safely in accordance with an industry-recognised assessment specification.

(*f*) Refrigerant detection

Refrigerant detectors and alarms are required in all refrigeration equipment plant rooms to prevent the exposure of workers to refrigerant concentrations higher than the HSE occupational exposure limits (see *EH40/2005: Workplace Exposure Limits* (HSE, 2005)) and to warn of higher toxic concentrations. Detectors are also required in plant rooms that contain HC or ammonia systems to start emergency ventilation and shut down any electrical equipment that is not suitable for operation in explosive atmospheres. Refrigerant detectors can also be used as a means of detecting refrigerant leaks although their effectiveness is affected by how well the plant room is ventilated. Refrigerant detectors are unlikely to be effective for detecting leaks from equipment installed outside.

The location of refrigerant detectors should take account of the density of the refrigerant. HCFCs, HFCs and HCs are heavier than air, and refrigerant detectors should therefore be located at low level. Ammonia vapour is lighter than air and detectors are therefore usually located above the refrigeration equipment.

Common types of detector include semiconductor sensors and infrared analysers. Electrochemical sensors are also used for ammonia detection. Semiconductor and electrochemical sensors are sensitive to other gases, including some cleaning chemicals, whilst infrared systems are more selective but are also more expensive. Detectors may be discrete single-point devices or aspirated systems that may have several sensing points connected to the sensor by air-sampling tubes and a small air pump. All types of detector require periodic recalibration and electrochemical sensors have a short lifetime.

3.3.3.4.3 Environmental impacts

The selection of a refrigerant and refrigeration system type should take account of the environmental impact of the refrigerant and the systems and legal requirements. These are mainly related to the leakage of refrigerant to the atmosphere and the end-of-life disposal of refrigerants and compressor oil. It is a legal requirement that all used refrigerant and oils are recovered for recycling or disposal using an environmentally acceptable method. Legal and other environmental-related requirements are detailed in section 3.3.2.4.

The selection of a refrigeration system and refrigerant should aim to minimise the TEWI of the system over its expected lifetime. TEWI takes account of the direct global warming potential of the refrigerant and the global warming impact of the energy used to drive the system.

3.3.3.4.4 Refrigerant leakage

Refrigerant leakage is the unwanted loss of refrigerant from a system and may, for example, be caused by defective gaskets, seals, joints, pipes and hoses. Refrigerant leakage may be gradual, which can be continuous or intermittent, or catastrophic due to sudden component failure or accident. Refrigerant losses may also occur during servicing and maintenance, and may be deliberate, accidental or unavoidable.

The consequences of refrigerant leakage include the following.

— *Environmental impact*: CFCs and HCFCs damage the ozone layer and most refrigerants contribute to global warming.

— *Higher running costs and reduced cooling performance:* refrigerant loss will cause a gradual reduction in refrigeration capacity and coefficient of performance system (COSP) (see section 3.3.2.3).

— *Health and safety hazards*: some refrigerants are toxic and/or flammable and many others are toxic at high concentrations.

— *Higher servicing costs*: F-Gas requirements require locating and repairing leaks and replenishing leaked refrigerant as well as keeping refrigerant records and undertaking follow-up leak tests.

— *Unreliability*: loss of refrigerant may cause refrigeration system compressors to operate at high suction superheats resulting in low suction pressure and causing system trips. This can also cause reduced lubrication and cooling of compressors and motors, reducing life expectancy.

— *Legal requirements*: knowingly allowing CFC, HCFC or HFC refrigerant to leak to atmosphere is an offence that could result in a large fine.

Leakage will cause a gradual deterioration in the system energy efficiency and cooling performance. Some systems are buffered, that is they hold a significant reserve of refrigerant and will show a delayed effect before the effects of leakage becomes apparent. Figure 3.78 shows the effect of refrigerant leakage on the CoP of one example unbuffered chiller and Figure 3.79 shows the effect on refrigeration capacity.

Refrigerant leakage can be minimised through good design and working practices. In particular, it is recommended that the following guidelines are followed.

— *Minimise the use of refrigeration*: it is important that a genuine requirement for mechanical refrigeration has been established by properly evaluating end-user requirements (section 3.3.1.3) and that options for free cooling have been considered (see section 3.3.3.2).

— *Minimise the quantity of refrigerant*: the quantity of refrigerant used should be minimised by selecting equipment with a low specific refrigerant charge (kg of refrigerant per kW nominal refrigeration capacity).

— *Use small systems in combination*: using several self-contained refrigeration systems with a lower individual refrigerant charge, rather than a single large system, reduces the total amount of refrigerant that can be released from a single leak. Compact refrigeration equipment generally has a lower charge than larger or extensive distributed systems (such as multi-split or variable refrigerant flow (VRF) systems).

— *Avoid large reserves*: avoid unnecessarily large reserves of refrigerant in 'buffered' systems, for example in liquid receivers, pumped recirculation systems or oversized liquid lines.

— *Treat any pipework vibration*: whether experienced during normal running or compressor start-up.

— *Minimise the number of joints, seals and valves*: brazed or welded joints are much more likely to be leak tight than demountable joints. Valves are a potential source of leaks although a sufficient number should be provided to minimise refrigerant loss during servicing and maintenance.

— *Select a refrigerant to minimise the impact of leakage*: different refrigerants have differing impacts on ozone depletion, global warming and health and safety. For example, ammonia has a zero global-warming potential and HCs have very low global warming potentials, whereas HCFCs and HFCs generally have high global-warming potentials (see Appendix 3.A3).

Further guidance is given by the *Code of Practice for the Minimisation of Refrigerant Emissions from Refrigerating Systems* (IoR, 2009b), BRE Information Paper IP1/94: *Minimising refrigerant emissions from air conditioning systems in buildings* (Butler, 1994) and the Institute of Refrigeration's 'REAL Zero' guidance (IoR, 2015b).

3.3.3.5 Direct expansion (DX) systems

3.3.3.5.1 General

The term 'direct expansion' describes an evaporator in which all the refrigerant supplied completely evaporates, producing superheated vapour at the exit. However, the term is also commonly used to describe a direct refrigeration or air conditioning system. Direct systems usually use a direct expansion evaporator but so do many indirect systems including water chillers. For the purposes of this Guide, the term 'direct expansion' (DX) covers single-room units, multi-split, ducted and variable refrigerant flow (VRF) systems. Direct expansion cooling may also be used for close control applications.

In general, direct DX systems are thermodynamically more efficient (i.e. they have a higher CoP) than indirect systems because they directly cool the substance or space being cooled without the use of an intermediate coolant and additional heat exchangers. However, in practice, other factors such as the surface area of evaporators and condensers, compressor and fan efficiencies can significantly affect efficiency and large, well-engineered chillers can be more efficient than some direct systems.

(*a*) Through-the-wall DX units

Also known as window-mounted units, these are packaged fan coil units (FCUs) that incorporate a room-air side DX evaporator and outside facing condenser coil, with all the

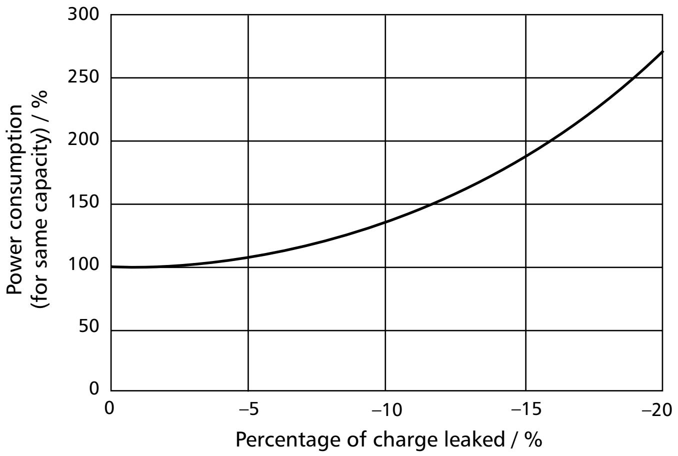


Figure 3.78 Effect of refrigerant leakage on COP: unbuffered chiller

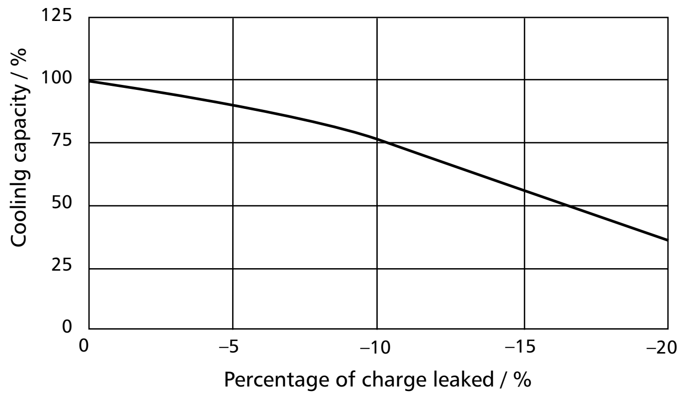


Figure 3.79 Effect of refrigerant leakage on refrigeration performance (cooling capacity): unbuffered chiller

other components including fans, filters, compressor and an expansion device in the same casing. The unit is installed in a wall or window with the unit protruding outside. The units are intended for single rooms and have built-in, self-contained controls. Many units can operate in a heat pump mode to provide heating as well as cooling.

Through-the-wall units are less suitable for large buildings due to the high maintenance overhead of many individual units, potential control difficulties and limited air-throw into deep rooms. Because of the need to keep the units compact, the evaporator and condenser are often quite small. This requires low off-coil air temperatures on the room side when in cooling mode, which can cause nuisance cold draughts and excessive dehumidification.

(*b*) Single-split DX units

These are similar to through-the-wall units except that the indoor and outdoor units are separate and connected by refrigerant pipes, which avoids the need for a large hole to be cut in the wall or window. The outdoor unit contains the compressor and condenser coil and can be roof or wall mounted. Indoor units offer considerable flexibility and may be wall or ceiling mounted. Some units are supplied with pre-charged flexible refrigerant pipes, which simplifies installation for situations where the indoor and outdoor units are very close together. Where the separation distance is greater, rigid pipework has to be installed onsite and particular care is required to prevent internal contamination or poor pipe layout if reliability problems are to be avoided (see section 3.3.6.8). The maximum pipe length is often around 100 m.

(*c*) Multi-split DX units

These are similar to single-split units except that many indoor units can be connected to a single outdoor unit. They are very similar to VRF units (see section 3.3.3.8) and most of the comments for VRF units apply to multi-split units.

(*d*) Ducted DX units

DX evaporator coils may be mounted in a duct or a central station air handling unit (AHU) for small commercial or residential buildings. The coils use copper tubes with aluminium fins for improved heat transfer. AHU systems typically provide up to 100 kW of cooling. As with multi-split DX systems, the requirements of BS EN 378 (BSI, 2008a–d) should be complied with, in particular with respect to the maximum refrigerant charge of the system. The maximum refrigerant charge is related to the practical limit of the refrigerant used and the volume of the smallest room served by the system.

3.3.3.6 Indirect systems

Secondary coolants (sometimes also known as heat transfer fluids, brines or secondary refrigerants), such as chilled water, brine or glycol mixes, are generally used on larger indirect systems where the volume of the primary refrigerant would be too large for environmental and/or cost reasons. The use of a secondary coolant involves an additional heat transfer process and therefore greater temperature difference, hence these systems are inherently less energy efficient than direct refrigeration systems.

Secondary coolants should ideally be non-toxic liquid with a high thermal conductivity, a high specific heat capacity and a low viscosity. For good heat transfer it is also desirable that the coolant velocity is high enough for turbulent flow. Table 3.27 below summarises some of the key properties of a range of common secondary refrigerants. Water has good heat transfer and transport properties and is the most widely used secondary coolant for applications above 0 °C, especially for air conditioning systems. For low temperature applications, calcium chloride, sodium chloride, ethylene glycol or propylene glycol are commonly used. Most of these substances require corrosion inhibitors to prevent damage to metal pipes and other components. Propylene glycol is often used in food industries on account of its low toxicity, despite its high viscosity, which results in relatively poor energy efficiency compared with other potential secondary coolants.

3.3.3.6.1 Two-phase secondary coolants

The performance of secondary systems can be improved by using two-phase secondary coolants.

Systems using carbon dioxide as a volatile secondary refrigerant are currently employed for large-scale industrial cooling systems, server room cooling systems and in supermarket retail cooling applications (Pearson, 1993).

The advantages of using carbon dioxide as a volatile secondary refrigerant include improved heat transfer coefficients and high latent cooling effect, which reduces the mass circulated and the size of pipework.

Other forms of two-phase secondary coolants include pumpable ice slurries (Paul, 2001) (not commonly in use in the UK), using ice crystals that have been formed on the surface of a scraped evaporator.

Care should be taken to ensure compatibility between the selected secondary coolants and the system plant and pipework components.

Table 3.27 Properties of common secondary refrigerants

Substance	Concentration by weight / %	Viscosity / centipoise	Freezing point / °C	Flow rate per 100 kW for 5 K temp. rise
Water	100	1.55	0.0	4.8
NaCl	12	1.75	–8.0	5.1
$CaCl_2$	12	2.01	–7.2	5.2
Ethylene glycol	25	2.70	–10.6	5.1
Propylene glycol	30	5.00	–10.6	4.9
Polydimethylsiloxane	100	1.91	–111.0	14.5

3.3.3.7 Absorption refrigeration

3.3.3.7.1 General

The term 'sorption' includes both absorption and adsorption, both being systems where refrigeration is effected by evaporation of a refrigerant, the vapour being absorbed or adsorbed by a liquid absorbent or solid adsorbent medium respectively, from which it is subsequently expelled at a higher partial pressure by heating and then liquefied by cooling. The number of fully developed adsorption systems is, however, relatively small. For further details see IEA Heat Pump Programme Annex 24: *Absorption Machines for Heating and Cooling in Future Energy Systems* (IEA Heat Pump Centre, 2000).

The working principle of an absorption cycle is similar to that of a mechanical vapour-compression system, the only major difference being the replacement of the compressor by a heat operated absorber–generator (see Figures 3.80 and 3.81).

In building services it is common for the heat from a combined heat and power generator to be utilised as the heat source when operating with lithium bromide/water absorption chillers. This is commonly termed combined cooling heating and power (a separate heat sink is required when cooling or hot water is not required).

The COP of absorption systems is generally inferior to that of vapour-compression systems but advantages can accrue where thermal energy (e.g. from a CHP system) is used to power the generator, or where quiet vibration-free operation is a criterion. The COP of vapour-compression machines is measured in terms of cooling to electrical power input whereas, for absorption machines, COP is measured in terms of cooling to thermal energy input. For definitions of COP for vapour-compression and absorption systems, see sections 3.3.2.3 and 3.3.3.7 'Coefficient of performance' respectively.

In air conditioning systems, the absorption process is commonly employed in packaged equipment for the production of chilled water using hot water, gas or steam as the heat energy source. Figures 3.81 and 3.82 (Tozer and James, 1996) show schematics of the two types of absorption chillers currently available.

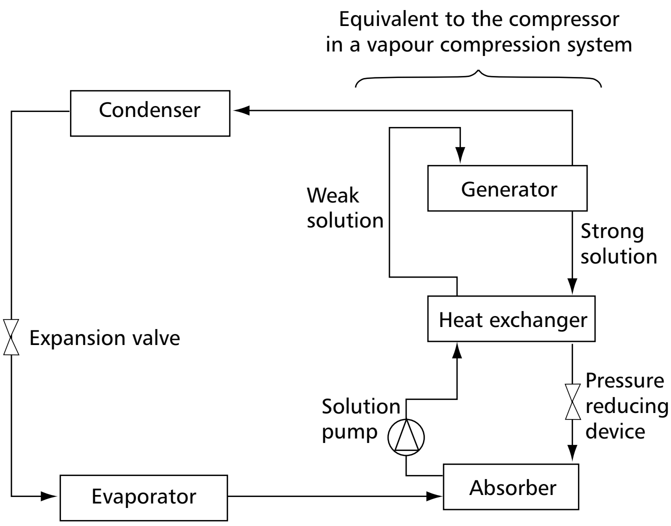


Figure 3.80 Single-effect absorption system (lithium bromide)

The double-effect lithium bromide absorption cycle incorporates two generators. The solution is pumped through the solution heat exchangers to the first effect generator and then the second effect generator. As the high temperature condenser heat is used to drive the low temperature generator, it provides a higher efficiency system.

The stages in the process are as follows.

- In the evaporator the refrigerant extracts heat by evaporation (q_r = heat of evaporation).
- The refrigerant vapour is absorbed and condensed into the solution in the absorber, thereby making the solution weaker ($q_r + q_l$ = heat of evaporation plus solution).
- The weak solution is pumped to high pressure and transferred to the generator.
- The addition of heat ($q_r + q_l$ = heat of evaporation plus solution) in the generator raises the temperature of the solution, separating and evaporating the refrigerant, thus making the solution stronger.
- The strong solution is returned to the absorber through the pressure-reducing device so maintaining the pressure difference between the high and low sides of the system.
- The refrigerant vapour driven out of solution at high pressure in the generator flows to the condenser where it is liquefied by removal of heat at constant pressure (q_r = heat of evaporation).
- The condensed liquid is fed through a pressure-reducing device into the evaporator where the cycle recommences.

The heat generated due to the absorption ($q_r + q_l$) and condensation (q_r) is usually removed by passing cooling water through these vessels in series in water-cooled systems or by air coils in air-cooled systems. Utilisation of waste heat, heat recovery, peak demand shaving and co-

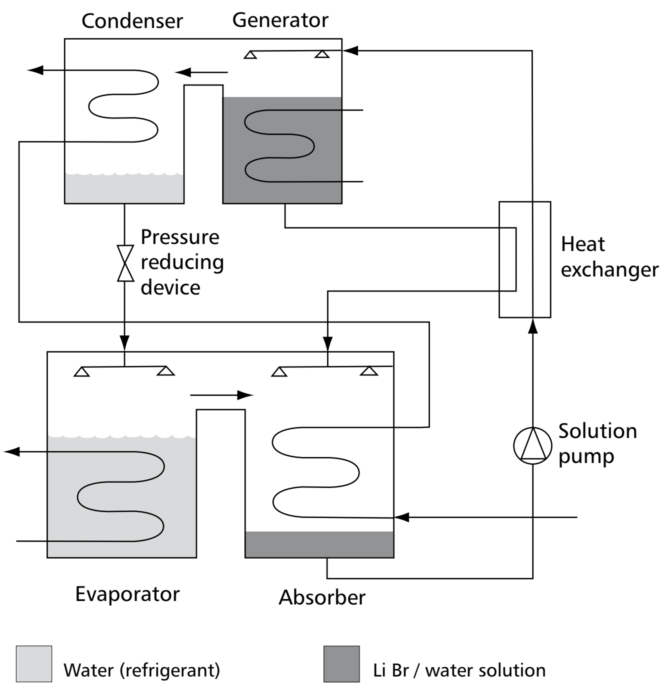


Figure 3.81 Single-effect absorption chiller

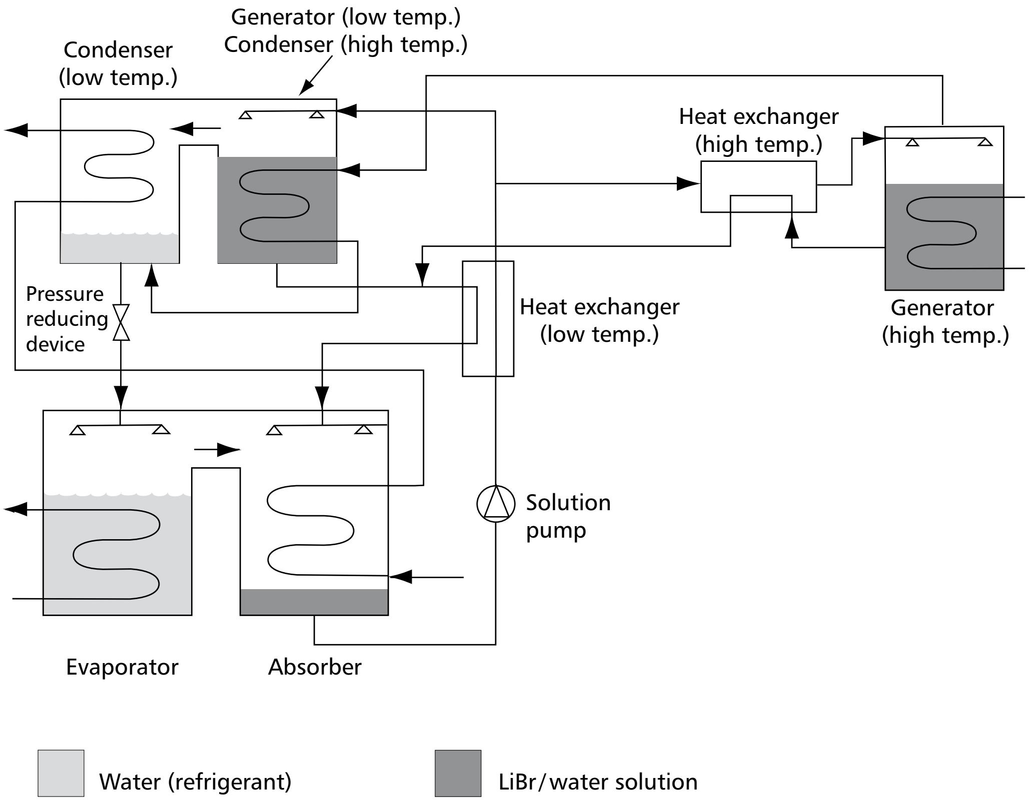


Figure 3.82 Double-effect absorption chiller

Table 3.28 Absorption chiller range (Tozer and James, 1996)

Heat source	Single effect			Double effect		
	Refrigerant	Condenser type	COP	Refrigerant	Condenser type	COP
Direct fired natural gas	Ammonia	Air cooled	0.5	Water	Water cooled	1.0
Hot water (80–130 °C) or steam (0.2–1.0 bar)	Water	Water cooled	0.7	—	—	—
Steam (3–9 bar)	—	—	—	Water	Water cooled	1.2
Engine exhaust gases (280–800°C)	—	—	—	Water	Water cooled	1.1

generation have been the major factors that have influenced the choice of absorption technology in the current market.

The choice between single-effect (Figure 3.81) and double-effect (Figure 3.82) absorption chillers is usually based on the temperature of the driving energy source, as double-effect absorption chillers require a higher temperature heat source than single-effect chillers (see Table 3.28).

Although absorption chillers are much less efficient than vapour compression refrigeration systems, the fact that their only major requirement is heat gives them greater flexibility. The carbon emissions resulting from the operation of an absorption chiller are also not as bad as the COP would appear to indicate because of the lower carbon overhead of gas or heat compared with delivered electricity. An effective application of absorption chillers is in conjunction with combined heat and power (CHP) systems. Absorption chillers can be used with CHP systems in the following ways.

Single-effect chillers:

— hot-water driven (80–130 °C)

— steam driven (1 bar).

Double-effect chillers:

— steam driven (3–9 bar)

— exhaust-gas driven (280–800 °C).

Although single-effect absorption chillers can be driven by low-pressure steam, the most common application is to use engine jacket cooling water. Many CHP systems, such as those commonly used in hospitals and industrial applications, use steam at approximately 8–15 bar and these are very suitable for steam-driven, double-effect absorption chillers designed to use steam at 8 bar. Where reciprocating engines or gas turbines are used, the hot gas exhaust, which is typically at 280–800 °C, can be used directly with hot-gas driven, double-effect absorption chillers. Further guidance on using absorption chillers with CHP, including sizing the chillers, is given by GPG 256: *An introduction to absorption cooling* (EEBBP, 1999a).

Triple-effect absorption chillers are being developed by a number of commercial manufacturers. They use the heat three times in the same way as the double-effect cycle, and gas-fired machines are expected to have CoPs in the order of 1.52.

3.3.3.7.2 GAX absorption chillers

In generator absorber heat exchanger (GAX) absorption chillers part of the thermal load on the generator is met by direct heat transfer from the absorber. Figure 3.83 shows this internal heat transfer process. The degree of internal

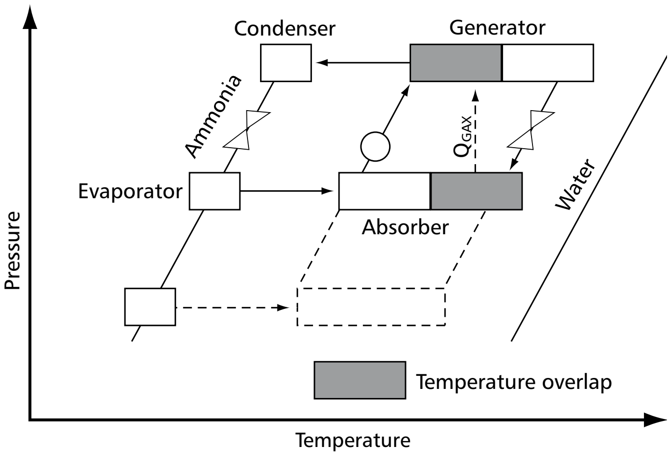


Figure 3.83 Double effect absorption chiller

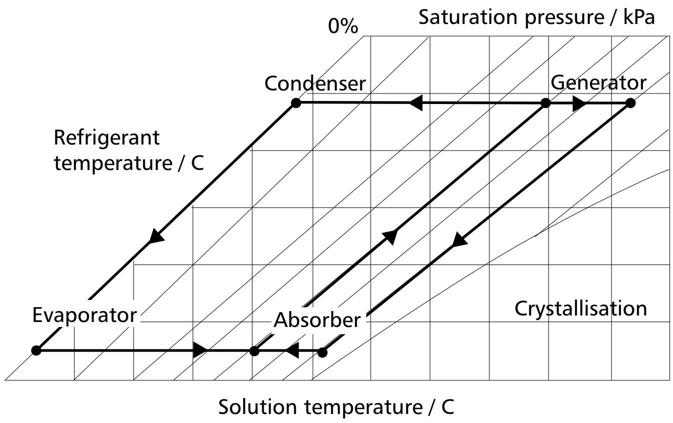


Figure 3.84 Pressure-temperature concentration diagram (PTX) for a water/LiBr system (single effect)

heat transfer or temperature overlap depends on the temperature of the evaporator, which should be as high as possible.

Although CoP improvements in the order of 22 per cent have been shown compared with a conventional single-effect, gas-fired chiller, the additional heat transfer process adds complexity and cost to the chiller and in practice it may be more economic to use a double-effect machine.

3.3.3.7.4 Refrigerant selection

Commercially available absorption chillers generally use water/lithium bromide (LiBr) or ammonia/water solutions depending on the cooled fluid temperature and the required cooling duty:

— for chilled-water temperatures >5 °C, a water/LiBr absorption machine is typically used with a water-cooled condenser, although dry coolers have been used in the UK

— for chilled-water temperatures <5 °C, an ammonia/water machine may be used; this can be either small air-cooled modules or water-cooled machines for industrial refrigeration applications.

In the water/LiBr machine, water is the refrigerant and cooling is based on the evaporation of water at very low pressures. Since water freezes below 0 °C, at this temperature the chilling temperature meets a physical limit. LiBr is soluble in water if the LiBr mass fraction of the mixture is less than about 70 per cent (see Figure 3.84).

Crystallisation of the LiBr solution will occur at higher concentrations, which will block the solution circuit, resulting in a failure of the refrigeration process and possible damage to the plant. This sets a limit for the temperature of the absorber. Poor control of temperature or a fast change of conditions may cause crystallisation, but appropriate controls, such as monitoring by level transducers, will minimise problems (see IEA Heat Pump Programme Annex 24: *Absorption Machines for Heating and Cooling in Future Energy Systems* (IEA Heat Pump Centre, 2000)). In order to supply sufficiently low temperatures to the absorber at high outside air temperatures, evaporative water cooling is usually used.

Corrosion inhibitors are essential in LiBr machines; long life will not be achieved without their use. Additives such as lithium chromate, lithium molybdate and lithium nitrate have been used, but this depends on the manufacturer.

In ammonia/water machines, ammonia is the refrigerant. This offers opportunities to provide refrigeration at temperatures down to –60 °C. Unlike LiBr, which is a salt, water has a vapour pressure of itself. This means that in the generator, besides ammonia vapour, a certain amount of water vapour will also be present. In the evaporator, this will lead to problems because the ammonia will evaporate more easily than the water. This results in an accumulation of water in the evaporator, undermining the chilling process. To prevent this, an extra device known as a rectifier is incorporated in the system to separate the water content from the vapour flow coming from the generator. The rectifier cools the vapour produced in the generator, therefore demanding more heat and reducing the COP. Despite rectification, a small fraction of water will still remain in the vapour. To minimise accumulation in the evaporator, there always has to be a flow of non-evaporated fluid (i.e. liquid) from the evaporator to the absorber.

Ammonia is soluble in water at all operating conditions so crystallisation will not occur. Consequently, at equal chilling temperatures, a higher absorber temperature is possible with water/ammonia compared with water/LiBr chillers. This allows the selection of dry coolers (see section 3.3.4.3) if required. It should be noted however that air cooling is less efficient than evaporative cooling and consequently uses more energy.

3.3.3.7.4 Safety

The designer has a responsibility to ensure that the design of the refrigeration system takes into account all the necessary provisions for safe installation, commissioning operation, maintenance and decommissioning of the system. The requirements of relevant UK health and safety related legislation are outlined in section 3.3.2.4.

Ammonia/water absorption chillers require special consideration as the refrigerant is ammonia and precautions are required in case of leakage. Ammonia detectors and alarms should be installed in chiller plant rooms. Ammonia/water solutions should be treated as special waste when being disposed of.

Skin contact with LiBr should be avoided and operators should wear appropriate protection when charging or decommissioning LiBr/water chillers. LiBr is highly corrosive to steel in the presence of oxygen. Corrosion is minimised by the addition of inhibitors such as lithium nitrate and other additives such as n-octanol. These chemicals are toxic and can cause chemical burns. Full protective clothing should be worn when handling, in accordance with manufacturers' instructions.

3.3.3.7.5 Environmental impacts

Absorption chillers do not use refrigerants or other substances that can cause ozone depletion or contribute to global warming. However, many of the substances are toxic and safe handling and disposal procedures should be complied with in accordance with manufacturers' instructions (see section 3.3.3.7.4 'Safety' above).

3.3.3.7.6 Refrigerant leakage

LiBr/water absorption chillers operate below atmospheric pressure and therefore refrigerant leakage is not an issue. However, the ingress of air is a problem and purge units are normally provided to remove such non-condensables from the system. Ammonia–water systems operate at positive pressure and precautions must therefore be made to prevent leakage. Small leaks are readily detectable by the characteristic pungent smell of ammonia. Detectors and alarms should be provided in plant rooms to prevent exposure to toxic concentrations.

3.3.3.7.7 Coefficient of performance

Table 3.29 gives typical CoPs for some common commercially available absorption chiller types.

The CoP of absorption chillers is considerably lower than for vapour-compression refrigeration systems. This means that absorption chillers have to reject considerably more heat than vapour-compression systems, resulting in a requirement for larger or a greater number of dry air coolers or cooling towers. The heat dissipation ratio (see below) is a useful practical method of comparing heat rejection requirements for competing systems.

3.3.3.7.8 Heat dissipation ratio

The heat dissipation ration (HDR) is a useful index for analysing condenser cooling water requirements. HDR is defined as the ratio of heat dissipated from the condenser and absorber with respect to the evaporator cooling duty:

$$\mathrm{HDR} = \frac{Q_{\mathrm{rej}}}{Q_{\mathrm{cooled}}} = 1 + \frac{1}{\mathrm{CoP}} \tag{3.2}$$

where Q_{rej} is the heat rejected at the condenser (W) and Q_{cooled} is the heat absorbed in the evaporator (refrigerating effect) (W).

Table 3.30 shows the HDR for a range of chillers. For example, a single-effect absorption chiller requires condensers at least 1.9 times larger than those required for a vapour-compression chiller with the same cooling capacity.

3.3.3.8 Variable refrigerant flow (VRF) systems

Variable refrigerant volume (VRV) or variable refrigerant flow (VRF) systems first appeared in the UK in the late 1980s having been introduced from Japan. These systems have become widely used in almost all types building heating and cooling applications. The term VRV or VRF has been adopted to describe the technology.

Table 3.29 Commercially available absorption chiller types (IEA Heat Pump Centre, 2001) and CoPs

Working pair	System	Driven/fired by (not direct)	Cooling capacity / kW	CoP
H_2O/LiBr	Single-effect absorption	Steam, water	40–6000	0.7
		Gas, oil	40–6000	0.6
H_2O/LiBr	Double-effect absorption	Steam	70–6000	1.2
		Gas, oil	5–6000	1.0
NH_3/ H_2O	Single-effect absorption	Gas	10–100	0.5
	Generator absorber heat exchanger (GAX)	Gas	10–100	0.68
H_2O/silica gel	Double-alternating absorption	Water	70–350	≤ 0.6
		Gas	70–350	≤ 0.5

Table 3.30 Heat dissipation ratios for absorption chillers and vapour-compression chillers

Type of chiller	Lithium bromide absorption		Vapour compression	
	Single-effect steam /hot water	Double-effect steam	Double-effect gas fired	Electric reciprocating
CoP	0.7	1.2	1.0	>4
HDR	2.5	1.8	2.0	<1.3
Ratio of absorption chiller HDR to vapour compression HDR	>1.9	>1.4	>1.5	1.0

VRV/VRF systems are complex multi-split systems with a refrigerant flow distributor control device. This device controls the refrigerant with a PID control loop with the inputs being sent and received on a communication system. These systems generally offer higher efficiency than recommended in Building Regulations Part L *Compliance Guide* (NBS, 2013b) by effective use of compressor, heat exchangers and controls. Further energy savings are available by combining heat recovery regimes allowing for rejected heat to be transferred with in a building or stored as hot water. This complexity means that correct system design is paramount.

Manufacturers can allow as many as 64 indoor units to be connected to a several outdoor units although 20–40 units are perhaps more typical. Individual indoor units can provide heating or cooling as required using either a three or two pipe system. Specialised heat recovery systems employ refrigerant distribution units with vapour and fluid separation and flow control. All types of system can provide simultaneous heating and cooling with scope for heat recovery from units providing cooling to units requiring heating. The systems are suitable for larger buildings and offer installation flexibility as the choice of indoor units can be extensive and offer alternatives to individual room units and chilled water based systems. (See section 3.2 for more details on split systems.)

The design and installation of the system, including maximum lengths for interconnecting pipe work, is crucial for reliable operation and it is vital that the manufacturers' instructions are fully complied with. Design criteria should take into consideration reductions in performance due to pipe runs, minimum and maximum ambient conditions. Designers should also pay special attention to the performance of a system in defrost and at lower ambient conditions.

Installation processes should be aligned with BS EN 378 (2008a–d) and must be carried out by a fully qualified F-Gas registered engineer to prevent issues with possible leakage of refrigerant from a system during its operational life. Manufacturers can offer leak detection systems that enable complex leak detection strategies to be applied. However, particular care is needed in the design of VRV/VRF systems serving restricted spaces where people may sleep, such as hotel bedrooms or hospitals as safety limits apply to the volume of refrigerant that could displace oxygen in the event of a significant leak.

3.3.3.9 Evaporative cooling

Evaporative cooling uses the evaporation of water to decrease the dry bulb temperature of air. There are two main categories of evaporative cooling:

— *Direct evaporative cooling*: water is evaporated directly into the supply air stream, adiabatically reducing the air stream's dry bulb temperature but increasing its absolute humidity. Direct coolers may use wetted media or air washers.

— *Indirect evaporative cooling*: two air streams are used. A secondary air stream (either exhaust air or outdoor air) is cooled directly using evaporation. The cooled secondary air stream is then used to cool the primary supply air indirectly through an air-to-air heat exchanger before being exhausted. The supply air is therefore sensibly cooled without increasing its absolute humidity.

Evaporative cooling works best in hot, dry climates, such as Arizona, and is not particularly well suited to the UK's relatively damp climate. Evaporative cooling systems generally require much higher airflow rates than conventional air conditioning systems because of their relatively high air supply temperatures, and therefore require larger fans and ducts. They also cannot deal with latent cooling loads, which can be a major disadvantage in the UK. For these reasons, evaporative cooling is often used in the UK in conjunction with a conventional refrigeration system or with a desiccant stage.

Further information and guidance on evaporative cooling is given section 3.2.

3.3.3.10 Other refrigeration technology

Vapour-compression, absorption-refrigeration and free-cooling systems are the dominant forms of refrigeration used for building refrigeration and air conditioning systems. However, a number of alternative forms of cooling technologies do exist or are being developed. There is considerable interest in these alternatives due to the environmental impact of many current refrigerants, especially those that can cause ozone depletion and/or are greenhouse gases.

3.3.3.10.1 Solid adsorption systems

The solid adsorbent process differs from standard absorption chillers in that the process is essentially intermittent in nature (see Figure 3.85). The process starts (*a*) with an input of high temperature heat to bring about desorption of the working fluid from the adsorbent accompanied by the rejection of heat as the working fluid condenses. When this process is complete the adsorbent is cooled (rejecting a further quantity of heat), which leads to

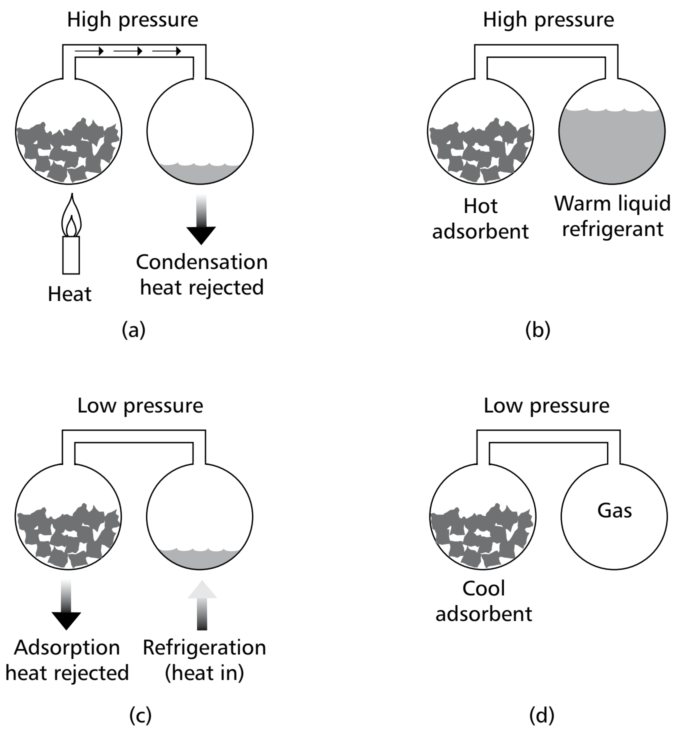


Figure 3.85 Processes within a simple low-efficiency adsorption cycle

the total pressure falling (*b*). This, in turn, causes the working fluid in the right-hand vessel to evaporate, thus producing the desired refrigerating effect (*c*). For continuous cooling, two such systems are necessary operating out of phase with one another.

The COP of this system appears to be rather low due to heat being consumed in heating the bed to its desorbing temperature, followed by rejection as the bed is cooled. Furthermore, the poor thermal diffusivity of porous or packed beds causes the process of heating or cooling to be an extremely slow one, lengthening the cycle time and reducing the power per unit mass of the bed. Solid absorption systems are currently the focus of university research. It is not known whether further development could overcome the many problems.

3.3.10.2 Stirling cycle

The Stirling cycle is based on a closed thermodynamic cycle with regeneration. It is quite often used as a benchmark for assessing other systems. The operation of a reversed Stirling cycle is shown in Figure 3.86, although this does not represent a practical arrangement. Figure 3.87 shows the pressure and volume changes to the working fluid throughout the cycle.

The Stirling cycle involves the following process steps.

— *Process 1–2*: the displacer remains stationary and the piston descends. The gas expands and its pressure falls. The gas temperature is cold but it remains constant because of heat transfer from the cold space.

— *Process 2–3*: the displacer descends and the piston rises. The gas is maintained at constant volume and it passes through the regenerator, which is hotter than the gas. The gas heats up due to heat transfer from the regenerator and its pressure rises; the regenerator cools.

— *Process 3–4*: the displacer remains stationary and the piston rises. The gas is compressed and its pressure rises. The gas is hot and remains at constant temperature because of heat transfer to a heat sink.

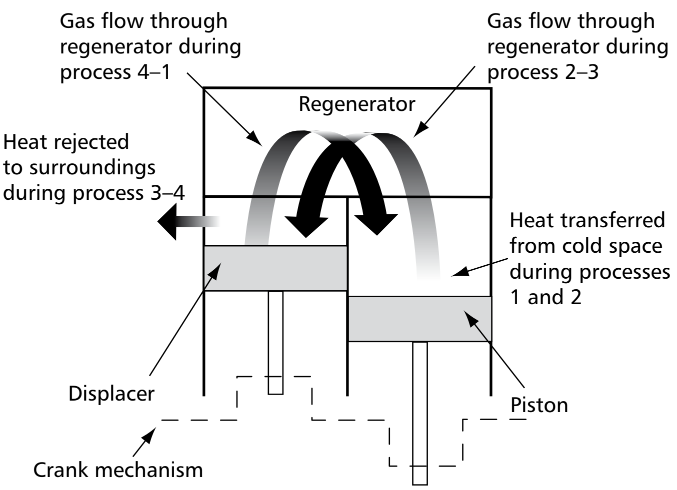


Figure 3.86 Reversed Stirling cycle: schematic

— *Process 4–1*: the displacer rises and the piston descends. The gas is maintained at constant volume and it passes through the regenerator, which is colder than the gas. The gas cools due to heat transfer to the regenerator and its pressure falls; the regenerator heats up.

A variety of practical configurations have been devised including a piston and displacer in the same cylinder, a piston and displacer in separate cylinders and two pistons. The resulting designs are typically quite complex. A further area of design complexity is the heat transfer arrangements since only relatively small areas are available for both the external and regenerative heat transfers. Consequently, rather complex heat exchanger designs tend to be required.

There are currently no commercial Stirling cycle machines available, although various prototype domestic refrigerators based on the Stirling cycle have been built. The higher complexity would be expected to result in a cost higher than conventional vapour-compression systems. Initial estimates of efficiency suggest that COP could be between 14 per cent and 40 per cent higher than equivalent vapour compression machines (Green et al., 1996).

3.3.3.10.3 Ice storage systems

Ice storage allows 'load shifting' or the manipulation of energy demand profiles. Some of the potential benefits of this include:

— reduced energy costs by moving peak demand to times when energy may be cheaper (e.g. overnight)

— reduced chiller size and cost (and refrigerant charge) from reducing peak chiller loads by spreading operation over a longer period

— overcoming building or local area power supply limitations by reducing peak chiller electrical power consumption

— overcoming local regulatory restrictions in some countries, e.g. the United States (California) on the construction of additional power generation plant; in these countries supply companies may positively encourage ice storage systems

— reduced space requirement for 'coolth' storage compared with chilled-water storage systems.

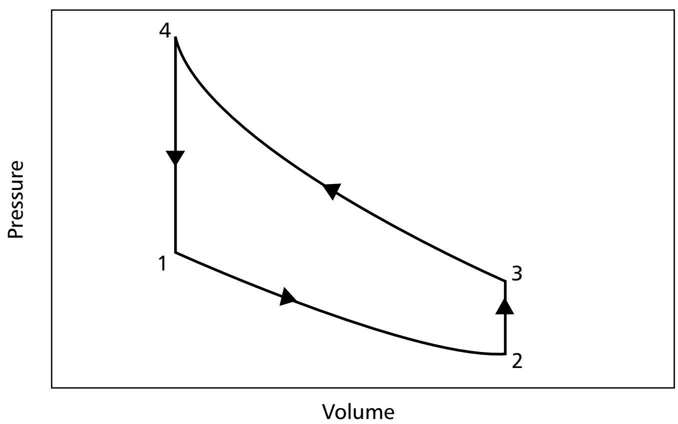


Figure 3.87 Stirling cycle: pressure–volume diagram

A significant disadvantage of ice storage is that the systems are not capable of providing 24-hour cooling unless designed to incorporate additional chiller plant to operate in parallel while ice is being generated overnight.

It should also be considered that the production and melting of ice for building cooling systems is inherently inefficient and causes a reduction in chiller COP due to the lower evaporating temperatures (see section 3.3.3.7, 'Coefficient of performance'). The increase in chiller energy consumption may as high as 15–20 per cent.

It has been established that in the UK electricity grid, power-station carbon-dioxide emissions per kW·h of delivered electricity are lower overnight than during the day. Appropriately designed and controlled chillers with ice storage systems can therefore reduce the overall carbon dioxide emissions arising from the daytime operation of building cooling systems (Beggs, 1997). In particular it is important that chillers are controlled such that their condensing pressure is allowed to 'float' to follow the reduced ambient air temperature at night. This may require the use of electronic expansion valves instead of conventional thermostatic expansion valves (TEVs). By maximising the benefit of lower night-time condensing pressure and temperature on CoP, the negative effect of reduced temperature necessary to produce ice is minimised. Where evaporating daytime air temperatures are high, the effect may even be to reduce overall chiller energy consumption. The designer should establish the overall impact on energy consumption and greenhouse gas emissions by carrying out a thorough energy and TEWI analysis.

There are a wide range of ice storage systems and control strategies. Control strategies should ideally ensure that the stored 'coolth' is used effectively and is not depleted prematurely. Some forms of ice storage, such as the ice on coil (ice builder) system, suffer a significant reduction in energy efficiency if the ice is only partially consumed during the day.

3.3.3.10.4 *Rotary (interotex) absorption chiller*

The interotex absorption chiller, commonly known as the 'rotex' machine, is a double-effect machine that uses process intensification through rotational force to improve heat and mass heat transfer. The active components of the machine rotate at a high speed (typically 550 rpm) subjecting the liquid films to very high gravitational forces thus providing reduced film thicknesses and enhanced transfer coefficients.

A cooling CoP of around 1 has been recorded based on a chilled water temperature of 7 °C and a temperature lift of 45 K from a prototype type machine (IEA Heat Pump Centre, 1999). The concept is more complex than a conventional double-effect absorption chiller and its practicality and economic viability has yet to be proven.

Potential future refrigeration technologies

The following technologies are in various stages of research and may emerge as competitors for various application in the future:

- thermoelectric cooling
- steam jet (ejector) refrigeration
- gas cycle (air cycle) refrigeration
- thermionic refrigeration
- magnetic refrigeration
- pulse tube refrigeration
- thermoacoustic refrigeration
- optical cooling
- vortex tube cooling.

3.3.4 Heat rejection and cooling-water equipment

Heat rejection plant is required to cool the condenser; the efficiency of this process will affect the system efficiency. Overall seasonal efficiencies are therefore influenced by energy efficient design of heat rejection systems. Where possible, opportunities for free cooling should be sought, especially for systems that are operated throughout the year (see section 3.3.3.2).

The basic types of condenser (see Table 3.21 above) are:

- *direct*: air cooled, direct water cooled or evaporative
- *indirect*: condenser heat is rejected via a water system by using cooling towers or dry air coolers or some other form of environmental cooling (see section 3.3.3.2).

Table 3.31 gives a comparison of machine CoPs together with the heat rejection that may be expected when evaporating at 5.0 °C and condensing at 35–40 °C. These are for water-cooled systems; the CoP would be lower for air-cooled systems. The CoPs are approximate and are for comparative purposes only.

Table 3.31 Approximate CoPs and heat rejection

Type	CoP*	Heat rejection/ cooling† (/ kW)
Reciprocating compressor	4	1.25
Scroll compressor	4	1.25
Centrifugal compressor	5.5	1.18
Screw compressor	5.5	1.18
Absorption machine (single effect)	0.68	2.47

Note:

$$\text{CoP} = \frac{\text{compressor cooling power (kW)}}{\text{compressor input power (kW)}}$$

* Evaporator temperature: 5 °C
† Condenser temperature: 35–40 °C

The data in Table 3.31 allow the heat dissipation ratio for different types of cooling technology to be assessed and are an important consideration for heat rejection plant selection, capital cost and life-cycle cost.

3.3.4.1 Air-cooled condensers

Air-cooled condensers are the simplest form of condenser heat rejection plant, in which air passes over finned tubes containing the condensing refrigerant. They are generally found on stand-alone plant such as packaged air conditioners, split systems or some packaged air-handling plant. The air is at ambient dry bulb temperature, therefore condensing pressures will always be in excess of the ambient dry bulb (commonly 15–20 K above ambient) with the implication that air-cooled condensing systems may be less efficient than evaporative-based water-cooled systems (see section 3.3.4.4 for further information).

3.3.4.2 Wet cooling towers

3.3.4.2.1 General

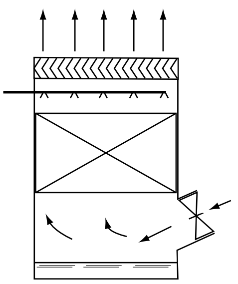
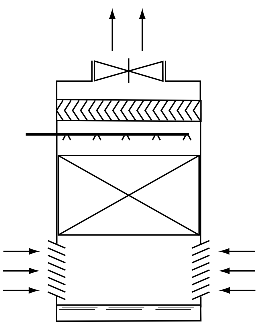
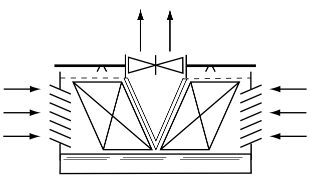
A cooling tower cools the condenser water by evaporative cooling. There are two types of wet cooling tower.

— *Open circuit*: water from the condenser is pumped to the cooling tower and is cooled by the evaporation of some of the condenser water. This requires all the water passing through the condenser to be treated and results in increased water consumption due to drift losses.

— *Closed circuit*: condenser water is circulated in a closed loop and a separate water circuit is pumped through the cooling tower, cooling the condenser water by transferring heat through a heat exchanger. This minimises water treatment costs but it also reduces energy efficiency due to the temperature difference across the heat exchanger, although this effect can be minimised by specifying a high-efficiency heat exchanger.

In a mechanical draught tower (see Table 3.32) the entering water is sprayed into the plastic fill packing and one or more fans force air through the packing to enhance evaporation and, hence, the cooling effect. The cooled

Table 3.32 Mechanical draught cooling towers

Type	Description
(*a*) Forced draught 	Fans are situated at the air intake and blow ambient air into the tower across the wet packing causing a portion of the water to be evaporated, thus removing heat from the remaining water Advantages: — fans located close to the ground, thus vibration is kept to a minimum — fractionally more efficient than induced draught since velocity pressure converted to static pressure does useful work, while the fan handles inlet cold air, and thus the weight of air per unit volume is greater than in the induced draught arrangement — fans and motors are situated in a comparatively dry air stream and are more easily accessible for maintenance Disadvantages: — limited fan size, thus a larger number of smaller fans of higher speed are needed compared with induced draught arrangement, resulting in more noise (but tower itself provides some attenuation) — tendency for ice to form on the fans in winter and block or throttle the intake — some types can be prone to recirculation of used air into the accessible low pressure fan inlet and resulting reduction in performance may be substantial; this occurs if outlet air velocities are low; the air may be ducted away at high velocity but at the expense of greater resistance and increased fan power requirements
(b) Induced draught  Counterflow  Cross draught	Fans are situated in the air outlet from the tower, usually on the top, but sometimes in the side or in the ducting Advantages: — large fans possible (hence low speed and low noise) — recirculation of air unlikely due to higher outlet velocity — more compact plan area than (*a*) due to absence of fans on side Disadvantages: — more prone to vibration since fan is mounted on superstructure — mechanical parts less readily accessible for maintenance — mechanical parts located in a hot, humid air stream — high inlet velocities can draw in rubbish; air filters can be fitted There are two types of induced draft cooling towers: 'counterflow' and 'cross draught': (*i*) Counterflow Fans create vertical air movement up the tower across the packing in opposition to the water flow Advantages: — maximum performance arrangement as the coldest water is in contact with the driest air — up to three sides of the tower can be obstructed by adjacent buildings, provided that the remaining air inlet(s) are suitably increased in size Disadvantages: — mechanical parts and water distribution are not always easily accessible for maintenance (*ii*) Cross draught Fans create horizontal air flow as the water falls across the air stream. Some types have a greater plan area than (c), but the air intakes can be full height of tower which is consequently of low silhouette, blending well with the architectural requirements. Rain ingress should be taken into account when considering water treatment dosage Advantages: — low silhouette Disadvantages: — some risk of recirculation of saturated vapour if sited in a confined space — if uncovered, distribution basin will collect rubbish; a cover should be provided unless installation is indoors — location demands unobstructed air flow towards each end of tower

water falls to the base reservoir and is pumped back to the condenser. Natural draught cooling towers are rarely used for building air conditioning applications due to their much greater height and high approach temperature.

Various European standards for cooling towers have been produced by the Eurovent/CECOMAF Cooling Towers Working Group, upon which all the major manufacturers are represented. The standard for testing is Eurovent 9/2-92: *Thermal Performance Acceptance Testing of Mechanical Draught Standardized Water Cooling Towers* (Eurovent, 1992). This standard forms the basis BS EN 13741 (BSI, 2003).

Table 3.32 above describes the main types of mechanical draught cooling towers. Mechanical draught cooling towers use fans to move the air through the tower, thus providing absolute control over the air supply, as opposed to 'atmospheric' or 'natural draught' types. With the use of efficient eliminators, drift losses have been reduced to as little as 0.001 per cent of water flow rate. The advantages of mechanical draught towers compared with natural draught towers include:

- compact (i.e. small plan area)
- close control over water temperature
- small static lift
- siting of tower is independent of prevailing wind direction (refer to HSE Approved Code of Practice and guidance L8: *Legionnaires' disease: The control of Legionella bacteria in water system*s (HSE, 2013)
- with efficient heat transfer packing, approach temperatures of 2–3 K are achievable, though 3–7 K is usually preferred.

The disadvantages include:

- fan powers can be higher than air-cooled condenser equivalent (see Table 3.32)
- recirculation of discharged air back into the air intake must be avoided or performance will suffer.

Centrifugal fans are generally used to achieve low operating noise levels but variable fan speed motors should be considered for very noise-sensitive locations.

The basic information required by the equipment manufacturer is as follows:

- design water flow rate
- design temperature range through which the water is to be cooled
- design ambient wet bulb temperature
- operational height above sea level
- any limitations on height, floor plan, weight, noise or appearance
- features that may affect the free flow of air to and from the unit
- preferably a drawing showing the tower location onsite.

3.3.4.2.2 Selection of cooling tower site

The location of the cooling tower should receive careful consideration. There should be sufficient free space around the tower to allow free flow of air both to the inlet and from the discharge outlet.

Recirculation of the hot discharge back into the inlet must be avoided, as it will substantially reduce performance. Discharge ducting or extended fan casings may be necessary to minimise recirculation risk and the effect of these components on fan power should be taken into account. The siting of the cooling tower should be such that the discharge air is not close to fresh air inlets and does not produce condensation upon nearby buildings and in the surrounding area.

The presence of exhaust heat from other equipment or of contaminated air from process plant (especially kitchen extract with high grease content) will reduce tower performance and may produce corrosive conditions. The tower should be sited as far away as possible, upwind of smoke stacks and other sources of pollution. Where local atmospheric air pollution is unavoidable, filters may be provided for cooling-tower air inlets. The tower location should be carefully studied in relation to the noise created by the air and water.

The local authorities should always be consulted on the connection of mains water supplies to tanks and pumping circuits. In general, it will not be permissible to connect pumps directly to the main and a break tank must be interposed. Local fire regulations should be consulted when a tower is to be installed, particularly if any hazard or opportunity for ignition of the tower is present.

3.3.4.2.3 Water treatment

Every water-cooling tower requires an appropriate water-quality management regime. This is essential to minimise the risk of legionellosis and to control corrosion and fouling (e.g. by bacterial growth, such as *Pseudomonas*). Biological contamination, however, can be controlled only through the use of biocides and such treatment should be initiated at system start-up and continued regularly thereafter. Poor water treatment can greatly increase energy and water costs. *Legionella* can be controlled if the tower is designed and operated in accordance with CIBSE TM13: *Minimising the risk of Legionnaires' disease* (CIBSE, 2013a). The designer and owner/operator should ensure compliance with relevant UK regulations (see section 3.3.2.4). Compliance with the HSE Approved Code of Practice and guidance L8: *Legionnaires' disease: The control of Legionella bacteria in water systems* (HSE, 2013) is mandatory. For information on corrosion and further information on water treatment see CIBSE Guide G: *Public health engineering* (CIBSE, 2014).

Where cooling towers need to be site-performance tested for confirmation of compliance with design conditions, the relevant standard for the UK is BS 4485-2: *Water cooling towers. Methods for performance testing* (BSI, 1988).

3.3.4.3 Dry air coolers

Dry air coolers are heat exchangers of construction similar to that of an air-cooled condenser. They are designed for cooling liquids (generally glycol–water) in a closed circuit. The freezing point of the liquid must usually be at least 5 K below the minimum winter ambient temperature of the site of installation. The cooling effect from night-time sky radiation should also be considered where pipework is exposed.

Selection is normally to suit each individual case, specifying maximum noise level, type of liquid, ambient temperature, liquid inlet temperature, liquid outlet temperature, maximum allowed pressure drop etc. They are simple in construction and operation with low installation and maintenance costs.

As the water-distribution system is closed, atmospheric contamination cannot occur and microbiological control of water quality is simplified. Ambient-air contamination could be a hazard and precautions similar to those for air-cooled condensers should be observed.

On some installations dry coolers have been used with sprayed water, which improves their efficiency due to the evaporative cooling. These units are referred to as 'wet-and-dry coolers' or 'adiabatically enhanced' dry coolers. Local regulations regarding water treatment must be complied with.

3.3.4.4 Evaporative condensers

Evaporative condensers are similar in operation to induced (or forced) draught water-cooling towers (see section 3.3.4.2). The main difference is that the refrigerant vapour is condensed and is circulated inside the tubes of the condensing coil, which is continuously wetted on the outside by means of a recirculated water system. Air is simultaneously blown upwards over the coil, causing a small portion of the recirculated water to evaporate. The coil section consists of the closely nested pipes, water-distribution system and moisture eliminators, enclosed in a galvanised steel or plastic casing. Eliminators are preferably constructed from ultraviolet-resistant polyvinyl chloride (PVC).

Evaporative condensers can be an energy efficient method of heat rejection from a refrigeration system to ambient air because the surface temperature of the condenser tubes approaches the air wet bulb temperature and there is no other intermediate heat exchange process required. The ambient wet bulb temperature is often 8–12 °C lower than the dry bulb temperature, resulting in power savings of up to 30 per cent compared with an equivalent air-cooled condenser system. However a study has shown that consumption from water pumping, part load operation and other ancillaries can override this advantage (Clark and Gillies, 2014). Water pumping requirements are also less than for a cooling tower because the outside of the tubes only need to be kept wetted. Evaporative condensers are particularly effective with refrigerant migration (thermosyphon) chillers as they extend the time that thermosyphoning can occur.

Evaporative condensers have the following disadvantages.

— They normally need to be located close to the compressors in order to avoid long refrigerant pipe runs that would otherwise cause oil return difficulties, excess pressure drop and excessive refrigerant charge.

— A good water treatment and cleaning programme is essential to minimise the risk of *Legionella* and the formation of scale, corrosion and fouling.

It is essential that the guidance on controlling legionella given in the HSE Approved Code of Practice and guidance L8: *Legionnaires' disease: The control of Legionella bacteria in water systems* (HSE, 2013) and CIBSE TM13: *Minimising the risk of Legionnaires' disease* (CIBSE, 2013a) are followed (see section 3.3.2.4 '*Legionella*'). However, the risk of *Legionella* is lower than with cooling towers, because the water volume is lower and restricted to the unit itself without extensive water circulation pipes and the risks associated with dead legs. It is important that manufacturers' recommendations for biocide dosing concentrations are adhered to, as excessive usage can cause rapid corrosion of galvanised coil tubes and other metalwork. Scale control is also important, as even small-scale levels can significantly reduce heat transfer efficiency. Scale formation is minimised if the tubes are kept uniformly and totally wetted.

An evaporative condenser is an extension of an air-cooled condenser. As well as air being blown over the tubes, the tubes themselves are continuously wetted by a recirculating water system. They are able to achieve a similar performance to water-cooled condensers and open-circuit cooling towers, but eliminate the condenser water pumps. See section 3.3.6.3 for further details.

3.3.4.5 Ground, river and lake water cooling

In most air conditioning applications where water is used, the water is recirculated and cooled by an evaporative process; make-up losses are catered for by the use of a storage tank connected to the mains supply via a ballcock or similar device.

Environmental cooling may be used directly for cooling a building. If it is not cold enough then it may be used as a heat sink for heat rejected from the condensers. Examples of environmental cooling include:

— ambient air

— ground water, or ground rejection

— rivers or lakes

— seawater.

Further information on these sources of cooling is given in section 3.3.3.2.

3.3.5 Heat pumps

The term 'heat pump' is used to describe a refrigeration system in which the useful output is measured in terms of rejected heat. They are primarily used for heating buildings but may be reversible, in which case either heating or cooling can be supplied.

Further general information on heat pumps, heat recovery and integrated systems can be found in *Refrigeration, Air Conditioning and Heat pumps* (Hundy et al., 2016).

3.3.5.1 Heat pump types

Heat pumps are available as:

— air-source heat pumps (ASHPs)

— ground-source heat pumps (GSHPs)

— water-source heat pumps (WSHPs).

Heat pumps are normally operated to deliver heating to a building but, all types are capable of heating, cooling or simultaneous heating and cooling depending on the application of the technology.

ASHPs are most commonly used as heating and cooling split air conditioning systems or VRF systems. Air handling 'rooftop' units also commonly incorporate a refrigerant-based cooling and heating system that operates as an ASHP when heating is required. One disadvantage of ASHPs is that they may require periodic defrost cycles to defrost the outdoor coil when operating in heating mode in winter.

GSHPs are typically connected to boreholes or closed-circuit loops embedded in the ground. Other possibilities include connection to boreholes or ground water via a heat exchanger. GSHPs are typically more efficient than ASHPs but have the disadvantage that they require significant ground installations to collect or reject source heat.

Some manufacturers have developed packaged chiller/heat pump technology to the extent that the heat source can be from:

— simultaneous heating and cooling

— integration of an evaporator provided with a ground-source water connection

— air source.

3.3.5.2 Heat collection

ASHPs collect heat from the air. The efficiency of an ASHP reduces as the outdoor air temperature decreases. Locating the condenser air intakes local to the building ventilation exhaust stream ensures that the air-on temperature to the heat source evaporator is maximised to improve efficiency.

GSHP heat collection falls into two major categories: open loop and closed loop.

Open-loop systems pump water to and from an aquifer either to extract heat (heating mode) or reject heat (cooling mode).

These are possible where a suitable aquifer is present and the requirements of the local environmental authorities can be met. Open-loop systems offer significant potential for passive cooling when bypassing the heat pump/chiller.

Closed-loop systems consist of sealed circuits of plastic pipework either installed in boreholes or trenches.

Please refer to CIBSE TM51: *Ground source heat pumps* (CIBSE, 2013b) for further information.

3.3.5.3 Efficiency

Refer to section 3.2.3.3.3 'Unitary heat pumps' for information.

3.3.6 Components

3.3.6.1 Introduction

This section provides information for designers and users on the equipment required for refrigeration and heat rejection systems. It sets out critical design issues relating to specific items of equipment and the key points to be considered in their selection. Where relevant, it provides references to statutory/mandatory regulations and guidance relating specifically to the design, installation or use of the equipment. Where appropriate, information is given on the advantages and disadvantages of alternative types of component.

3.3.6.2 Evaporators

There are a number of different evaporator design options and the construction is dependent on the refrigerant employed.

Evaporators used with HFCs or HCs are typically constructed with copper tubes (with aluminium fins on air side). Evaporators used with ammonia are typically galvanised mild steel tubes (with aluminium fins on air side) or stainless steel tubes (with aluminium or stainless steel fins on air side). Evaporator FCUs used with carbon dioxide are typically stainless steel tubes to withstand the higher design pressures (with aluminium or stainless steel fins on air side).

3.3.6.2.1 Shell and tube direct expansion (DX) evaporators

A shell and tube DX evaporator comprises a steel vessel (shell) containing straight tubes, located between end tube plates, with removable end covers. The refrigerant flows through the tubes. Water or another secondary coolant passes through the shell and over the tubes. A series of baffles within the shell improves heat transfer between the tubes and the secondary coolant.

Shell and tube DX evaporators are generally used for water temperatures down to 4 °C or at lower temperatures for cooling secondary coolants such as brine or glycol. Refrigerant is distributed through the tubes in single or multiple pass, and control is either by thermostatic or electronic expansion valve. The latter allows more effective use of evaporating surface, accommodating greater fluctuations in load (see section 3.3.6.5). Shell and tube DX evaporators are sometimes referred to as 'chillers'.

Advantages:

— small refrigerant charge inside the tubes

— direct system (only one heat transfer operation).

Disadvantages:

— coolant side is generally inaccessible (chemical cleaning can be employed but caution is necessary)

— load fluctuations are more difficult to control than with flooded evaporators (see also section 3.3.6.5)

— prone to freezing when used as a water chiller.

3.3.6.2.2 Flooded refrigerant evaporators

The construction is similar to a shell-and-tube evaporator but the refrigerant is contained in the shell and the water or secondary coolant flows through the tubes. However, the shell incorporates a larger space at the top to provide liquid droplet separation to ensure dry gas return to the compressor. These evaporators are suitable for water temperatures down to 4 °C and lower for secondary coolants such as brines and other fluids.

Advantages:

— permits increased system efficiency compared with DX as the refrigerant evaporating temperature is higher than in a DX shell-and-tube evaporator (typically 4–5 K higher)

— large fluctuations in load are accommodated without risk of freezing (an expansion device that is sensitive to liquid level is normally used)

— direct system with higher rate of heat transfer than DX types.

Disadvantages:

— large refrigerant charge

— higher cost than DX evaporators.

3.3.6.2.3 Gasketed-plate heat exchangers

A gasketed-plate heat exchanger consists of a pack of pressed profile metal plates with portholes for the passage of the two fluids between which heat transfer will take place. The plate pack is compressed between a frame plate and a pressure plate by means of tightening bolts. The plates are fitted with a gasket that seals the channel and directs the fluids into alternate channels. This arrangement allows additional plates to be easily added to increase the duty of the heat exchanger. The channels formed between the plates are arranged so that the refrigerant flows in one channel and the coolant in the other. Very low coolant and refrigerant temperature differences are possible (under 2 K) making plate heat exchangers very suitable for systems that employ free-cooling techniques.

Refrigerant distribution to ensure equalised flow in all the sections is particularly important. The heat exchanger should be vertically mounted and the refrigerant should enter at the bottom. Many designs have in-built distributors.

Advantages:

— very low refrigerant charge (internal volume is only about 10 per cent of that for flooded evaporators)

— heat transfer coefficients can be three to four times greater than that of a shell-and-tube heat exchanger.

Disadvantages:

— oil fouling can occur, affecting heat transfer (careful attention should be given to the type of refrigerant and oil used in the circuit and the manufacturer's approval should be obtained with regard to compatibility of refrigerants and oils)

— prone to freezing due to low mass of coolant inside the heat exchanger (but less susceptible to permanent damage than shell and tube evaporators.

3.3.6.2.4 Brazed/welded-plate heat exchangers

The brazed/welded-plate heat exchanger is a variant of the gasketed-plate heat exchanger, but it cannot be dismantled for cleaning. It is composed of a number of pressed profile plates brazed or welded together. The plates are normally stainless steel. Brazed plates are coated with copper on one side. The assembly is clamped together with end plates and heated in a vacuum oven until the copper melts and forms a brazed joint. The channels between the two plates can be varied in their cross sectional dimensions to achieve the optimum heat transfer coefficient for the required application.

The advantages and disadvantages of brazed/welded-plate heat exchangers are the same as for gasketed-plate heat exchangers.

3.3.6.2.5 Evaporator fan coil units (FCUs)

Evaporator FCUs are available in various forms and constructed of material that is compatible with the refrigerant employed. Refrigerant passes through the FCU tubes and air passes over the tubes. The refrigerant evaporates, absorbing heat from the air and the air is cooled.

Evaporator FCUs come in various forms and can be selected with FCU casing and cooling coil protective coatings depending on the application. Typical applications include variable refrigerant volume (VRV) indoor units and chill and cold store units.

3.3.6.3 Condensers

There are a number of different condenser design options and the construction is dependent on the refrigerant employed.

Condensers used with HFCs or HCs are typically constructed with copper tubes (with aluminium fins on air side). Condensers used with ammonia are typically galvanised mild steel tubes (with aluminium fins on air side) or with stainless steel tubes (with aluminium or stainless steel fins on air side). Condensers used with carbon dioxide are typically stainless steel tubes to withstand the higher design pressures (with aluminium or stainless steel fins on air side).

3.3.6.3.1 Air-cooled condensers

Air-cooled condensers consist of aluminium finned tubes, in which the refrigerant is condensed, which air passes over. They are usually constructed in single units or banks. There are a number of condenser fin coating and material options depending on application (e.g. corrosive atmospheres).

The condensing temperature is related to the dry bulb temperature of the cooling air and is higher than the condensing temperature of systems that use evaporative heat rejection. The condensing temperature and pressure should be kept as low as possible by using a large condenser surface area, although there are practical and cost limits to this. In practice, the ΔT between the cooling air inlet and the condensing temperature is around 15–20 K. High ΔT will result in inefficiency of the refrigeration process and will lead to an increased risk of the system high-pressure cut-out tripping in hot weather causing a loss of cooling capacity.

Air-cooled condensers are particularly well suited to small packaged systems but due to their size and relatively high ΔTs, they are seldom used for systems greater than about 1000 kW of cooling, especially in hot climates.

Many systems require that the condensing pressure be controlled to maintain it above a certain minimum value to ensure satisfactory operation of the expansion device (see section 3.3.6.5). Condensing pressure is commonly controlled by speed control or staging of the condenser cooling fans, although sometimes motorised dampers are used. To minimise noise levels, low speed propellor or centrifugal fans are often used and discharge is directed upwards. The location and layout of condensers should follow the guidance given in section 3.3.4.2 to minimise short cycling and consequent head-pressure difficulties.

3.3.6.3.2 Water-cooled shell-and-tube condensers

Water-cooled shell-and-tube condensers comprise a welded pressure vessel containing the condensing surface in the form of plain or finned straight tubes located between end tube plates with removable end covers for access to the water tubes. These condensers are frequently used with cooling towers or dry air coolers.

For refrigerants other than ammonia, the water tubes are normally copper with special alternatives for seawater or polluted water conditions, such as rivers, pond or lakes. For ammonia, the tubes are to be steel or stainless steel.

3.3.6.3.3 Gasketed/brazed-plate heat exchangers

When used as a condenser, the same considerations should be used as noted above for evaporators, particularly the significant reduction in size compared with a shell-and-tube vessel, but, due to the higher temperatures, the entrained oil will have a lower viscosity and therefore oil fouling should be less of a problem. Refrigerant distribution is not an issue for condensers.

3.3.6.4 Compressors

There are five basic types of compressor currently used in the refrigeration industry:

— reciprocating (piston cylinder)

— sliding vane

— scroll

— screw

— centrifugal.

The type of compressor used in an installation depends on the application and cooling capacity required. It is quite common to apply compressors in multiples and this extends the range for a particular application accordingly.

Current practice is related to the required cooling capacity:

— large installations:
  - centrifugal (typically above 1000 kW)
  - screw compressors

— medium to large installations:
  - semi-hermetic screw compressors (typically above 400 kW),
  - mini-centrifugal compressors
  - multiple-scroll compressors
  - reciprocating compressors (less common)

— low to medium installations:
  - predominantly scroll compressors.

Most compressors are driven by an enclosed electric motor and are either hermetic or semi-hermetic.

3.3.6.4.1 Hermetic compressors

These are built into a welded shell with no access to internal parts for servicing or repair.

Sliding-vane, scroll and small reciprocating machines are usually hermetic and are typically suction gas cooled, which means that the motor is cooled by the refrigerant vapour before it is compressed, which tends to reduce the capacity of the compressor.

3.3.6.4.2 Semi-hermetic compressors

These have removable covers that allow limited access to internal components for servicing repair.

Larger reciprocating, screw and centrifugal types can be semi-hermetic. These are, again, typically suction gas cooled.

Centrifugal compressors may incorporate a separate DX motor cooling system. This is also a parasitic compressor load.

3.3.6.4.3 Open-drive compressors

Large reciprocating and screw compressors can be open drive.

Open-drive compressors are typically used for systems operating with ammonia as a refrigerant. This is primarily due to copper windings not being suitable for use with ammonia.

Open compressors are driven by an external motor, or in some cases by an engine via a drive shaft that passes through a shaft seal.

The shaft seal is a common cause of refrigerant leaks and for this reason open compressors should not be considered for HFC refrigerants.

3.3.6.4.4 Refrigerant suitability

Ammonia requires open compressors, reciprocating or screw. Oil cooling is generally necessary.

HFCs can be applied with any enclosed motor type.

R134a is most suited to screws and centrifugal compressors.

R410A is higher pressure than HCFCs such as R22 and requires suitably designed scrolls. R410A has, however, been used in water-cooled screw compressor applications.

HCs can be applied with any enclosed motor type but may require special electrical protection, and manufacturer approval should be sought.

Carbon dioxide requires dedicated machines, either reciprocating or scroll.

3.3.6.4.5 Scroll compressors

Scroll compressors are hermetically sealed, rotary, positive-displacement machines with one fixed and one orbiting scroll that progressively compresses refrigerant with a constant volume ratio. They have comparable or slightly higher efficiencies than reciprocating machines at typical air conditioning application temperatures. Most types of scroll compressor are compliant in that they allow some radial or axial movement of the scroll, which allows them to cope with some liquid returned to the compressor.

Lubricating oil is stored at low pressure in the crankcase and circulated by an internal pump. Some scroll compressors have motors cooled by discharge gas and in this case the oil is stored at high pressure. This is sometimes termed a 'high side shell'.

Noise and vibration levels are less than for reciprocating compressors. Some scrolls are provided with an economiser facility (see screw compressors section for description).

Scroll compressors are applied in chillers, VRF and heat pump systems over a wide size range.

3.3.6.4.6 Reciprocating compressors

Reciprocating compressors are positive-displacement machines with the refrigerant vapour compressed by pistons moving in a close-fitting bore. This type was widely used in air conditioning applications, being adaptable in size, number of cylinders, speed and method of drive. Each cylinder has an automatic pressure-actuated suction and discharge valve and the bearings are oil lubricated. Lubricating oil is stored at low pressure in the crankcase and circulated by an internal pump. Reciprocating compressors are available in a wide range of sizes ranging from a single-cylinder type to eight cylinders or more.

Reciprocating compressors are being superseded in many packaged chiller applications by scroll, screw and mini-centrifugal compressors.

3.3.6.4.7 Rolling-piston compressors

Rolling-piston or sliding-vane compressors have one or two blades, which do not rotate but are held by springs against an eccentric, rotating roller. These compressors require discharge valves. This type has been developed extensively for domestic appliances, packaged air conditioners and similar applications up to a cooling duty of approximately 15 kW.

3.3.6.4.8 Screw compressors

Screw compressors are high-speed positive-displacement machines, compression being obtained by the interaction of two screw-cut rotors or a single rotor meshing with two toothed wheels.

Most screw compressors can be optionally equipped with an economiser, which allows an additional charge of refrigerant gas to be pumped (a form of supercharging). A port positioned in the compressor casing is connected to an intermediate heat exchanger/liquid sub-cooler vessel. The higher pressure in this vessel allows the additional charge to flow into the port and be compressed together with the gas induced by the normal suction process.

This arrangement is termed an economiser, and provides an increase in cooling capacity that is significantly higher than the extra power consumption, thus improving the compressor CoP. The additional gas is provided by evaporating some liquid from the liquid line in the heat exchanger, usually a plate heat exchanger (PHE). This sub-cools the main liquid, passing through the PHE to the main expansion device. A larger capacity per kilogram flow in the evaporator is achieved whilst the compressor pumps only the same suction mass flow.

Screw compressor economiser operation is normally at or near 100 per cent slide-valve position and is therefore not beneficial at part load.

The benefit of the economiser is normally in the order of 10 per cent increase in capacity in the evaporator. This needs to be accounted for in the system heat rejection.

Screw compressors are cooled by oil injected into the machine to seal the running clearances between the rotors and casing. Oil separators are generally integral to the compressor or included in packaged units. Although the design is specific to the particular manufacturer, it is important that the specifier understands that some machines rely on a pressure differential across the compressor for lubrication, and that head-pressure control may be necessary to maintain minimum oil pressure. This may force the compressor to operate at artificially high discharge pressures at a resultant energy penalty.

3.3.6.4.9 Centrifugal compressors

Centrifugal machines are 'dynamic'-type compression devices and can be single or multi-stage. The centrifugal action on the refrigerant gas allows large volumes to be compressed over low compression ratios with a relatively compact machine. They range in size from approximately 300 kW to more than 15 MW. Centrifugal compressors can be either hermetic or open. They are also available with economisers.

Hermetic units incorporate an induction motor and an internal gear, which allows the impellers to run at speeds of between 18 000 and 20 000 rpm. Open-type machines can be driven by electric motors, steam turbines, gas turbines and gas engines. For capacities larger than about 7000 kW all machines are of the open type.

Mini-centrifugal machines are typically two-stage. They range in size from approximately 250 kW to 500 kW and are typically semi-hermetic. They are also available with economisers.

The compressor typically incorporates a variable-speed electric motor that drives a shaft located by magnetic bearings. The frictionless drive means that the system requires no oil. The variable-speed motor drives the shaft at up to 35 000 rpm.

The full-load efficiency of this compressor design is comparable with screw compressors and centrifugal compressors, however the part load efficiency is significantly better making this design competitive with respect to life-cycle cost analysis.

3.3.6.5 Expansion devices

The purpose of the expansion device is to control the flow of refrigerant from the high-pressure condensing side of the system into the low-pressure evaporator.

In most cases, the pressure reduction is achieved through a variable flow orifice, either modulating or two-position.

Expansion valves are classified according to the method of control.

3.3.6.5.1 Capillary tubes and restrictors

A long small-bore capillary tube is used in domestic refrigerators and freezers and some other small direct-refrigeration and air conditioning systems.

This device has a fixed restriction, usually in the form of a small bore tube, the drop in pressure being determined by the length and diameter of the tube. These devices can be very effective and do not require potentially inefficient superheating in the evaporator. However, they are prone to blockage and do not allow for any adjustment of superheat levels.

Tube bores of 0.8–2 mm with lengths of 1–4 m are common. The capillary tube is only fitted on factory-built and tested equipment, with exact refrigerant charges. It is not applicable to field-installed systems.

A restrictor in the form of a precision-drilled orifice is sometimes used instead of a capillary tube. For reversible systems, it can take the form of a bullet, which is free to move horizontally by a small amount. It is pressed against a seat, forcing the refrigerant through the central restriction, which acts as an expansion device. When the flow reverses, the bullet moves back to a second seat, but grooving allows flow around the outside as well as through it, so that the restriction is very small.

3.3.6.5.2 Thermostatic expansion valve (TEV)

This is an automatic valve that controls the rate of liquid refrigerant flow to the evaporator whilst maintaining a predetermined degree of superheat at the evaporator outlet. Some superheat, typically 5 K, is necessary for TEVs although this does result in a small reduction of evaporator effectiveness and the evaporator should be sized accordingly. This is necessary to ensure that all of the gas has evaporated to prevent liquid carryover to the compressor, which could result in loss of efficiency and cause mechanical damage.

There are two types of thermostatic expansion valve: 'externally equalised' and 'internally equalised'. Externally equalised types are normally used where an accurate flow rate of liquid refrigerant together with good modulation is required; a typical application is in air conditioning systems.

A TEV controls superheat, and small superheat fluctuation is normal. However, an incorrectly sized or adjusted valve can result in high fluctuations, causing inefficiencies and liquid carry over in extreme cases (hunting).

A TEV does not directly maintain the correct evaporating temperature. When there is less heat, input to the evaporator superheat reduces. This causes the valve to close, reducing the refrigerant flow and thereby refrigerating effect. There is an associated increase in superheat and decrease in pressure, leading to a decrease in the evaporating temperature. If heat input increases, the reverse occurs.

A major disadvantage of TEVs is that they require a certain minimum refrigerant pressure difference across them. In cold weather the condensing temperature often has to be held artificially high (termed 'head-pressure' control) to ensure correct operation of the TEV, resulting in higher than necessary compressor pressure ratio and increased power.

In typical systems with air-cooled condensers, this is usually achieved by modulating the condenser cooling fans. Reduction in fan power requirement can offset additional compressor power. Slightly modified and more expensive balanced port TEVs can reduce the minimum head-pressure requirement.

3.3.6.5.3 Electronic expansion valve

Electronic expansion valves use an external electronic signal rather than a thermostat and provide much closer control of superheat by virtue of the more accurate sensing of the superheat temperature in relation to the saturation temperature. A controller is required. They can operate with a wider range of evaporating and condensing temperatures than TEVs and do not require the condensing temperature to be held artificially high in cold weather (head pressure can 'float'). This allows input power savings at low ambient temperatures.

The disadvantage of these valves is that they are more expensive and more difficult to set up than TEVs. Failure to commission them properly may lead to damaging liquid slugging of compressors. Electronic expansion valves should always be considered for large direct systems and chillers, where the energy savings may easily outweigh the extra cost. Electronic expansion valves may also be integrated into an electronic or microprocessor control system.

3.3.6.5.4 Float-valve regulators

There are two types of float valve that can be used depending on whether they are fitted to the high-pressure side of the system or the low-pressure side.

A high-side float valve controls the flow of liquid refrigerant passing from the condenser to the evaporator. It is used with systems with 'flooded' evaporators, which are normally chillers with cooling capacities greater than about 400 kW. Careful attention is necessary with regard to the correct charge of refrigerant. Overcharging will result in liquid refrigerant returning to the compressor with subsequent damage. Undercharging will result in insufficient refrigerant in the evaporator, causing erratic and inefficient operation.

A low-side float valve directly controls the level of refrigerant in the evaporator or in a vessel feeding the evaporator. It is important that a minimum liquid level is maintained under all load conditions.

3.3.6.6 Water chillers

The vast majority of chillers are vapour-compression refrigeration systems, usually supplied as a factory-assembled unit, sometimes having a remote condenser.

Factory-built packaged chillers are generally preferable to site-erected systems. Site-erected systems increase the risk of refrigerant leakage and generally have longer runs of pipework, which increases the refrigerant charge. There are considerable differences between chillers supplied by different manufacturers, and the designer must make careful comparisons before choosing a supplier.

Table 3.33 summarises chiller types with reference to compressor technology.

3.3.6.6.1 Absorption chillers

Absorption chillers are heat driven machines and do not have a compressor. They are larger and heavier than their vapour-compression equivalents, so the designer must confirm the weight and dimensions with the manufacturer.

Various types of absorption chiller are available, but generally the choice of chiller type is determined by the temperature level of the available heat source and the temperature level of the load. Further information is given in section 3.3.3.7.

3.3.6.6.2 Multiple chillers

In large air conditioning systems, it is a common practice to split the refrigeration capacity between multiple machines in parallel with chilled water control. Unless careful attention is given to low load-balance calculations, frequent compressor cycling can occur, exceeding the manufacturer's limits. It is also essential to co-ordinate the design of the control of the air handling equipment with that of the refrigerating machines — the choice being between a constant-flow and a variable-flow chilled-water system.

Large systems may require the use of several chillers to meet the required capacity and/or to provide plant redundancy. In these cases the following circuit arrangements should be considered (see Figure 3.88).

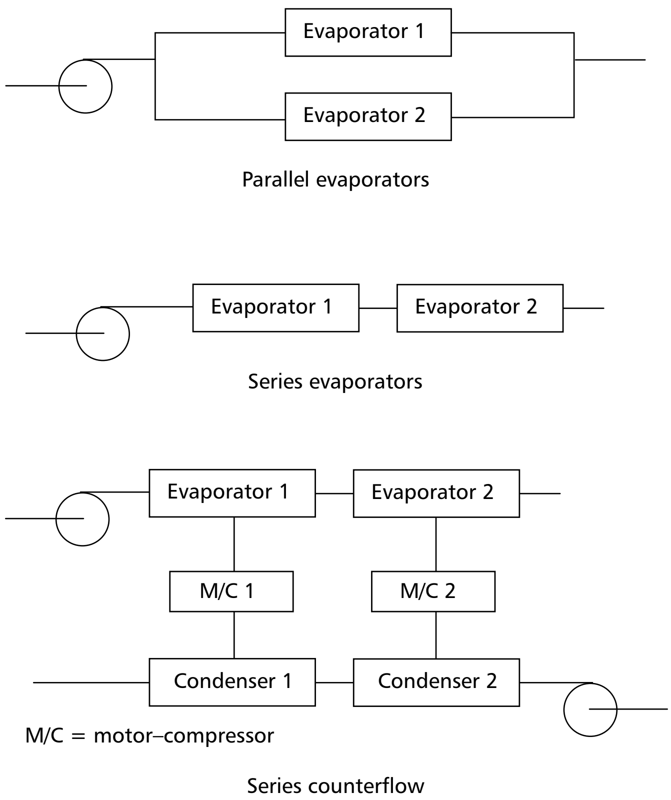


Figure 3.88 Chilled water systems: evaporator/condenser arrangements

Table 3.33 Overview of vapour compression chillers

Type	Sub-group	Cooling range / kW	Refrigerant type	Capacity control*
Semi-hermetic	Reciprocating	100–1000	HFC	Cylinder unloading
Semi-hermetic	Single screw	200–2000	HFC	Moving plate
Semi-hermetic	Twin screw	200–3000	HFC	Slide valve
Hermetic	Twin screw	200–1000	HFC	Slider system, staged ports, variable speed
Hermetic	Scroll	5–750	HFC	Compressor staging
Hermetic	Mini-centrifugal	250–1500	HFC	Variable speed
Hermetic	Centrifugal (multi-stage)	300–15000	HFC	Inlet guide vanes (all cases); variable speed (some cases)
Open	Reciprocating	100–1000	Ammonia	Cylinder unloading
Open	Screw	200–3000	Ammonia	Slider valve, variable speed

* See section 3.3.7

(*a*) *Parallel evaporators*: parallel circuits allow multi-pass heat exchangers at a relatively low water pressure drop, consequently a lower pump power is required than for a series circuit. However, a slightly higher compressor power is required than for a series circuit, due to both machines having the same evaporating temperature. The designer should design the controls carefully to avoid short cycling under partial load conditions. There is also a danger of freezing one evaporator when the other is switched off and control is by a common thermostat downstream of the evaporators.

(*b*) *Series evaporators*: compared with the parallel arrangement, systems of this type use a higher chilled water pressure drop, so a higher pump power is required. Consequently, a single-pass evaporator may well be necessary, and component design could suffer. The compressor power is slightly lower than for a parallel arrangement, as the upstream machine will have a higher evaporating temperature than the downstream machine. Chilled water temperature is generally easier to control than with the parallel arrangement.

(*c*) *Parallel condensers*: parallel condenser water-cooling circuits are sometimes an advantage if more than one cooling tower is to be used. Generally, where a parallel arrangement is used, the compressor power will be greater but the water pump power will be lower, based on a similar argument to that used for the evaporators.

(*d*) *Series counterflow*: although having a higher pressure drop, this arrangement of evaporators and condensers can result in a lower compressor power than either parallel or series arrangements for multiple machine installations, particularly for heat reclaim schemes.

(*e*) *Steam turbine with centrifugal and absorption chillers*: this system is suitable for total-energy installations where steam is available at pressures above 10 bar; it is used to obtain maximum economy in the use of steam to operate refrigeration machines. Steam passes in series through a back-pressure steam turbine driving a centrifugal water chiller and then, at the back-pressure, into a pair of absorption machines, each equal in capacity to the centrifugal unit. The evaporators are connected in series–parallel, the chilled water passing through the pair of parallel connected absorption machines and then through the centrifugal machine. The three condensers are usually, though not necessarily, connected in parallel. Double-effect steam-driven absorption chillers should also be considered (see section 3.3.3.7 and IEA Heat Pump Programme Annex 24: *Absorption Machines for Heating and Cooling in Future Energy Systems* (IEA Heat Pump Centre, 2001) for further details).

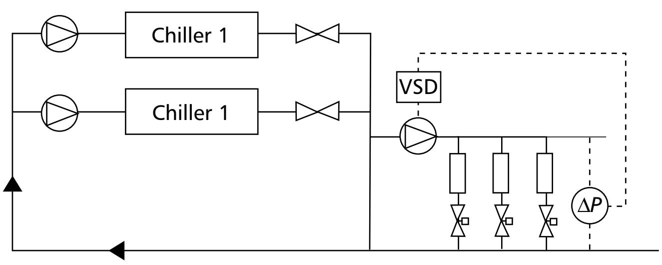


Figure 3.89 Constant primary variable secondary system

Of the above options (*a*) and (*b*) are the most popular, (*c*) and (*d*) are not popular and (*e*) is very seldom used.

3.3.6.6.3 Variable-flow pumping

Where building cooling loads are variable, significant energy savings can be achieved from the use of variable-speed pumps. Figure 3.89 shows a constant primary variable secondary flow system. A conventional decoupled primary and secondary arrangement is used to maintain constant flow rate through the chiller evaporators and provide near failsafe operation.

The variable-speed secondary pump is controlled to maintain constant differential pressure in the system to ensure a minimum flow rate is achieved through any coil. Flow rate through each individual coil is variable according to load by the position of its two-way control valve.

Modern chillers allow variation in the chilled-water flow rate according to the load, as long as a minimum flow rate value is maintained. This allows a variable primary variable secondary flow system to be used, as shown in Figure 3.90.

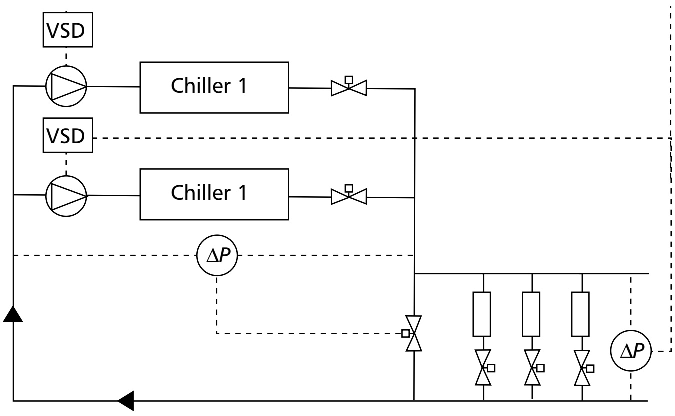


Figure 3.90 Variable primary variable secondary flow system

As well as reducing the number of pumps, this approach can result in even greater energy savings than the constant primary variable secondary flow system (Taylor, 2002; Avery, 2001). The primary pump speed may be controlled according to the pressure drop across the chillers. Alternatively, the primary circuit flow rate may be measured directly using a flow meter, although the designer should be aware of the need to install the flow meter in strict accordance with the manufacturer's recommendations with respect to length of straight pipe upstream and down of the flow meter and the need for regular recalibration.

The disadvantage of the variable primary variable secondary flow system is greater complexity both for the designer and the operator and a greater risk of control failure. Particular care is needed in the control of chiller sequencing, and it is recommended that the designer seeks the advice of the chiller manufacturer. It is recommended that these systems only be installed in buildings with competent on-site staff who have been trained to understand the system design and operation. The effect of variable flow rate on the chiller evaporator water-side heat transfer coefficient also needs to be considered. In general, plate evaporators are less sensitive than shell-and-tube evaporators, which may require design changes to the water-side baffling. The chiller manufacturer should be consulted about the suitability for variable flow rate operation and to confirm the safe minimum flow rate.

3.3.6.7 Fan coil units

Refer to section 3.2.3.3.1 for information.

3.3.6.8 Refrigerant pipework

The design and installation of refrigerant pipework can critically affect the system performance and reliability and requires particular care for split and multi-split systems and any other site-erected system.

— *Liquid-line pressure drop*: should be kept to a minimum to ensure that there is no flash gas at the expansion device, as this can significantly reduce the cooling capacity of the system. In liquid lines, the liquid temperature is close to the refrigerant saturation temperature and the effect of pressure drop is to reduce the temperature difference increasing the risk of flash gas forming at the expansion valve.

— *Suction-and-discharge line pressure drop*: a balance has to be made between the effect on compressor performance and the minimum velocity required for oil return. Where the direction of refrigerant flow is upwards then the velocity must be high enough to entrain the oil. One way of accomplishing this is to use a double riser with oil traps or reduced size risers.

— *Cleanliness*: it is vital that ingress of dirt is avoided, as this could lead to internal blockages, especially in systems with capillary tube expansion devices. To avoid internal oxidisation, dry nitrogen is often purged through pipework during brazing.

— *Refrigerant leakage*: demountable flare and screwed joints should be avoided wherever possible to minimise the risk of leakage. The number of joints should be minimised and where necessary should be brazed using competent personnel following industry good-practice working methods. Pipework should not be installed where mechanical damage is likely. Pipework should be strength and pressure tested before evacuation (see BS EN 378 (BSI, 2008a–d)).

3.3.7 Controls

Appropriate and properly commissioned controls are essential to maintaining the desired levels of performance and safety with good energy efficiency. Guidance on control systems is given by CIBSE Guide H: *Building control systems* (CIBSE, 2009).

3.3.7.1 Capacity controls

3.3.7.1.1 Reciprocating compressors

Capacity control of reciprocating equipment is achieved in one of five ways, as follows.

— *Multi-modular*: several compressors are incorporated in a chiller package, each compressor representing a step of capacity. It is important that the chiller control system is compatible with the compressor safety controls, which should be set to avoid frequent cycling (see also section 3.3.7.4).

— *Cylinder unloading*: several methods are available but it is most common for the suction valve on one or more cylinders to be maintained in a raised position by hydraulic pressure, so allowing the refrigerant gas to pass back and forth without check and thereby reducing the mass flow through the compressor. A minimum gas flow must be maintained to minimise motor overheating and ensure adequate oil return. Insufficient oil return will adversely affect the operation of compressors or contribute to nuisance tripping of the oil safety switch. It is recommended that long hours of operation with unloaded cylinders are avoided.

— *Speed variation*: the output of a reciprocating compressor is directly proportional to the speed of shaft rotation, which may be changed by varying the speed of the prime mover. A certain minimum speed must be maintained for lubrication to be effective. Two-speed compressors have been used in the past but variable speed utilising inverter control is becoming more common. Whilst accepting that the majority of compressors used will be semi-hermetic or hermetic, and thus the responsibility of the manufacturer, it is important to note that multi-speed and inverter control applications must conform to the compressor manufacturers' instructions, otherwise there may be problems with damage to windings during operation. This may be due to fluctuations ('spikes') in the electricity supply or to the compressor power requirement during speed changes. It is important that the designer gives careful consideration to the inverter specification. Electronically commutated (EC) motors are also becoming available in latest compressor technology.

— *Hot gas bypass*: the load on the compressor is maintained while the evaporator capacity is varied. The most effective arrangement is for the hot refrigerant gas to bypass the condenser and inject the refrigerant into the system downstream of the expansion valve and upstream of the evaporator. It should be noted that this method of capacity control offers no energy economies at part load and, depending on method chosen, can result in high compressor discharge temperatures and therefore should be avoided. Extensive operation can cause compressor damage.

— *Evaporator pressure regulator*: this is a means of maintaining the evaporator pressure by throttling the flow of gas to the suction of the compressor. Energy efficiency is compromised but not as much as for hot gas bypass systems. Evaporating temperatures will rise so this is better applied to sensible cooling applications.

Cylinder unloading is becoming less popular with the need to seek effective system part load efficiencies. Speed-variation and multi-compressor circuiting is more economical due to the greater reduction in power consumption at part load operation.

3.3.7.1.2 Centrifugal compressors

In most centrifugal applications, the machine must respond to two basic variables:

— refrigeration load

— entering condensing water/air temperatures.

A centrifugal compressor is, for a given speed, a relatively constant volume device compared with a multi-cylinder reciprocating compressor where cylinders can be deactivated progressively to accommodate load changes.

The control system must be able to alter both the head and flow output of the compressor in response to load changes. This is possible by using one or both of the following methods:

— refrigerant flow control by variable inlet guide vanes

— variable speed control.

Speed control is generally the most efficient method. However, its use is limited to drives whose speed can be economically and efficiently varied and to applications where the discharge pressure (head requirement) falls with a decrease in load (this restriction only applies to centrifugal compressors).

The most generally accepted method of flow control, particularly for hermetic centrifugal compressors, is that of variable inlet guide vanes. The vanes are usually located just before the inlet to the impeller wheel (or first impellor wheel in multi-stage compressors) of the compressor and are controlled by the temperature of the water leaving the evaporator. This method offers good efficiency over a wide range of capacity. At half load condition, for example, the power required may be only 45 per cent of the full load power.

Hot gas bypass is useful to extend the control range of a machine to very low loads, particularly where the system head requirements (condenser pressure) remain high, thereby avoiding compressor surge. Instead of discharging gas into the compressor inlet, which can cause high temperature problems, the hot condenser gas is passed through a pipe and valve to the bottom of the cooler, thereby providing a 'false' load on the cooler. In this manner, the compressor experiences a constant load. However, this technique should not be used continuously but only for occasional part-load conditions.

Designers should be aware that centrifugal compressor manufacturers often quote a time limit for continuous part-load operation.

Surge is caused by flow breakdown in the impeller passageways; the impeller can no longer maintain the required system pressure and a periodic partial or complete flow reversal through the impeller occurs. Surge is characterised by a marked increased in the operating noise level and by wide fluctuations in discharge pressure eventually leading to shutdown. For this reason, it is often not practical to run a centrifugal compressor at part load under high summer condenser temperature conditions. Designers should be aware that for the same cooling capacity, different sizes of compressor have different surge lines.

3.3.7.1.3 Screw compressors

Capacity control is normally obtained by varying the compressor displacement using a sliding valve to retard the point at which compression begins and, at the same time, reducing the size of the discharge port to obtain the desired volume ratio (see also section 3.3.6.4). This typically allows 10 per cent to 100 per cent capacity control although below 60 per cent of full load the compressor efficiency is very low. Variable motor speed control using an inverter is also increasingly used and at low loads offers higher efficiency than the slide valve method. Another form of capacity control is the use of multiple screw compressors.

3.3.7.1.4 Scroll compressors

Capacity control can be obtained using two-speed motors or multiple compressors.

Electronic capacity modulating systems that momentarily separate the scrolls axially and can provide between 10 per cent and 100 per cent capacity variation have been available for a number of years. Because the shaft continues to rotate at full speed, the compressor lubrication system is not affected at part load whilst the compressor efficiency is compromised.

Some manufacturers also offer variable motor speed scroll compressors designed specifically to ensure lubrication over a speed range.

3.3.7.2 Capacity controls for absorption chillers

The capacity of centrifugal compressors is controlled by regulating the amount of heat supplied to the generator as hot water, steam or natural gas. This varies the ability of the solution to absorb the refrigerant and therefore the evaporation rate in the evaporator.

Some manufacturers also offer a variable flow solution pump, which can significantly improve CoP at part load.

3.3.7.3 Operational controls

The inclusion of a building management system provides the designer with a number of additional ways to maximise the operating efficiency of the refrigeration plant by precise control of the plant items to exactly match the system requirements. One such example is the ability to vary the chilled-water flow temperature or flow rate to match exactly the cooling requirements of the system, rather than allowing the plant to control to a single set-point temperature. To maximise the benefits of the controls system, it is important that the control system communicates correctly with the refrigeration plant and vice versa. Failure to address this at the design stage can result in problems with final commissioning on site or, at worst, the controls system failing to control the refrigeration plant to the level specified by the designer.

Control methods that may improve the energy efficiency of refrigeration plant include the following:

— *Variable set-point temperature on chilled-water systems*: at periods of low cooling load it might be possible to raise the chilled water set-point temperature to a value higher than the normal value of around 6 °C. This simple measure could be controlled by a building management system (BMS).The flow temperature is slowly decreased until the space cooling requirement is matched, usually monitored by the position of the cooling valves on the cooling coils. If any valve is 100 per cent open then demand is not met in all areas; if all are below 100 per cent open then demand is exceeded and chilled-water flow temperature need not be reduced any further. The compatibility of this particular method with the chiller should be confirmed with the chiller manufacturer.

— *Staging the operation of multiple refrigeration plant to meet the required demand*: this depends on the system design, e.g. for example, with constant speed primary pumping it is generally more efficient to run one chiller at full load than two at partial load (see CIBSE Guide H: *Building control systems* (CIBSE, 2009)).

— *Variable chilled-water flow rate, either on the secondary side only or on the primary and secondary sides*: further details of these techniques are given in section 3.3.6.6. The suitability of primary chilled-water flow rate must be confirmed with the chiller manufacturer. This approach allows the designer to keep the pump power to a minimum value.

— *Reduced condenser water temperature*: the condenser water temperature should be kept as low as possible, although it is essential that the chiller manufacturer's specifications on minimum temperature values are adhered to.

3.3.7.4 Safety devices

It is essential that safety devices are not to operate the plant under normal conditions. Safety devices are provided to ensure that, in the event of a fault developing, the plant shuts down in such a way that there is no risk of injury to personnel, and equipment is protected from damage. Where particular operational conditions may result in frequent recycling, safety devices should be of the manual reset type. Table 3.34 lists types of safety devices and their function.

Although safety devices are usually dealt with by the equipment manufacturer, the designer should ensure that the provisions of BS EN 378 (BSI, 2008a–d) and the IOR Safety Codes (IoR, 2009c/d, 2013, 2015a), are complied with, e.g. refrigerant pressure relief devices should discharge to a safe place, and all safety cut-outs and switches should be tested during commissioning and a professional maintenance programme. It is recommended that, if the compressor/machine is fitted with capacity control, these tests be carried out with the compressor/machine at minimum capacity.

3.3.8 Commissioning, operation and energy management

3.3.8.1 Commissioning

The process of commissioning is to ensure that the equipment is performing correctly and to specification. In addition to checking functionality, there should be a performance inspection at least to the level required by the Energy Performance of Buildings Directive (EPBD) (inspection of air conditioning systems) (EC, 2010). If the equipment cannot be operated at design conditions because of insufficient load or low ambient temperatures, the commissioning process must be extended over a period of time before handover.

Key requirements for successfully commissioning of a refrigeration and heat rejection system are provided in CIBSE Commissioning Code R (CIBSE, 2002) and sections 6.3 and 6.4 of BS EN 378-2: 2008 (BSI, 2008b).

The appointment of a specialist commissioning manager is recommended for all installations.

The plant installer should ensure that the electrical power supply circuit protection and cabling have been tested in advance of applying power to the refrigeration plant. Some plant may require the power supply to be connected for a period of time in advance of commissioning to ensure (e.g. to heat the compressor oil).

Table 3.34 Type and function of safety devices

Safety device	Function
Mechanical refrigeration:	
— high refrigerant pressure cut-out	Breaks circuit on excessive refrigerant pressure rise
— low refrigerant pressure cut-out	Breaks circuit on fall in refrigerant pressure
— low oil pressure cut-out	Protects against failure of lubricating system
— high oil temperature cut-out	Protects against failure of lubricating system or if bearing failure occurs
— low refrigerant temperature cut-out	Protects against low evaporating temperatures
Electronic temperature sensor	
— pressure relief device	Protects against high refrigerant pressure (static)
— low water temperature cut-out	Protects against evaporator freezing (in water chillers)
— flow switches	Protects against reduced fluid flow through evaporator or condenser
Absorption refrigeration:	
— low refrigerant temperature cut-out	Protects against evaporator freezing
— low chilled water temperature cut-out	Protects against evaporator freezing
— high solution temperature cut-out	Protects against over-concentration of the solution and consequent crystallisation
— low cooling water temperature cut-out	Protects against over-concentration of the solution and consequent crystallisation
— low switches	Protects against reduced fluid flow through evaporator or condenser

The mechanical connections to refrigeration and heat rejection plant may consist of:

— refrigerant pipework
— chilled-water pipework
— condenser-water pipework.

Refrigerant pipework should be clean throughout the installation process. Some refrigerant pipework may require non-destructive testing (NDT) to demonstrate that the appropriate quality of welding has been achieved (particularly ammonia pipework). All refrigerant pipework should be strength and tightness tested in accordance with the requirements of BS EN 378 (BSI, 2008a–d). Refrigerant pipework should be evacuated to dehydrate in advance of charging the system and bringing into operation.

Chilled and condenser-water pipework should incorporate strainers and flow measurement and control devices in accordance with manufacturer requirements. Pipework should be NDT and pressure tested as appropriate to the installation and flushed and cleaned before being brought into service.

All pipework requiring insulation should be insulated in advance of being brought into service.

The local authority should be notified of cooling towers, which should be flushed and cleaned and provided with an automated water treatment system in advance of being brought into operation.

All control interfaces should be connected and tested as part of commissioning.

The commissioning of refrigeration plant often requires a dual approach.

Where the refrigerant plant has chilled or condenser-water connections it is important to ensure that the correct flows are achieved prior to attempting to operate the refrigeration plant.

A specialist refrigeration commissioning engineer (ideally a representative of the system manufacturer, installer or their agent) should carry out the specialist commissioning of the refrigeration plant.

Commissioning of the refrigeration plant should include:

— testing of safety devices
— testing of control devices
— testing of sequence control routines for multiple pump and chiller installations.
— validation of interfaces with the building management systems (sequence control, fault and status reporting).

A full commissioning report should be provided for each item of refrigeration plant including chilled and condenser water flows, verification of safety and control tests, refrigerant charge, operating status of main components, operating temperatures and pressures for the water and refrigerant circuits.

Commissioning should also include training of the site engineering support in the O&M of the refrigeration plant and the associated chilled and condenser-water systems as applicable.

The final commissioning report should be signed off by the operator.

3.3.8.2 Logging and inspection

Regular logging is vital in determining early signs of performance deterioration and hence prevention of breakdown. The log should consist of the following measurements, where applicable, as an absolute minimum to be taken and recorded when the plant is operating in stable manner, as far as can be achieved:

— ambient conditions, dry and wet bulb temperatures
— refrigerant temperatures at expansion valve inlet and evaporator outlet
— refrigerant pressures and temperatures at compressor suction and discharge
— secondary fluid temperatures at heat exchanger inlet and outlets
— pump, fan and filter pressures
— setting of all adjustable controls
— electric motor currents.

3.3.8.3 Operation and maintenance (O&M)

The building owner should be provided with a logbook, in which the installed plant and its function are described in simple language for the everyday use of the owner. The logbook should be in addition to the more detailed information provided in the O&M manuals and the health and safety file.

Information that should be provided in the building logbook includes:

— a description of the building and its building services systems, including intended purpose
— a simple description of the operation and control strategies of the building services systems
— the location(s) of relevant plant and simplified system schematics
— input power and output ratings of the services plant
— a summary of the commissioning report
— maintenance instructions
— details of sub-meters and advice on their use
— results of airtightness testing.

A suitable template for the production of the building logbook is available as CIBSE TM31: *Building log books* (CIBSE, 2003).

The operational efficiency of the system depends to a certain degree on the ability and commitment of the end user. This is because the end user is ultimately responsible for such items as the implementation of a planned preventative maintenance scheme and monitoring the system for faults or failures of plant or system components. For example, the correct setting of time clocks or other controls with respect to the occupancy or process time periods will contribute significantly to overall efficiency.

It is therefore important to ensure that the relevant responsible person for the end user understands the system and is also made aware of their responsibilities with regard to the operation of the plant. This is now a requirement of Building Regulations Approved Document L2A (NBS, 2013a), which requires that the owner and/or operator of the building is provided with a logbook giving details of the installed building services and controls, their method of O&M and other details that show how energy consumption can be monitored and controlled. This information should be provided in summary form, suitable for day-to-day use and should be in addition to the more detailed information provided as part of the operation and maintenance manuals and health and safety file.

CIBSE TM31: *Building log books* (CIBSE, 2003) provides specific guidance on the preparation of building logbooks.

The need for specific user training should be considered in cases where the plant and systems are particularly complex or unusual.

Building Regulations Approved Document L2A (NBS, 2013a) also requires that sufficient sub-metering is provided so that the owner and/or operator can monitor and control energy use (see section 3.3.1.2.1).

It is the designer's responsibility under the Construction (Design and Management) Regulations 2007 (TSO, 2007b) to ensure that future maintenance of the plant can be carried out safely.

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Appendix 3.A1: Techniques for assessment of ventilation

3.A1.1 General

There are a number of assessment techniques available to calculate ventilation and cooling requirements and to look in detail at air movement. This appendix provides an overview of some of these techniques. CIBSE AM11 (2015b) and AM10 (2005) provide more detail on dynamic thermal simulation and assessment techniques for natural ventilation respectively.

3.A1.2 Ventilation and cooling

Chapter 1 of CIBSE Guide A (2015a) includes air flow rate requirements for ventilation purposes. Air flow rate requirements for cooling purposes are normally based either on restricting peak summer temperatures in passive buildings or to meet the peak cooling load. Analyses may start by looking at peak temperatures to evaluate of the building's potential without mechanical cooling. These would assess the ventilation rates (natural and/or mechanical) and passive measures needed to meet summer temperature limits. Where cooling is to be provided, the cooling needed to maintain the temperature limits would be assessed. Air flow rates may then be calculated to deliver this cooling to the space.

There is a range of analysis methods available to suit different applications and stages in the design process. Design charts based on parametric analysis may be used (e.g. the BRE's *Environmental Design Guide* (Rennie and Parand, 1998)), although the user can work only within the range of variables covered by the charts. Chapter 5 of CIBSE Guide A (2015a) provides design information on the use of thermal dynamic models for calculating peak summertime temperatures and peak space cooling loads. Simple (dynamic) models (e.g. the admittance procedure) may be used to assess cooling loads and the probability of overheating. These approaches are based on a 24-hour design cycle and are suitable for mechanically cooled buildings with a repetitive diurnal operating cycle. However, where this is not an accurate reflection of building operation due to thermal mass or passive operation, dynamic thermal simulation may be used.

Appropriate consideration should be given to issues of weather data, control and thermal mass depending on the application. Selection of appropriate weather data is discussed in CIBSE AM10 (2005). Different data will be required for different purposes. For example, to estimate energy consumption, average weather data for the region will usually be the most appropriate. Data, including more extreme conditions, will be appropriate to test the ability of the building to accommodate various levels of internal heat gain and predict peak temperatures. Site-specific weather data can be of interest, but may have been collected over a relatively short period and may not necessarily be representative. It is frequently impossible to use such data to construct meaningful statistics to identify the percentage of time a specified internal temperature would be likely to be exceeded. There is also a danger that the design may lack robustness, being tailored to a unique weather sequence and reacting in a different and unpredicted way to more normal weather peaks. A more robust choice will often be made in order to analyse the building in relation to appropriate national UK data and to make simple corrections to suit the differences between this and the site data: e.g. August average temperature and diurnal swing and August 2.5% exceeded peak temperature and the associated diurnal swing.

Loads and system performance often depend on more than one weather variable. Cooling and humidity conditions will be a function of wet bulb as well as dry bulb temperature. The performance of natural ventilation systems in particular can be affected by solar and wind conditions as well as temperatures, as these are often used to drive the ventilation. Design conditions for the individual weather variables will rarely coincide.

Controls used in the thermal model should reflect what can be expected to occur in practice. This is a particular issue in natural ventilation systems with manual control. Account should be taken of the way occupants use windows. Data are available on occupancy effects on natural ventilation, primarily based on the domestic sector. This work is summarised in AIVC Technical Note 23 (Dubrul C, 1988).

Thermal mass should be modelled with appropriate surface heat transfer values and representation of heat flow within the mass (see 'Thermal storage performance', page 3-42). High thermal mass buildings must be allowed to come to their natural thermal equilibrium by having a lengthy period of simulation prior to the period over which the modelling results are reported and compared; 15 days is usually enough for this 'pre-conditioning' period, although a few buildings require longer. This can be tried first with 10 and 20 days and the results compared to check for significant differences. If a hot spell is being simulated, peak weather data should not be used throughout, as this will under-value the heat-absorbing benefits of the thermal mass. Instead, pre-conditioning with average weather for the season concerned can be undertaken, followed by a step change to the peak weather sequence — which in the UK seldom lasts more than 5 days. The design day is typically the third in the peak weather sequence.

3.A1.3 Air movement

Analyses of air movement may be needed, particularly for natural ventilation applications and air movement in large spaces such as atria. These provide information on air velocity, movement and temperature; volume flow rate; and optimal opening sizes, shapes and positions. Techniques available include computational fluid dynamics (cfd), physical models and air flow models. For room air distribution, performance is sometimes critically dependent on details of equipment design, and full-scale mock-ups may be required.

3.A1.3.1 Computational fluid dynamics (CFD)

CFD is a technique for predicting air movement that can address questions such as stratification and local air movement. It therefore has particular application to consideration of large spaces such as atria. CFD methods can predict temperature and velocity distributions in a space and can be applied to assessments of comfort involving more of the influencing parameters than is possible in zonal models. Because of the extensive nature of the computations and the time varying nature of the natural driving forces, CFD is normally only used to generate 'snapshots' of how the design would work at a given point in time.

Another potential application for CFD is external flows around the building. The purpose is to generate the wind pressure coefficients needed by all models to predict natural airflow rates.

3.A1.3.2 Physical models

Physical models are especially useful for giving the non-technical members of the client and design team a good visualisation of airflow behaviour. By their nature, physical models are implicit design tools; assumptions need to be made then tested. The two main techniques relating to natural ventilation design are the salt bath technique and wind tunnel testing.

Salt bath

The salt bath technique is used to test stack driven ventilation strategies. Stack-driven flows are analysed at small scale in the laboratory using a model of the building immersed in a perspex bath containing saline solutions of different concentrations. The method models fluid flow, not surface heat transfer, and therefore cannot predict local effects such as solar patching on the floor of an atrium. Like the CFD technique it provides only a snapshot of performance.

Wind tunnel

Wind tunnel testing is the main source of information on wind pressure coefficients. It is not a method for proving the design of a natural ventilation system, since it only deals with external flows around a building.

Air flow models

Air flow models may be used to analyse natural ventilation air flow rates based on driving pressure differences and openings. These range from single zone models to more complex multi-zone models. Single zone models (Liddament, 1996) are appropriate where the building is open plan and there is no temperature stratification in the space. Building types that approximate to these requirements are dwellings, many industrial buildings and small open plan office buildings. Multi-zone models subdivide the building into a number of individual spaces, substantially increasing the complexity of the analysis (Feustel and Dieris, 1991).

Software combining multi-zone flow models with thermal simulation analysis is also available. This software can provide an integrated analysis the internal temperature distribution and the stack induced natural ventilation flow rates (Kendrick, 1993).

Appendix 3.A2: Psychrometric processes

The following table illustrates the basic psychrometric processes and lists the equipment concerned. See section 3.2.4 for details of the various items of equipment.

Table 3.A2.1 Basic psychrometric processes

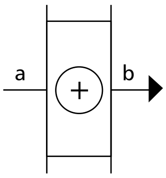
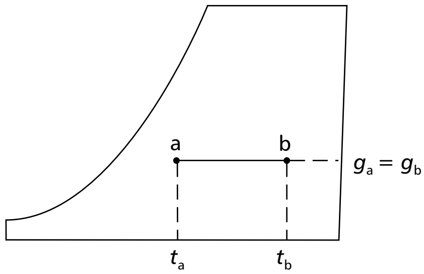
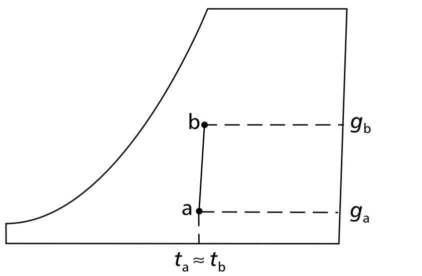
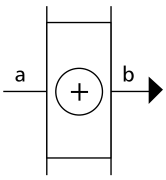
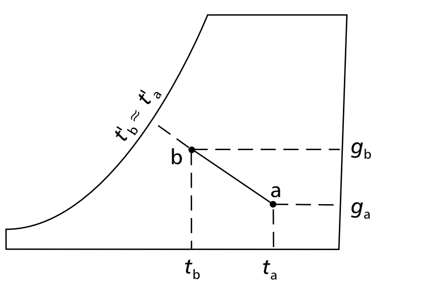
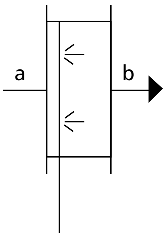
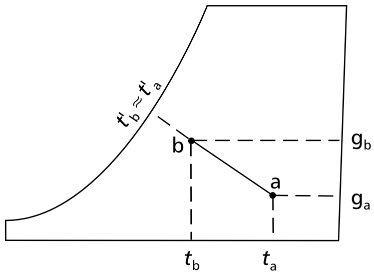
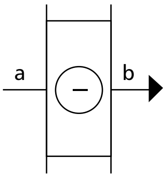
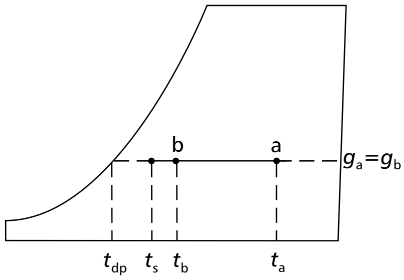
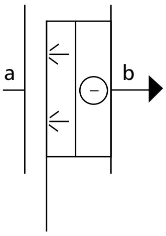
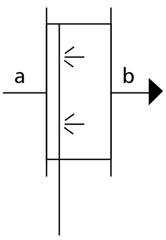
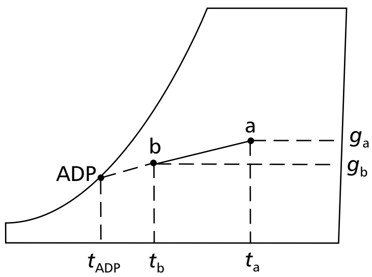
Process	Method	Remarks	Psychrometric process
Heating	Electric	No additional plant required. High energy costs. Wiring and switch gear costs high for large duties. Usually only step control available.	
	Steam	Small heat transfer surface. Plant cost high unless steam required for other services. Condensate return can present difficulties. Modulating control available (2-way valve).	
	Hot water	Simple and reasonably cheap plant and distribution system. Integrates well with other heating systems. Some simplicity sacrificed to decrease heat surface with HTHW. Modulating control available (2- or 3-way valve).	
	Direct firing	Least expensive in specific cases. Can involve problems of combustion air and flue requirements. On/off control is common for smaller units while high/low flame is usually available for larger units.	
Humidification	Steam injection	Electrically heated, self-contained unit or unit supplied by mains steam. Water treatment advisable. Small space occupied. Mains units have modulating control (2-way valve), electric units are normally on/off. Mains units may require condensate drain.	
	Water injection	Involves atomising process (spinning disc, compressed air etc.). Some types are non-recirculatory and require drainage. Air is sensibly cooled as water evaporates. Contaminants from untreated water will enter airstream. Water treatment including biocidal control is essential. Space occupied depends on type. Some units mount on duct wall, other in duct line. Control is usually on/off by stopping atomiser or water supply; larger units in multiple form may be stepped. Normally modulation is not recommended unless water flow is large.	
	Spray washer	Bulky equipment requiring good access to tray and sprays. Also dehumidifies if supplied with chilled water (see Cooling — Air washer). Air sensibly cooled as water evaporates unless water is heated (not normal). Requires water treatment (including biocidal control) and bleed and recirculating pump. Removes both gaseous and particulate air contaminants but with low efficiency. Control indirect by modulation of inlet air condition (pre-heater or mixing dampers) or by by-pass and mixing. Saturation efficiencies range from approximately 70% for one bank facing upstream, to 85–90% for two banks opposed. Water quantity per bank is of the order of 0.4 litre·s^{-1} per m^3·s^{-1} of air flow. Air velocity is of the order of 2.5 m·s^{-1}.	
	Capillary washer	Similar to spray washer but less bulky and provides better air filtering. Has smaller cooling capacity than spray washer when used with chilled water. May require addition of cooling coil. Filtration efficiency is good.	

Table continues

Table 3.A2.1 Basic psychrometric processes — *continued*

Process	Method	Remarks	Psychrometric process
Humidification 	Sprayed cooling coil (not subject to refrigeration)	Utilises cooling coil as wetted pack for humidifying. Action as washer but sprays less prone to blocking. Eliminators not required unless air velocity to high. Requires more space than non-sprayed coil but less space than washer. Water treatment advisable, bleed essential (see cooling coil). Control as for spray. Can be used to cool coil water circuit with low air on temperature, thus making t'_b greater than t'_a. This is sometimes used in an induction system primary plant. Saturation efficiency is of the order of 0.5–1.0 litre·s^{-1} per m^3·s^{-1} of air flow. Air velocity is of the order of 2.5 m·s^{-1}.	
Cooling 	Indirect cooling coil	Supplied with chilled water or brine (usually 2 or 3 °C below apparatus dew-point required). As water is in closed circuit (except for head tank) there is no water contamination from air or evaporation. Contact factor depends on number of rows of pipes deep. Chilled water enters at air off-side. Drain is required. Control by modulating water temperature or flow rate (3-way valve). Normal to keep constant flow rate through chiller.	
	Direct cooling coil (direct expansion coil)	Coil is evaporator of refrigeration circuit. May be cheaper overall than indirect system, but usually involves refrigerant circuit site work. Control by steps, or modulated, depending on refrigeration system. May need special circuitry. Drain is required. Complex and costly for larger installations. May be excluded by local legislation for some applications.	
	Sprayed cooling coil (subject to refrigeration)	With spray off, coil operates exactly as cooling coil. Spray sometimes used to increase surface in contact with air, results in larger contact factor. Saturation efficiency of the order of 80–90%. Water quantity of the order of 0.5–1.0 litre·s^{-1} per m^3·s^{-1} of airflow. Air velocity of the order of 2.5 m·s^{-1}.	(*a*) Sensible cooling
	Air washer (spray washer)	See general remarks on Humidification — Spray washer. Sprays supplied with chilled water, which is liable to contamination through air washing and evaporation if also humidifying. Use with normal, non-cleanable direct expansion chiller not recommended. Overflow required. Contact factor determined by spray design and number of banks. Control by change of spray water temperature (diverting chilled water back to chiller). Saturation efficiencies range from approximately 70% for one bank facing upstream to 85–90% for two banks opposed. Water quantity per bank is of the order of 0.4 litre·s^{-1} per m^3·s^{-1} of air. Air velocity is of the order of 2.5 m·s^{-1}.	 (*b*) Cooling and dehumidifying

Appendix 3.A3: Summary data for refrigerants

Safety group	Refrigerant number	Description (composition by % weight)	Chemical formula	Practical limit / $kg \cdot m^{-3}$	Flammability (lower limit), concentration in air		GWP	ODP
					/ $kg \cdot m^{-3}$	/ % (vol.)		
A1	R22	HCFC	$CHClF_2$	0.3	N/A	N/A	1700	0.055
A1	R125	HFC	CF_3CHF_2	0.39	N/A	N/A	3400	0
A1	R134a	HFC	CF_3CH_2F	0.25	N/A	N/A	1300	0
A2L*	R32	HFC	CH_2F_2	0.298	0.307	14	550	0
A1	R404A	R125/143a/134a (44/52/4)	$CF_3CHF_2 + CF_3CH_3 + CF_3CH_2F$	0.48	N/A	N/A	3800	0
A1	R407C	R32/125/134a (23/25/52)	$CH_2F_2 + CF_3CHF_2 + CF_3CH_2F$	0.31	N/A	N/A	1600	0
A1	R407F	R32/125/134a (30/30/40)	$CH_2F_2 + CF_3CHF_2 + CF_3CH_2F$	0.32	N/A	N/A	1800	0
A1	R410A	R32/125 (50/50)	$CH_2F_2 + CF_3CHF_2$	0.44	N/A	N/A	1980	0
A2	R413A	R134a/218/600a (88/9/3)	$CF_3CH_2F + C_3F_8 + CH(CH_3)_3$	0.08	N/A	N/A	1920	0
A1	R417A	R125/134a/600a (46.5/50/3.5)	$CF_3CHF_2 + CF_3CH_2F + CH_3CH_2CH_2CH_3$	0.15	N/A	N/A	1950	0
B1	R123	HCFC	CF_3CHCl_2	0.1	N/A	N/A	120	0.02
B2	R717	Ammonia	NH_3	0.00035	0.104	15	0	0
A1	R744	Carbon dioxide	CO_2	0.1	N/A	N/A	1	0
A3	R290	Propane	$CH_3CH_2CH_3$	0.008	0.038	2.1	3	0
A3	R600	Butane	$CH_3CH_2CH_2CH_3$	0.008	0.048	1.5	3	0
A3	R600a	Isobutane	$CH(CH_3)_3$	0.008	0.043	1.8	3	0
A2L*	R1234yf †	HFO	$CF_3CF{=}CH_2$	0.06	0.299	6.2	4	0
A2L*	R1234ze †	HFO	$CF_3CH{=}CHF$	—	‡	5.8	7	0
A3	R1270	Propylene	C_3H_6	0.008	0.043	2.5	3	0

* The anticipated A2L mildly flammable category is not listed in BS EN 378-1: 2008 + A2: 2012. The A2L category has been listed by ASHRAE.

† These refrigerants are not listed in BS EN 378-1: 2008 + A2: 2012. Some refrigerant suppliers' information is given.

‡ R1234ze does not exhibit flame limits under standard test conditions, but does so at temperatures above 30 °C. The lower flammability limit stated is at 60 °C.

Appendix 3.A4: Pressure–enthalpy charts for refrigerants

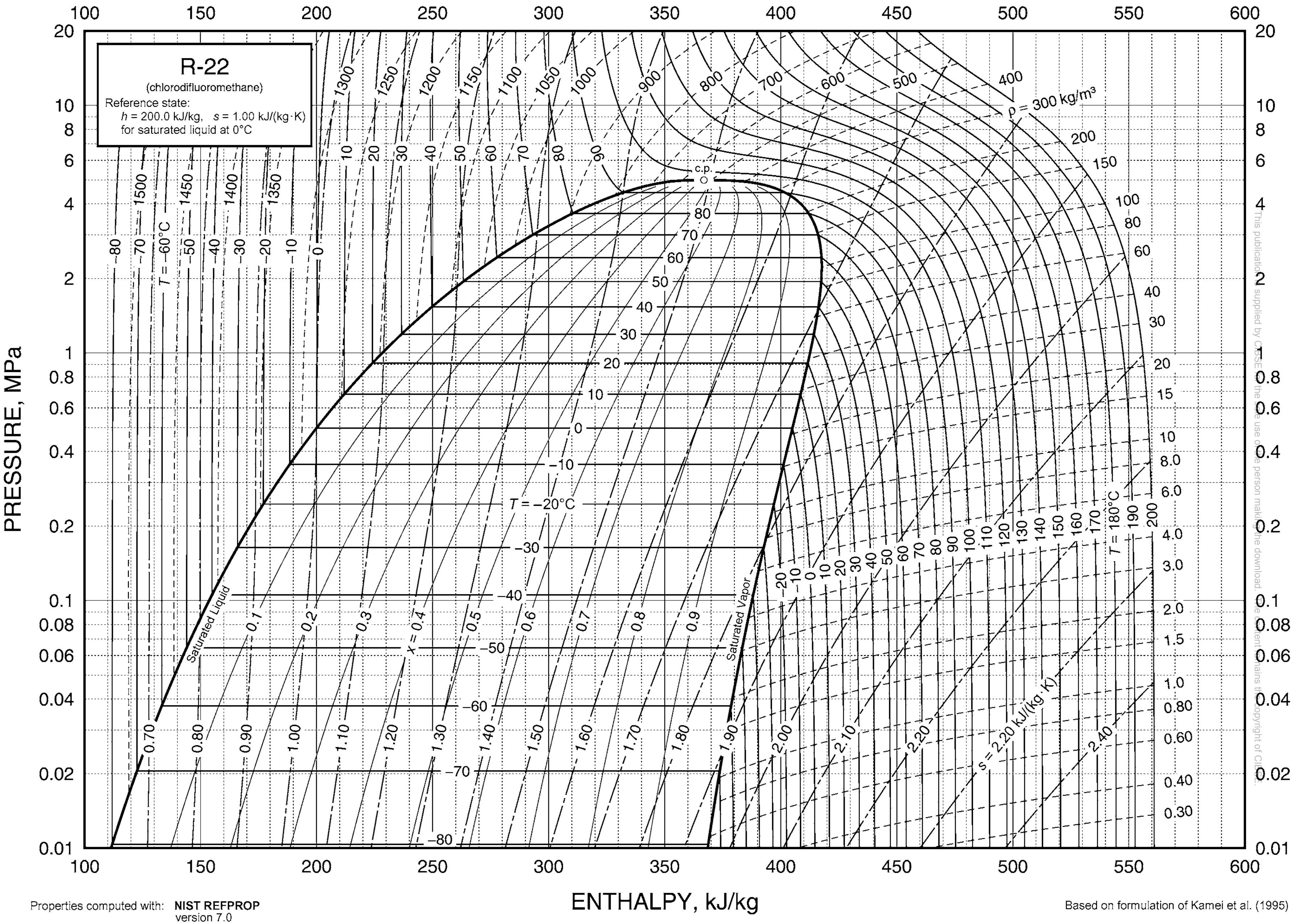


Figure 3.A2.1 Pressure–enthalpy chart for R22 (reproduced from ASHRAE Handbook: *Fundamentals* (2013) by kind permission of ASHRAE)

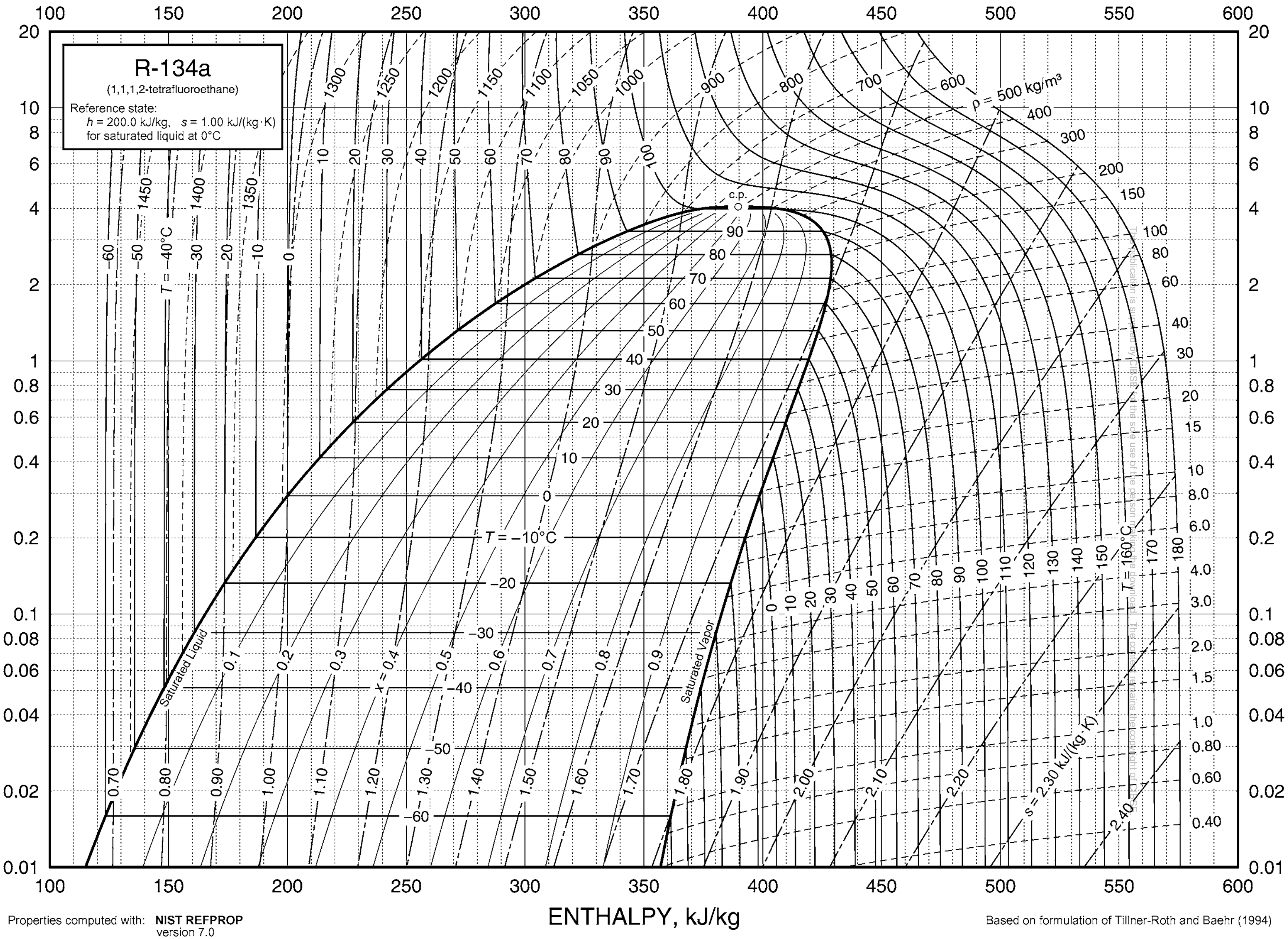


Figure 3.A2.2 Pressure–enthalpy chart for R134a (reproduced from ASHRAE Handbook: *Fundamentals* (2013) by kind permission of ASHRAE)

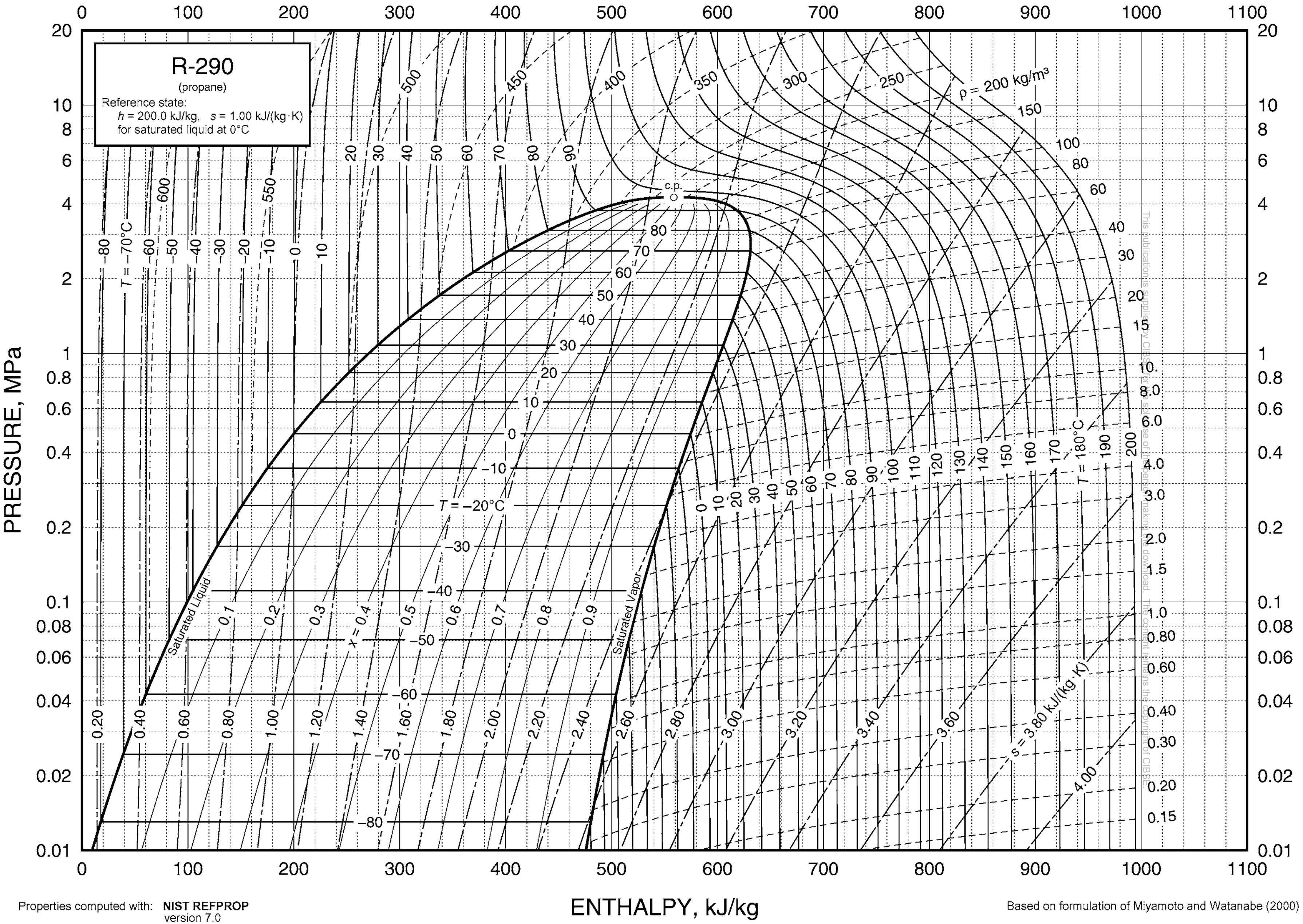


Figure 3.A2.3 Pressure–enthalpy chart for R290 (propane) (reproduced from ASHRAE Handbook: *Fundamentals* (2013) by kind permission of ASHRAE)

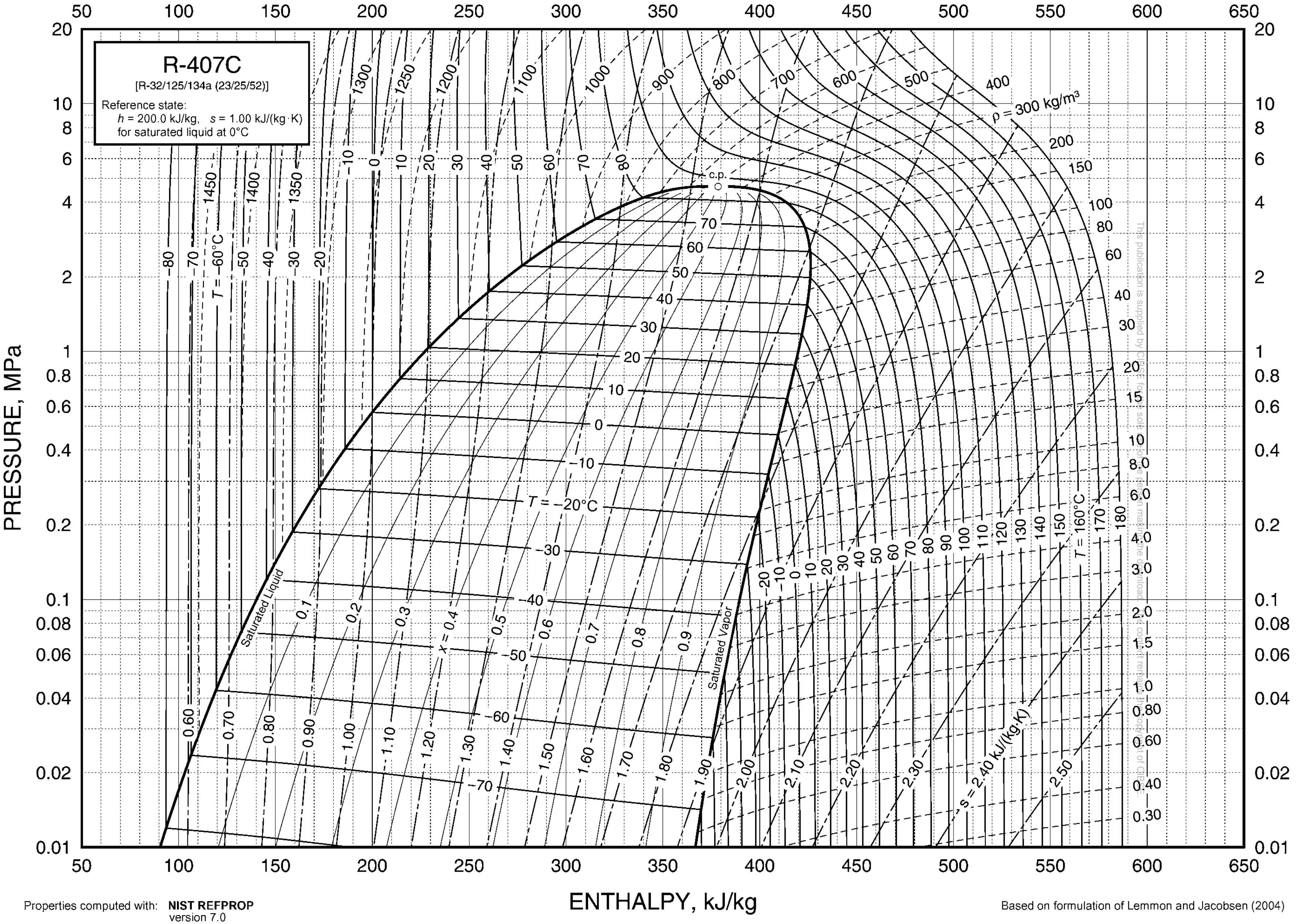


Figure 3.A2.4 Pressure–enthalpy chart for R407C (reproduced from ASHRAE Handbook: *Fundamentals* (2013) by kind permission of ASHRAE)

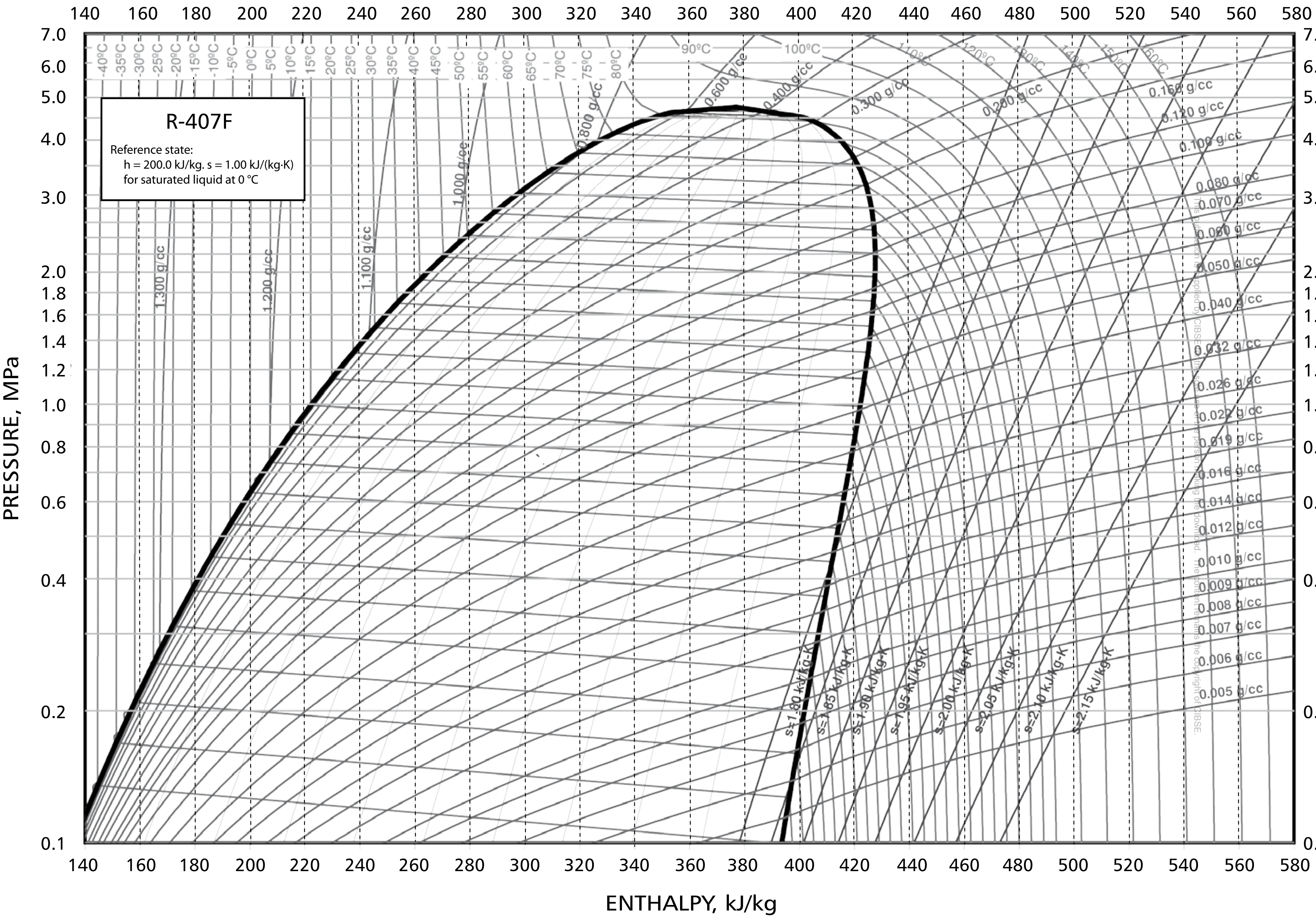


Figure 3.A2.5 Pressure–enthalpy chart for R407F (reproduced courtesy of Honeywell Refrigerants)

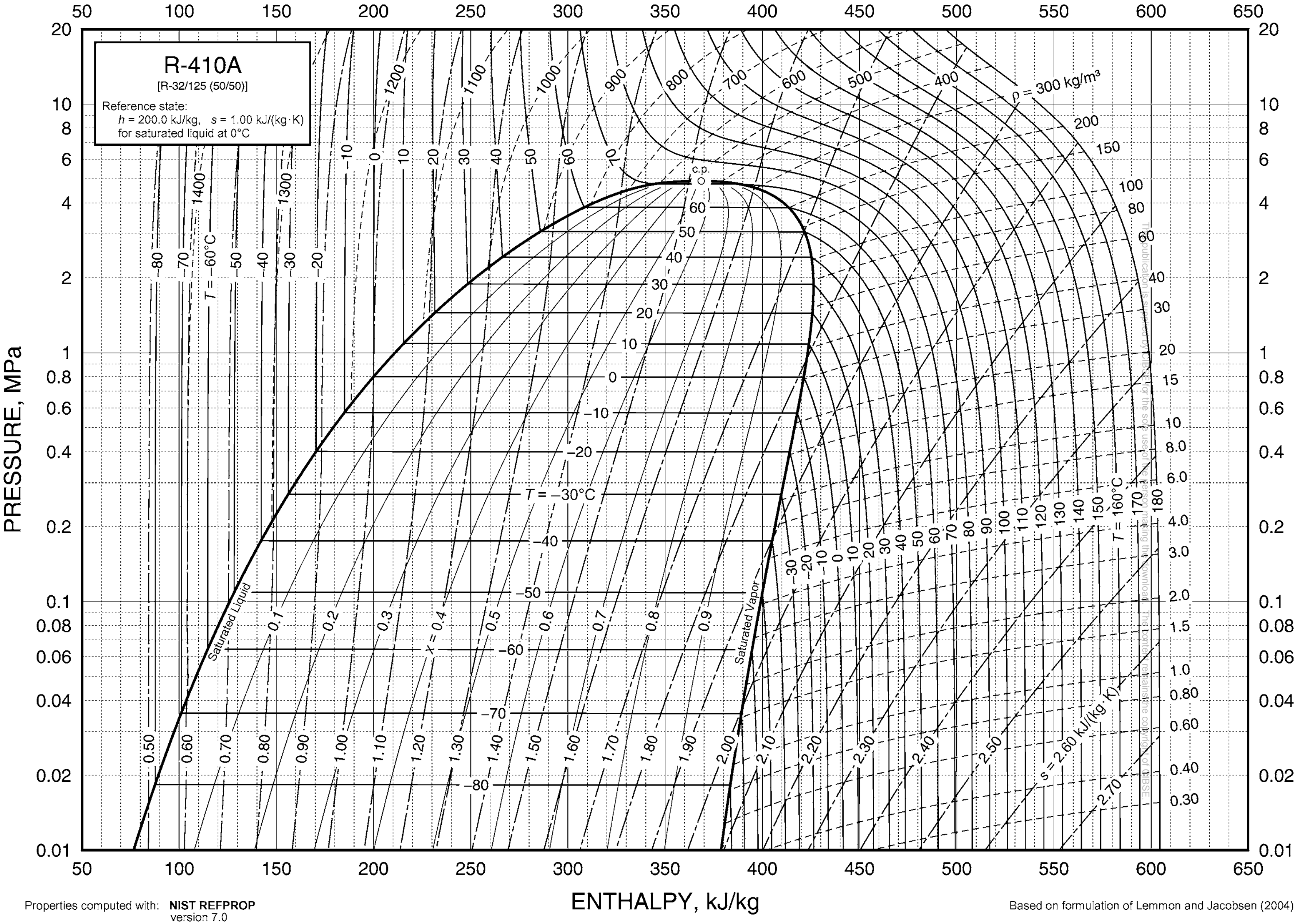


Figure 3.A2.6 Pressure–enthalpy chart for R410A (reproduced from ASHRAE Handbook: *Fundamentals* (2013) by kind permission of ASHRAE)

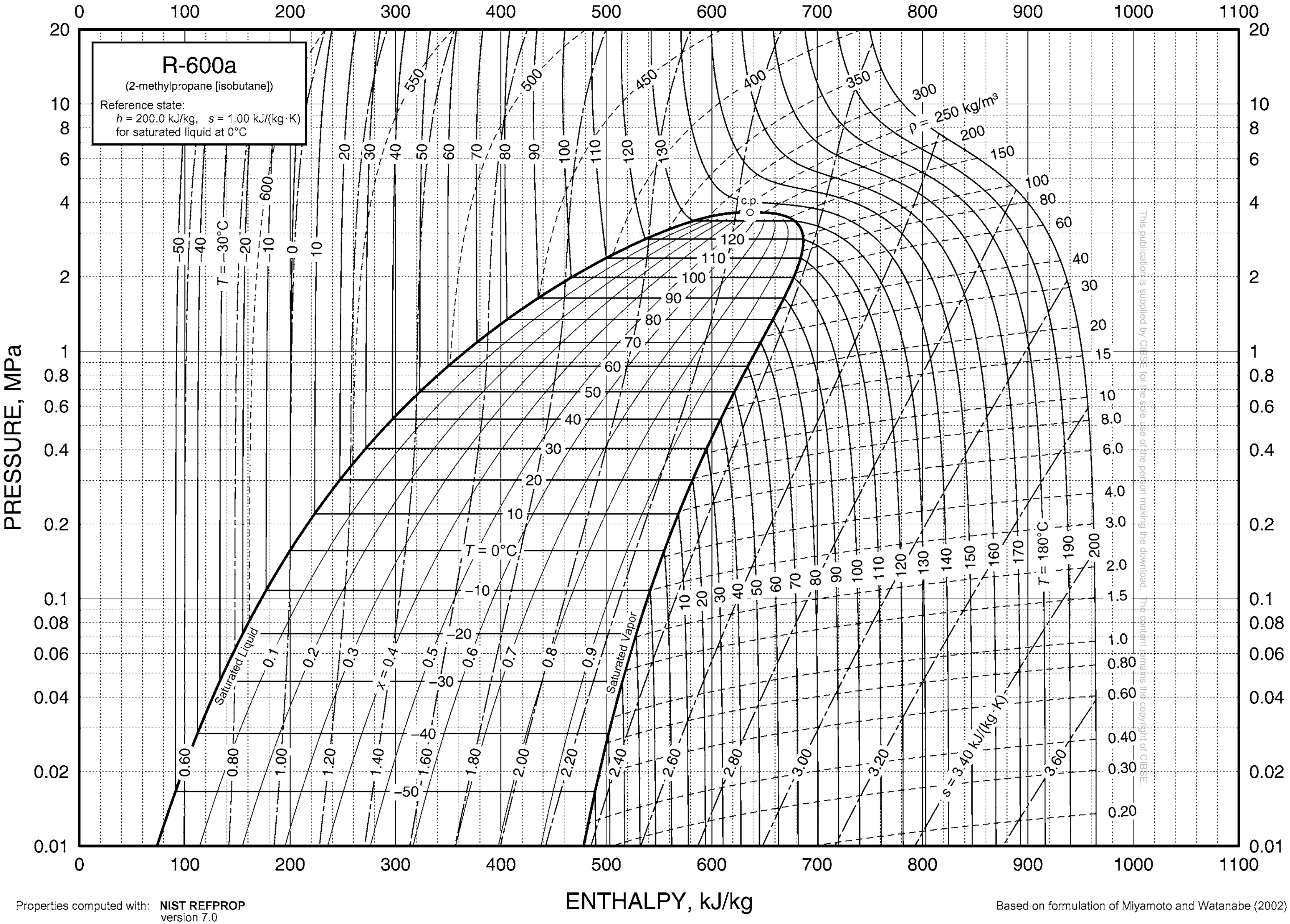


Figure 3.A2.7 Pressure–enthalpy chart for R600a (isobutane) (reproduced from ASHRAE Handbook: *Fundamentals* (2013) by kind permission of ASHRAE)

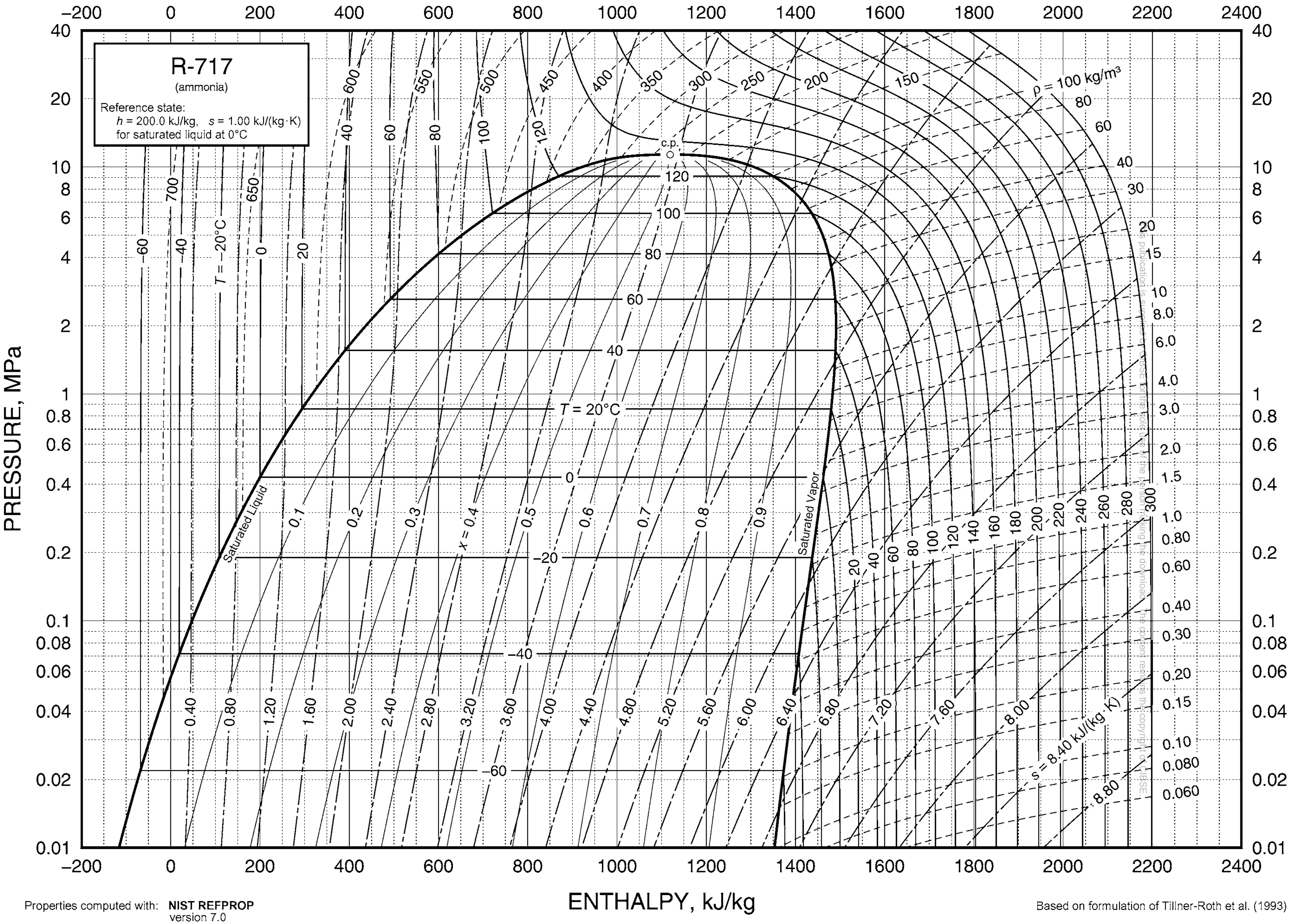


Figure 3.A2.8 Pressure–enthalpy chart for R717 (reproduced from ASHRAE Handbook: *Fundamentals* (2013) by kind permission of ASHRAE)

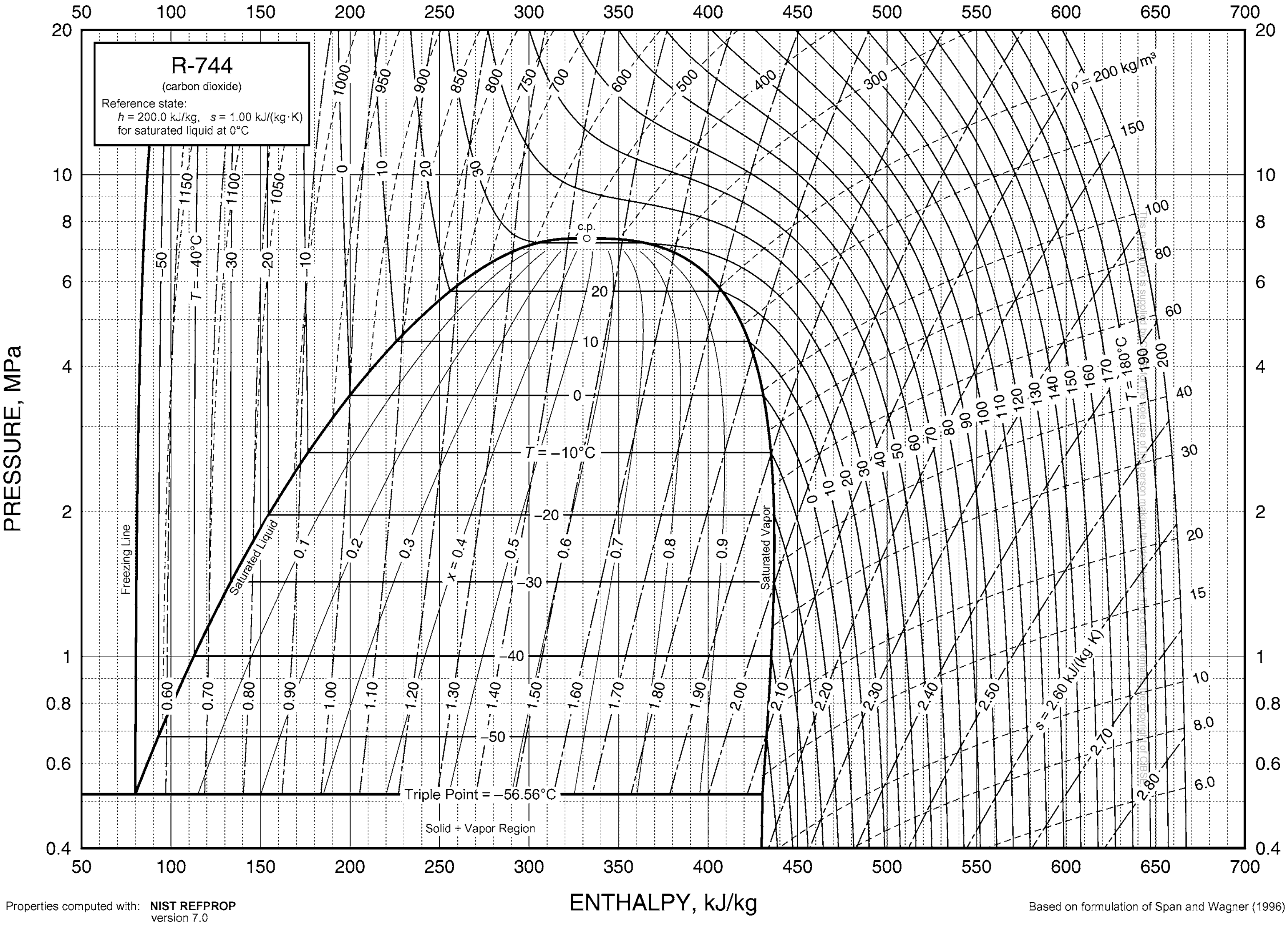


Figure 3.A2.9 Pressure–enthalpy chart for R744 (reproduced from ASHRAE Handbook: *Fundamentals* (2013) by kind permission of ASHRAE)

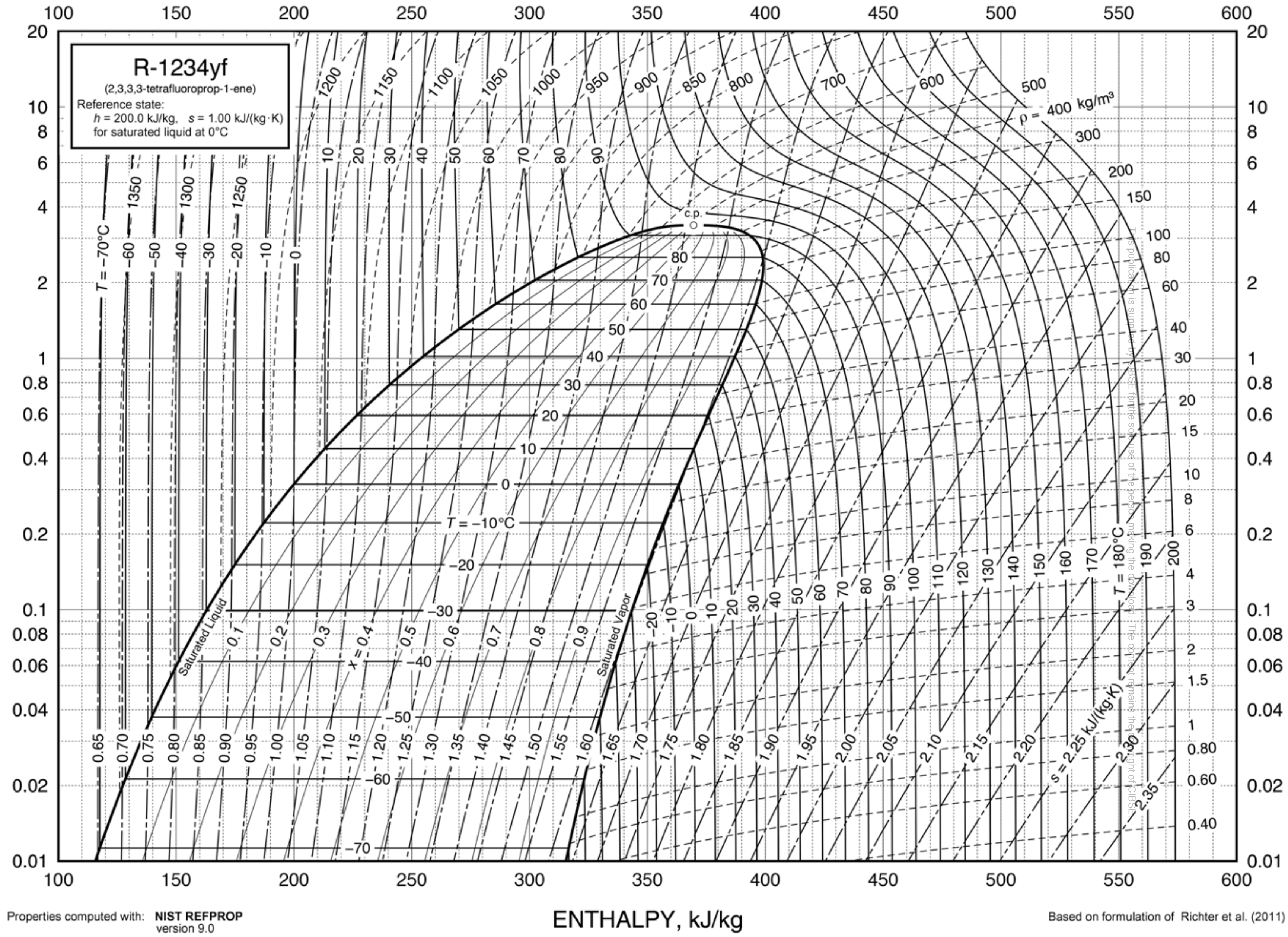


Figure 3.A2.10 Pressure–enthalpy chart for R1234yf (reproduced from ASHRAE Handbook: *Fundamentals* (2013) by kind permission of ASHRAE)

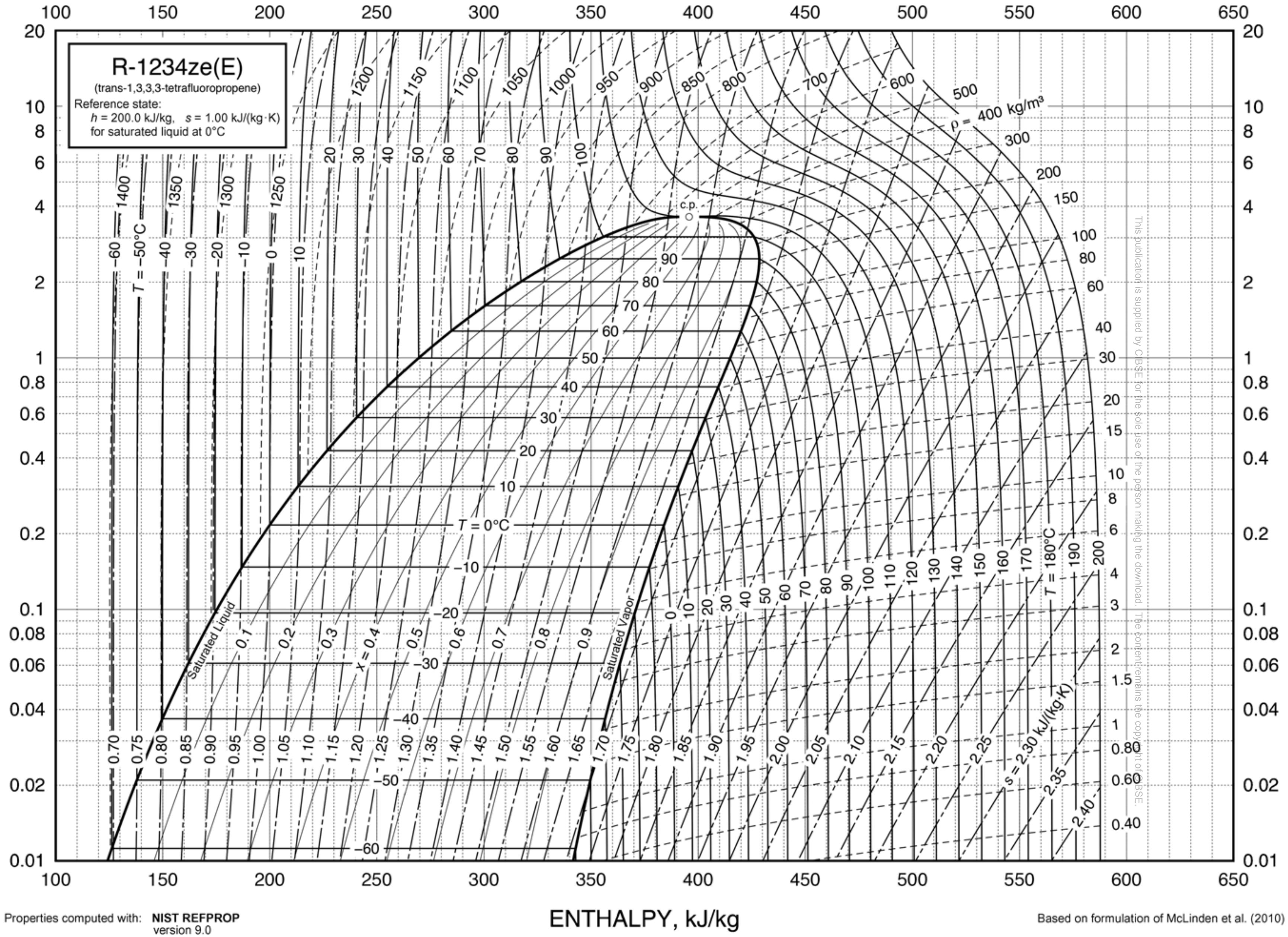


Figure 3.A2.11 Pressure–enthalpy chart for R1234ze (reproduced from ASHRAE Handbook: *Fundamentals* (2013) by kind permission of ASHRAE)

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Note: page numbers in *italics* refer to figures; page numbers in **bold** refer to tables