Ventilation and air Conditioning

CIBSE Guide B2

CIBSE

Department of Trade and Industry

Registered charity number 278104

ISBN 1 903287 162

Typeset by CIBSE Publications Department

Printed in Great Britain by Page Bros. (Norwich) Ltd., Norwich, Norfolk, NR6 6SA

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.

Foreword

The ventilation of buildings has been a major design issue of recent years. The decision whether to ventilate naturally or mechanically, and the implications for comfort, health, productivity and energy use have been much debated. This has influenced the design of buildings, dictating form and aesthetic, with some notable examples of innovative natural ventilation, such as the Contact Theatre in Manchester and the energy-efficient mechanical ventilation in the Elizabeth Fry Building at the University of East Anglia in Norwich. The traditional views that naturally ventilated buildings must be narrow in plan and that mechanically ventilated buildings are not energy-efficient have been countered by innovative design, both architectural and engineering. It is now generally accepted that there is not a preferred solution to ventilation design, whether mechanical or natural, and that appropriate ventilation design is needed, with the emphasis on design!

Ventilation provides fresh air for breathing, to dilute and exhaust pollutants and odours and often to exhaust heat gains. Typically people spend around 90% of their time in buildings ***so*** poor ventilation has an impact on their comfort, health, well being and productivity. Appropriate ventilation solutions should be energy-efficient and effective in relation to health and comfort. Excessive ventilation incurs energy penalties and insufficient ventilation leads to poor internal air quality. A balance is needed, which may vary over time with occupancy, thermal conditions and pollution levels.

Ventilation design affects the form and space planning of a building. It is important to make an early design decision about the choice of the ventilation system, and it should not happen by default. Failures are common and this affects the lifetime performance and cost of the building.

Traditional ventilation was natural, with openings located to facilitate the supply and extract under the natural forces of temperature and wind. However, many buildings relied on uncontrolled air leakage through cracks and gaps in the external envelope for ventilation, with windows only being opened during incidents of peak heat gains or pollution. As buildings became more complex, mechanical ventilation became essential. It was often combined with heating, cooling, humidity control and filtration, providing air conditioned buildings with little environmental contact with the outside. Significant fan power was needed to move the air around the buildings, and air conditioning became associated with profligate use **of** energy. This was coupled with growing concern over poor indoor air quality, often due to ineffective room air distribution and re-circulation of exhaust air. Maintaining and operating such complex systems on limited budgets proved difficult. Mechanical ventilation gained a bad reputation compared to natural ventilation, and was frequently blamed for sick building syndrome and other health and comfort complaints.

The response was to rediscover natural ventilation, with the system often strongly represented in building form and features. Architects identified the ‘greenness’ of their buildings through naturally ventilated atria and ventilation stacks. But the effective design of such buildings is often extremely complex, sometimes costly and often inappropriate for urban sites with high levels of noise and pollution. They may not always live up to their ‘low energy’ image.

Through innovations in design and application, mechanical ventilation systems have now become more energy-efficient and air quality has improved. Fan power has been reduced by developing more efficient fans, efficient air distribution and better controls. The amount of air distributed by the system is often reduced to just that amount required to supply ventilation for the occupants by reducing the cooling load on the space through improved facade design and control of solar gains and by reducing internal heat gains from lights and power.

Typically, an office might require 1–2 air changes per hour **(ACH)** for occupants’ ventilation alone but it might require 6 **ACH** and more to exhaust heat gains. Larger fans, ducts and associated space for equipment and distribution are needed to provide this ventilation. The modern approach is to de-couple the cooling from the ventilation. Thermal mass is increasingly used to help stabilise internal temperatures and avoid peak temperature situations. Chilled beams **or** ceilings are now used to provide additional cooling where heat loads are high, in order to keep the ventilation supply down to that required for the occupants. Underfloor ‘displacement’ systems give greater ventilation effectiveness with less air supply needed for good air quality. These developments have brought mechanical ventilation, with its traditional advantages of heat recovery and filtration, into the portfolio of energy-efficient design strategies.

There is now a growing interest in combining mechanical and natural ventilation through hybrid or 'mixed-mode' solutions. Seasonal hybrid systems might use natural ventilation in summer, when windows can be freely opened, and mechanical in winter, when it is cold outside and heat recovery from exhaust to supply air provides an energy efficiency and comfort advantage. Spatial hybrid solutions might have mechanical ventilation for the internal zones with natural ventilation at the perimeter.

There is now a spectrum of solutions from pure natural ventilation to total air conditioning, with the most appropriate solution often somewhere in between; that is, the hybrid system. Whatever the solution, the design of the ventilation system needs careful consideration. Today ventilation design is detailed and may use wind tunnels and computer models.

Revisions to Part L of the Building Regulations for England and Wales and the new Approved Document L set challenging new targets for building design to reduce energy consumption and carbon emissions. This will lead to the introduction of airtightness testing on new buildings and limits on total system fan power. Proper commissioning will be required on completion of the building. The Regulations in Scotland are also under revision to address these issues. Designers and clients are growing more aware of the environmental performance of their buildings and the problems and costs of getting it wrong.

This new CIBSE Guide: *Ventilation and air conditioning* addresses these new strategies for building ventilation and the recent developments in natural and mechanical ventilation systems. It updates sections B2 and B3 of the 1986 edition of CIBSE Guide B. It has been prepared by a task group drawn from a wide range of backgrounds and interests, and it seeks to provide up to date guidance on the issues affecting the selection, design and specification of ventilation and air conditioning systems. It also seeks to address the relationship between the form and fabric of the building and the chosen ventilation strategy.

The Guide will be a valuable tool for designers, both as a guide to current design thinking and as a means of access to a wealth of further guidance and information which is contained in its pages.

Professor Phil Jones
Chairman, CIBSE Guide B2 Steering Committee

Acknowledgement

This work was funded in part by the Department of Trade and Industry under the Partners in Innovation Scheme (formerly known as the DETR Partners in Innovation Scheme) and the CIBSE Research Fund. It was also supported by the various organisations represented on the Guide B2 Steering Committee. This document is published with the consent of the Department of Trade and Industry, but the views expressed are not necessarily accepted or endorsed by the Department.

Guide B2 Steering Committee

Professor Phil Jones (Chairman) (Welsh School of Architecture, Cardiff University)
Wayne Aston (Willan Building Services Ltd.)
Nick Barnard (Oscar Faber)
John Boxall (FBE Management Ltd.)
May Cassar (Bartlett School of Architecture)
Dr Andrew Cripps (Buro Happold Consulting Engineers)
Richard Daniels (Department for Education and Skills, Architects and Building Branch)
Mike Duggan (Federation of Environmental Trade Associations)
Dr Paul Evans (FBE Management Ltd.)
Les Fothergill (Department of the Environment, Food and Rural Affairs)
Dr George Henderson (W S Atkins plc, on behalf of the Department of Trade and Industry)
Dr Roger Hitchin (Building Research Energy Conservation Unit)
Denice Jaunzens (BRE Ltd.)
Ted King (Department of the Environment, Food and Rural Affairs)
Dr Geoff Leventhall (consultant)
Luke Neville (Brian Warwicker Partnership)
Derrick Newson (consultant, representing the Heating and Ventilating Contractors' Association)

Fergus Nicol (Oxford Brookes University)
Nigel Pavey (F C Foreman Ltd.)
Mike Price (Biddle Air Systems Ltd.)
Mike Smith (Building Services Research and Information Association)
Dr Helen Sutcliffe (FBE Management Ltd.)
Simon Steed (AMEC Design and Management Ltd.)
Chris Twinn (Ove Arup & Partners)
Christine Wiech (Max Fordham & Partners)
John Wright (Willan Building Services Ltd.)

Principal authors

Nick Barnard (Oscar Faber)
Denice Jaunzens (BRE Ltd.)

Contributors

Mike Burton (Oscar Faber)
May Cassar (Bartlett School of Architecture)
Richard Daniels (Department for Education and Skills, Architects and Building Branch)
Hywel Davies (Hywel Davies Consultancy)
Alan Fox (Oscar Faber)
Matthew Hignell (Oscar Faber)
Graham Millard (Oscar Faber)
Richard Pearce (Oscar Faber)
Iain Shaw (Oscar Faber)
Simon Steed (AMEC Design and Management Ltd.)
Chris Twinn (Ove Arup & Partners)

Editor

Ken Butcher

CIBSE Research Manager

Hywel Davies

CIBSE Publishing Manager

Jacqueline Balian

Contents

1 Introduction

1.1 General

Ventilation and air conditioning of buildings are subjects of increasing interest because of their contribution to effective building performance and occupant satisfaction and the increasing focus on energy consumption and carbon emissions from buildings. A particular cause of interest is the revision of Part L(1) of the Building Regulations for England and Wales and the equivalent Part J of the Building Standards (Scotland) Regulations(2), which will set challenging new targets for energy efficiency of buildings. There is also a growing awareness of the connection between ventilation, building envelope and structural design issues(3), and there is growing interest in the whole life costs(4) and performance(5) of buildings. Since building services are required to operate throughout the life of the building, their operating costs are a very significant element of the whole life costs of the system. For all these reasons there is a need for up-to-date guidance on the design of these systems.

1.2 Overview of Guide B2

Guide B2: *Ventilation and air conditioning* has its origins in sections 2 and 3 of the 1986 edition of CIBSE Guide B(6). It has been comprehensively revised to take account of developments in the intervening years, in particular to incorporate guidance on the technologies for low energy cooling which have emerged in the past decade. It is intended to be used by practising designers who hold a basic knowledge of the fundamentals of building physics and building services engineering.

The presentation of the material has also been revised. Section 2 provide describes an integrated approach to design that addresses issues of location, orientation and structural form and discusses their impact on the ventilation strategy for the building. The rationale for this is that some of the earliest decisions about form, fabric and orientation have major implications for the ventilation strategy. With the increasing use of single point responsibility in procurement, there is growing scope for these issues to be addressed together, rather than form and fabric leading ventilation strategy. Some of the changes to Part L of the Building Regulations (or Part J in Scotland) may also stimulate greater co-ordination at the early design stage. To be effective, it is important that this collaboration continues throughout the design process.

Section 3 describes the basic requirements for ventilation of offices, which are the most common type of non-domestic buildings. This is followed by sub-sections reviewing the specialist requirements of a wide range of other building types, from television studios to transport facilities.

Section 4 gives detailed guidance on natural, mechanical and mixed mode ventilation systems and air conditioning, building on some of the basic issues addressed in section 2.

Section 5 provides information about a wide range of equipment used in ventilation and air conditioning systems, in particular covering some of the newer items of technology such as chilled ceilings and beams.

The overall process of design development, from the initial outline design through system selection and detailed equipment specification, is summarised schematically in Figure 1.1.

1.3 Energy efficiency

The UK is committed to significantly reducing carbon emissions by the year 2010, with a target of a 20% cut based on 1990 levels. As well as maintaining the role of the Energy Efficiency Best Programme to promote energy efficiency, the government has also introduced the Climate Change Levy, effectively a specific tax on energy use, and enhanced capital allowances for certain energy efficient measures.

It is intended that this will stimulate a greater interest in energy efficiency measures amongst building owners and operators, and that energy efficiency will be given a greater prominence in decisions about building design.

Allied to this is the introduction of the revised Part L of the Building Regulations in England and Wales(1). This sets significantly more challenging 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 major refurbishment.

Recent studies under the government's Energy Efficiency Best Practice programme suggest that there is likely to be a significant increase in energy consumption related to air conditioning. To meet the targets for reduced carbon emissions it is particularly important to ensure that such systems are as energy efficient as possible.

1.4 Whole life cost

It is now a requirement of public sector purchasers that they move to whole life cost based procurement(8). The Private Finance Initiative (PFI) has already stimulated a marked increase in interest in whole life costing and there has been a growth in the availability of data to support the activity(9).

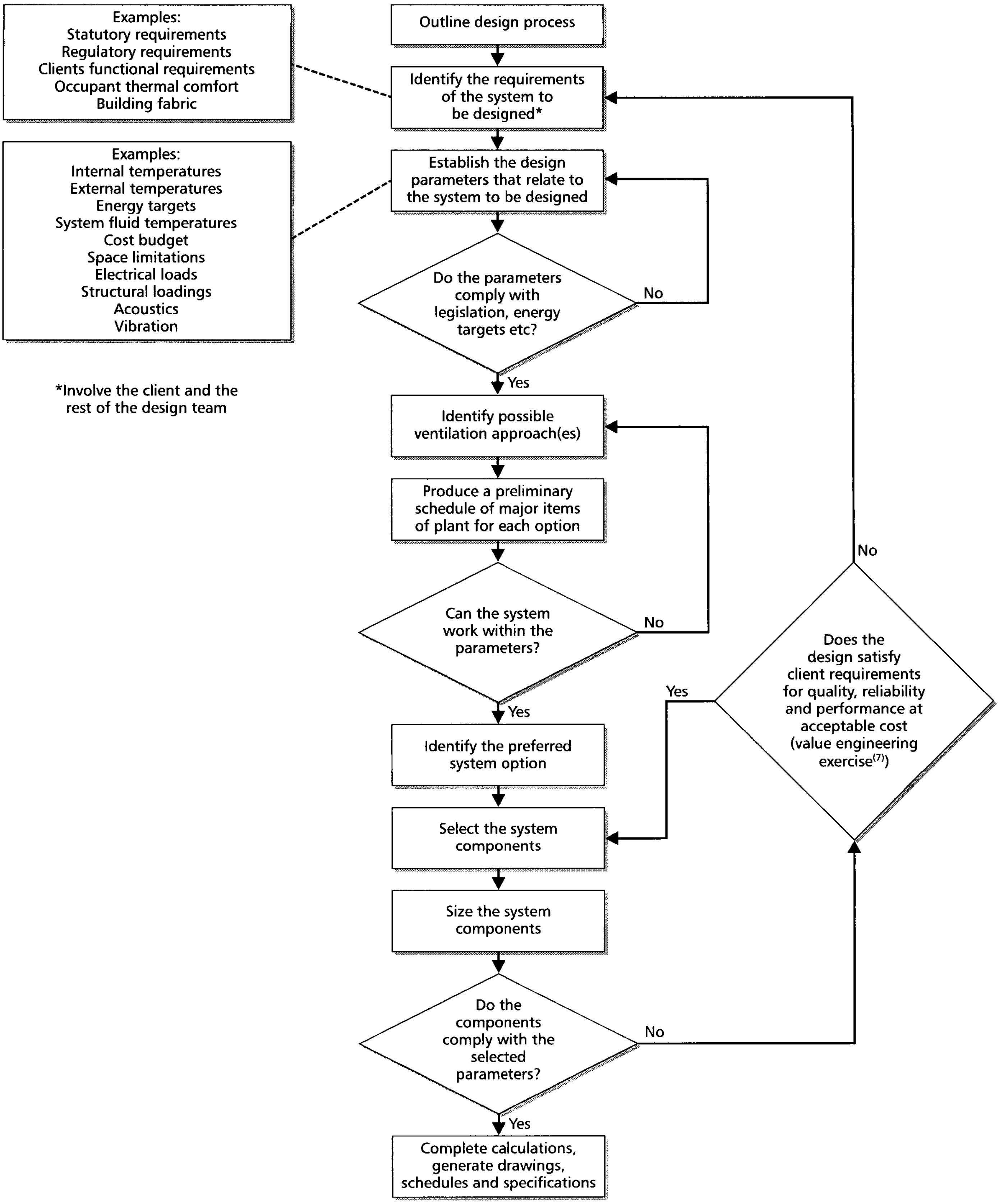


Figure 1.1 Outline design process

Once again, appropriate design of ventilation and air conditioning systems can significantly reduce the whole life costs of the system. Potentially costly modifications and alterations can be avoided by ensuring that the system requirements are properly defined and the design fully addresses the requirements, and is compatible with the chosen form, fabric and orientation.

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.

Increasingly, professionals designing ventilation systems will need to take account of whole life requirements, and this Guide will be an aid to them as they do so.

1.5 Building performance

There has been growing evidence for a number of years that the effectiveness of building ventilation has a significant effect on the performance of those working in the building. Poor indoor air quality impairs the performance of employees in a workspace. Evans et al.[10] have estimated that design, build and operating costs are in the ratio 1:5:200. It can therefore be seen that poor standards of building ventilation can have a significant negative effect on operating costs through their adverse effect on employee performance. It is worthwhile for building owners and operators to pay careful attention to ensure that buildings are appropriately ventilated and provide a healthy and effective environment that will not impair the productivity of their employees.

In terms of business costs, the cost of running and staffing the business is the most significant. The costs of designing, installing and operating effective ventilation systems are small compared to the costs to the business of impaired performance arising from poor ventilation. Over a system life of ten to fifteen years a 1% reduction in productivity may easily equal any 'savings' made on the design and installation costs of the system and its operation by inappropriate cost saving measures. The use of this Guide and the adoption of an integrated approach to the design of the fabric of the building and its services will help to avoid such false economies and deliver buildings which more fully meet the needs of their owners, operators and occupiers, and which provide more productive places for the people who work in them.

References

1 The Building Regulations 2000 Approved Document L: Conservation of fuel and power (2001 edition) (London: The Stationery Office) (in preparation)

2 Technical standards for compliance with the Building Standards (Scotland) Regulations 1990 (London: The Stationery Office) (1999)

3 Thomas R (ed.) *Environmental Design: An Introduction for Architects and Engineers* (2nd edition) (London: E & F N Spon) (1999)

4 *Whole Life Costing: A client's guide* Construction Clients Forum 2000 (London: Confederation of Construction Clients) (2000)

5 Allard R Get a whole-life *Building* **266** (18) 69 (May 2001)

6 *Ventilation and air conditioning (requirements)* and *Ventilation and air conditioning (systems and equipment)* in CIBSE Guide B: *Installation and equipment data* (London: Chartered Institution of Building Services Engineers) (1986) (out of print)

7 Hayden G W and Parsloe C J *Value engineering of building services* BSRIA Applications Guide AG 15/96 (Bracknell: Building Services Research and Information Association) (1996)

8 *Whole life costs* Construction Procurement Guidance No. 7 (London: Office of Government Commerce) (2000) (www.ogc.gov.uk)

9 *Building Services Component Manual Life Manual* (Oxford: Blackwell Science) (2001)

10 Evans R, Haste N, Jones A and Haryott R *The long term costs of owning and using buildings* (London: Royal Academy of Engineering) (1998)

2 Integrated approach

2.1 Introduction

Ventilation is essential to the provision of a safe, healthy, productive and comfortable living or working environmente in selecting an appropriate ventilation strategy, and ultimately system, consideration must be given primarily to meeting the requirements of the people and processes that occupy the building without being excessive and therefore wasteful. However the pursuance of an integrated design approach to achieve this also links the ventilation strategy with the design of the building fabric, in that as a pre-requisite all reasonable steps should be taken to maximise the potential of the building fabric. This is also commonly referred to as the 'passive approach'. In particular, an appropriate degree of airtightness should be aimed at[1]. The design process must be based on a clear understanding of client and end user needs and expectations and must be followed by effective commissioning, handover and building management. Close collaboration between the architect, services and structural engineers and the client is essential from the earliest stages of the outline design process.

Section 2 considers:

- the identification of key building performance requirements for the application of ventilation in support of these requirements
- key factors to be considered in terms of an integrated approach to building design
- issues relevant to the selection of specific ventilation strategies, i.e. natural or mechanical ventilation, comfort cooling or air conditioning, or mixed mode; see section 4 for more detailed information on these issues.

Information on the determination of suitable ventilation rates is given in section 3. See also CIBSE Guide A: ***Environmental design***[2] for further information on comfort criteria.

2.2 Establishing key performance requirements

The key performance requirements that need to be clarified before a ventilation strategy can be selected are summarised in Table 2.1. Ideally, where the issues highlighted in the table have not been covered within the specification documents, 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 in these respects.

The design team should also be able to advise the client of the cost implications (on a whole life basis[5,7] if requested) of meeting their stated requirements. Requirements may subsequently be adjusted over the course of the project to meet financia1 constraints or changing business needs. The design team must also be able to advise on the impact of any such changes on the ultimate building performance.

An appreciation of the issues shown in Table 2.1 is an essential part of the briefing process and suitable references on specific issues are provided. Further guidance on briefing as it applies to building services is given in ***Project Management Handbook for Building Services***[3].

2.2.1 Energy and environmental targets

The chosen ventilation 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:

- CIBSE TM22: ***Energy Assessment and Reporting Methodology***[13], which provides energy benchmarks and target assessment methods for dealing with banks and similar agencies, hotels, offices and mixed use buildings. Table 2.2, reproduced from TM22 provides energy usage benchmarks for 'good practice' and 'typical' performance, based on four generic office classifications. TM22 also contains a breakdown by end usage for fans, lighting and desk equipment for each office type.
- The series of Energy Consumption Guides[14], published under the government's Energy Efficiency Best Practice Programme, which provide energy benchmarks and targets for industrial buildings and sites, offices, public houses, hotels, hospitals, domestic properties, nursing and residential homes, and other non-domestic sectors.
- Building Maintenance Information's report ***Energy benchmarking in the retail sector***[15], which provides energy benchmarks within the retail sector.

Table 2.1 Establishing performance requirements

Issue	Requirement/ comments
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, see section 2.3.1
Energy/environmental targets	Use of existing specifications or appropriate advice from the design team required, see section 2.2.1 Compatibility with indoor environment standards
Indoor environmental standards	Use of existing standards or appropriate advice from the design team required, see section 3 and CIBSE Guide A[2]
	Areas or objects with special requirements
Provision of controls	Individual, local, team, zone or centralised basis
	Required closeness of control (e.g. of temperature, humidity, air quality, air flow)
	The required interaction of the end user with the building services, see section 2.2.2
	The required basis of control, e.g. temperature, CO_2, CO or other
Demands of the building occupants and activities[4]	The business process(es) to be undertaken in the building may demand specified levels of availability of ventilation
	Work patterns over space and over time (regularity, shifts, team structure)
	Cellular and open plan mix with associated partitioning strategy and likelihood of change
taken into	Occupancy numbers and anticipated maximum occupancy over the building lifetime that might need to be account
	Average occupancy density and any areas of high or low density
	Functions of space use, processes contained therein and subsequent internal loads (e.g. standard office space, meeting rooms, lecture theatres, photocopying rooms, sports hall, laboratories, manufacturing environments, retail space)
	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[5-8]
	Requirements for calculations to be carried out on systems or system elements and the basis for these calculations[4,9,10]
	Has the client been involved in discussions of acceptable design risk?
	The importance of part load performance
Reliability	The business process(es) to be undertaken in the building may demand specified levels of reliability of the ventilation systems
Maintenance requirements[11]	Understanding of the client's ability to carry out, or resource, maintenance
	Client willingness for maintenance to take place in the occupied space
	Any requirement for 'standard' or 'familiar' components
Associated systems	Implications of any particular requirements, e.g. fire, security, lighting, acoustic consideration.
Security	Restrictions on size and location of any openings
Future needs	Adaptability, i.e. the identified need to cope with future change of use
	Flexibility, i.e. the identified need to cope with future changes in work practices within the current building use[2]
	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), design which makes sensible agreed allowances for future changes and over-design[12]
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 projects

Table 2.2 Office system and building energy benchmarks[13]

Fuel/application	Delivered energy for stated office classification* / kW·h·m⁻²							
	Type 1		Type 2		Type 3		Type 4	
	Good practice	Typical	Good practice	Typical	Good practice	Typical	Good practice	Typical
Fossil fuels:								
— gas/oil heating and hot water	79	151	79	151	97	178	107	201
— catering (gas)	0	0	0	0	0	0	7	9
Electricity:								
— cooling	0	0	1	2	14	31	21	41
— fans, pumps and controls	2	6	4	8	30	60	36	67
— humidification	0	0	0	0	8	18	12	23
— lighting	14	23	22	38	27	54	29	60
— office equipment	12	18	20	27	23	31	23	32
— computer room	0	0	0	0	14	18	87	105
Total gas or oil	79	151	79	151	97	178	114	210
Total electricity	33	54	54	85	128	226	234	358

* Type 1: cellular, naturally ventilated; Type 2: open plan, naturally ventilated; Type 3: standard air conditioned; Type 4: prestige air conditioned

- The *Building Research Establishment Environmental Assessment Method* (BREEAM)[16], which provides an environmental assessment methodology for industrial units, offices, superstores and supermarkets and housing.

- BSRIA's *Environmental code of practice for buildings and their services*[17] and associated publications, which provide 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.

2.2.2 Provision of controls: the end user perspective

The provision of a suitable mechanism for the end user to control conditions within their workplace is fundamental to users' satisfaction with the internal environment. Any requirements of the client must be considered in the light of the designer's own experience of end user behaviour, in particular:

- ensuring fairness and consistency of control by avoiding occupants being unduly affected by controls from which they do not benefit

- providing rapid acting controls that give feedback to occupants to demonstrate response

- making sensible decisions with regards to the choice of manual versus automatic control (manual overrides should be provided wherever practical; any automatic change in state should happen gradually to avoid feelings of discomfort, and should only affect those who benefit from it)

- removing unnecessary complexity by providing controls that are simple, intuitive, well labelled and visible.

Further guidance on these issues can be found in work by BRE[18] and Bordass et al[19].

2.3 Interaction with building fabric, services and facilities

2.3.1 Building fabric

The final ventilation rate is based on fresh air requirements and any additional ventilation required for comfort and cooling purposes based on estimates of:

- internal gains determined by the occupants, e.g. occupancy itself, lighting and small power loads, see section 3

- internal gains determined by the fabric, e.g. insulation, glazing, thermal mass, as discussed below.

Although the architect is traditionally associated with making many of the fabric-related decisions, the building services engineer must be able to advise on their implications for the building services and, ultimately, ventilation, energy use etc. The building services engineer must therefore be involved in the decision making process as far as is practical, and at as early a stage in the process as possible. The building services engineer should also be consulted prior to any changes in decisions made previously, as to their impact on the ventilation system performance.

In order to engage effectively with the architect, the building services engineer must be able to enter into a dialogue on the issues introduced in Table 2.3, as a minimum. (Note that this table focuses solely on issues relating to the interaction between the building fabric and services. To these must be added, for example, consideration of the building function and broader issues, as raised in Table 2.1). Where the ventilation strategy for the building depends on its thermal mass, early consultation with the structural engineer is also needed to consider, for example, the implications for roof design. At some point it may also be necessary to involve a façade specialist, who could advise the client accordingly.

Table 2.3 Issues influencing the choice of ventilation strategy

Issue	Comments
Location (see Figure 2.1)[21]	Adjacent buildings can adversely affect wind patterns. The proximity of external sources of pollution can influence the feasibility of natural ventilation. The proximity of external sources of noise can impact on the feasibility of natural ventilation.
Pollution	Local levels of air pollution may limit the opportunity for natural ventilation. It may not be possible to provide air inlets at positions suitable for natural ventilation given the inability to filter the incoming air successfully(22).
Orientation[20]	Buildings with their main facades facing north and south are much easier to protect from excessive solar gain in summer. West façade solar gain is the most difficult to control as high gains occur late in the day. Low sun angles occurring at certain times of year affect both east and west facing facades.
Form[20]	At building depths greater than 15 m the ventilation strategy becomes more complex; the limit for daylighting and single-sided natural ventilation is often taken as 6 m. An atrium can enhance the potential for natural ventilation, see section 2.5.1.
	Tall buildings can affect the choice of ventilation system due to wind speeds and exposure. Adequate floor to ceiling heights, are required for displacement ventilation and buoyancy driven natural ventilation; a minimum floor to ceiling height of 2.7 m is recommended, see section 4.3.
Insu1ation	Insulation located on the external surface de-couples the mass of the structure from the external surface and enables it to stabilise the internal environment. In well-insulated buildings provision must be made for the removal of excess heat, for example through night cooling, see section 4.7.
Infiltration[1,29]	Ventilation strategies, whether natural or mechanically driven, depend on the building fabric being appropriately airtight. This implies a good practice standard of 5 m3.h-1 per m2 of façade (excluding consideration of the ground floor) and requires suitable detailing. Site quality checks should be followed by air leakage pressure testing as part of the commissioning requirement.
Shading[20,21]	The appropriate use of external planting or other features can reduce solar gain. In terms of effective reduction of solar gain, shading devices can be ranked in order of effectiveness as follows: external (most effective), mid pane, internal (least effective), see Figure 2.2.
	Horizontal shading elements are most appropriate for reducing high angle solar gains, for example in summer time on south facing facades. Vertical shading devices are most appropriate for reducing low angle solar gain, e.g. on east and west facades. Control of solar shading devices should be linked with that of the ventilation system. Glare must be controlled to avoid a default to 'blinds-down' and 'lights-on' operation.
Window choice[22]	Openable areas must be controllable in both summer and winter, e.g. large openings for still summer days and trickle ventilation for the winter time. Window shape can affect ventilation performance; deep windows can provide better ventilation than shallow. High level openings provide cross ventilation, low level openings provide local ventilation, although draughts should be avoided at working level. The location of the opening areas affects the ability of the window to contribute to night cooling (see section 4.7). Window operation must not be affected by the choice of shading device. See section 5.3 for details of window characteristics.
Glazing[23,24]	Total solar heat transmission through window glazing can vary over a sixfold range, depending on the combination of glass and shading mechanisms selected. Figure 2.2 shows the relative effectiveness of eight glazing and shading systems. Figure 2.2 underlines the importance of decisions about glazing and shading to the overall ventilation strategy.
	At concept stage the percentage of glazed area (normally 20-40% of façade area) and selection of glazing type must balance thermal, ventilation and lighting needs. The choice includes single, double, triple glazing with selective coatings or gaseous fill. The type of coating may have a greater influence than the glazing type. Ideal glazing is transparent to long-wave radiation and reflective to short-wave radiation. Selective low-emissivity double-glazing is equivalent to air-filled triple-glazing.
	The use of tinted glazing may increase the use of supplementary electric lighting, increasing internal heat gains and energy use. Window frame construction and detailing must also be considered.
Thermal mass	Thermal mass is used to reduce peak cooling demands and stabilise internal radiant and air temperatures. The first 50 to 100 mm of the structure is most effective on a 24-hour basis. Thermal mass can be introduced into the ceiling/floor slab (most effective), walls or partitions, but must be 'accessible' in all cases. Heat transfer can be via the surface of the material or via cores/channels within it. The exposure of thermal mass has architectural and other servicing implications, although these effects can be reduced, e.g. by the use of perforated ceilings. See section 4.7 for further details of incorporating thermal mass.

It is important to note that maximising the 'Passive contribution' to be gained from the building fabric itself requires an understanding of both the advantages and disadvantages of this approach. For example, external shading reduces the need for cooling but increased insulation and airtightness may lead to the need for increased ventilation and cooling.

For a detailed explanation of the role of the building fabric in contributing to an energy efficient solution see CIBSE Guide F[20] and other publications referenced in Table 2.3. It is also important to consider the risks of air leakage through the building fabric and its subsequent impact on infiltration rates and heat loss calculations[1]. The most common air-leakage risks are:

- at the junctions between the main structural elements, e.g. wall to roof, wall to floor, and wall to foundation
- at the joints between walling components, e.g. sealant or gasket joints between heavyweight or curtain walling panels, overlapping joints between lightweight sheet metal wall panels and boundaries of different cladding/walling systems
- around windows, doors and roof lights, e.g. between window or doorframes and walls or floors,

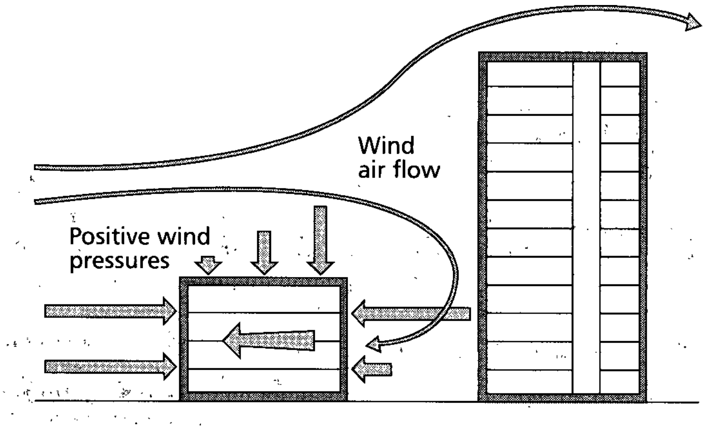


Figure 2.1 Impact of localised wind effects

between doors and windows and their frames, and between frames and sills

- through gaps in membranes, linings and finishes, e.g. in wall membranes and dry linings, in ceiling linings and boundaries with wall linings and gaps in floor finishes and around skirtings
- at service penetrations, e.g. electrical sockets and conduits, gas and electricity entry points, ventilation pipes for sanitary waste, overflow pipes and flues
- around access and emergency openings, e.g. to roof space, to roof, to floors and to services and delivery points
- through some building materials, e.g. brickwork may be permeable especially where construction quality is low.

2.3.2 Interaction with the lighting system

The design strategy for daylight provision forms part of the selection process for window and glazing types and shading devices. Guidance on the overall process is available elsewhere(23). Integration of the electric lighting system to minimise its impact on the design and operation of the ventilation system requires that internal gains from the lighting be minimised through(25,26):

- the selection of appropriate light levels, differentiating between permanently occupied workspaces and circulation areas; task lighting should be considered to provide increased light levels locally.
- the selection of efficient light fittings (recognising that decorative fittings may have a lower efficiency)
- the installation of an appropriate lighting control system, based on a combination of time of day, occupant presence and light levels
- the use of ventilated light fittings to reduce internal gains.

Consideration should also be given to the impact of the chosen ventilation strategy on the lighting system, e.g. the use of uplighting with exposed thermal mass(27). The integration of exposed thermal mass is discussed in section 4.7.

2.3.3 Small power loads

Small power loads arising from IT and other office-type equipment are an increasingly significant component of internal gains. Accounting for them in the design of the ventilation system requires a realistic calculation of their impact in terms of peak load and anticipated diversity. In order to reduce internal gains the designer should:

- encourage the client to select low energy equipment and introduce power cut-off mechanisms
- locate shared equipment, e.g. vending machines, photocopiers, in a space that can be readily cooled.

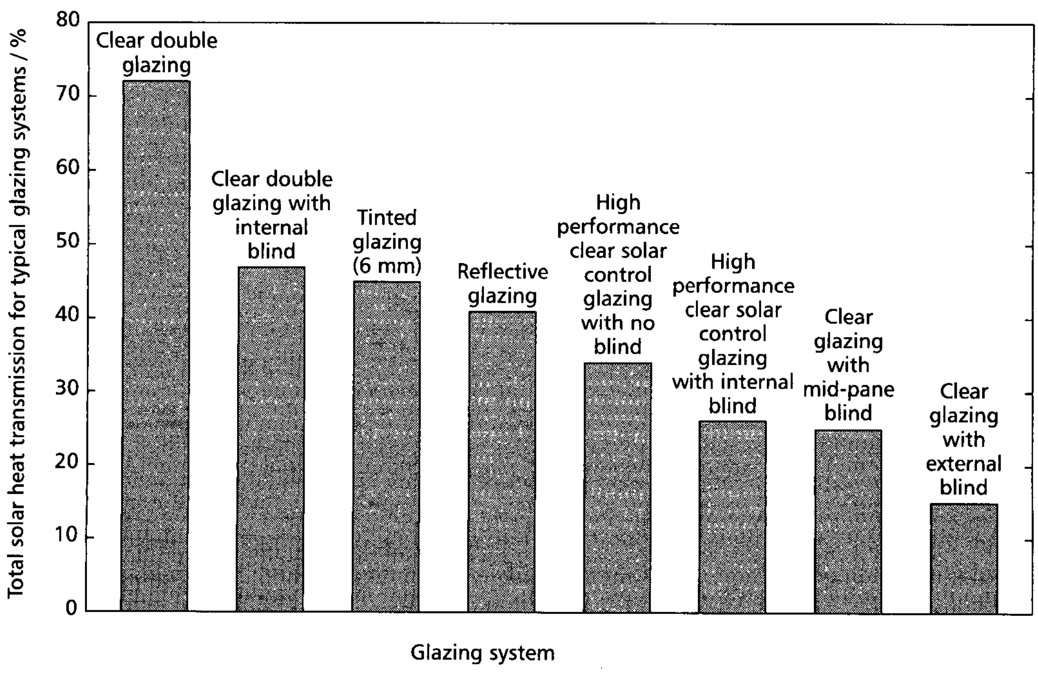


Figure 2.2 Total relative solar heat transmission for typical glazing systems (courtesy of Ove Arup & Partners)

Table 2.4 Purposes of ventilation

Purpose	Explanation
To provide sufficient 'background' ventilation for occupants in terms of air quality for breathing and odour control	Typical rates need to be increased where smoking is permitted or additional sources of pollution are present. Most pollutants originate from sources other than people but in such cases general ventilation has been shown to be much less effective than treating the problems at source: e.g. by specification, cleanliness and local extraction.
To provide natural cooling during the occupied period	Care must be taken to avoid excessive air change rates that may cause draughts or disturb documents. Higher rates may be practicable in spaces occupied transitionally, such as atria. The balance point above which mechanical cooling will provide a more effective solution should be considered.
To provide natural cooling outside the normal occupied period	Night cooling or 'night purging' can remove heat built-up in a structure and its contents, and provide-some pre-cooling for the following day. Practical limitations will exist in terms of acceptable secure openable areas in the case of natural ventilation and on duct size and fan energy consumption for ducted mechanical systems.
To exhaust heat and/or pollutants from localised sources or areas	Examples are kitchens, toilets, vending areas and equipment rooms. This enables adjacent areas to be more comfortable with less conditioning of the air. Such systems often need to operate for longer hours than those serving the main spaces, therefore independent extract systems are preferred.
To act as a carrier mechanism for mechanical cooling and/or humidity control	This can be either via an all-air system, in which the air is treated centrally, or via air/water or unitary systems in which the air is recirculated and treated locally.
To prevent condensation within the building fabric[28]	Adequate ventilation for condensation control exceeds the minimum rate of fresh air necessary for health and comfort. There is a specific need to address the ventilation of areas where moisture generating activities occur.
To enable the efficient operation of processes	Needs are entirely dependent on the process. Ventilation may be required to ensure safe combustion or to ensure that machinery is maintained within a suitable temperature range, e.g. lift motor rooms.

2.4 Purpose of ventilation systems

In designing any ventilation system it is necessary to understand the functions required of it, see Table 2.4. In summary these are:

- to provide adequate indoor air quality by removing and/or diluting pollutants from occupied spaces
- to provide adequate ventilation for the effective operation of processes
- to provide a heat exchange mechanism
- to prevent condensation within the building fabric.

Consideration of the requirements of each function within offices is given in section 3.2 and in other sectors in sections 3.3 to 3.24. In winter, any heat exchange above that needed to control air quality has a heating energy penalty. The relative importance of excess winter ventilation increases with increasing thermal insulation standards. In summer, however, ventilation rates above those required for reasons of air quality may reduce the demand for mechanical cooling, although this will only be possible if the outside air temperature is lower than the room temperature. Even if inside and outside temperatures are similar, increased air movement can create a sense of freshness and increased occupant satisfaction with the internal environment. The advantage of 'free' cooling by natural rather than mechanical ventilation is that the fan energy, as well as the heat gained by the supply air, is eliminated.

The design requirement for an energy efficient ventilation system is to create a satisfactory internal environment given that the cooling potential of natural ventilation is limited, see sections 2.5 and 4.3. There is also less flexibility for air distribution since natural ventilation usually relies on supply air from the perimeter of the building. In contrast, mechanical ventilation can be supplied to any part of a building through the distribution ducts.

2.5 Choice of ventilation strategy

This section gives an overview of the following strategies:

- natural ventilation
- mechanical ventilation
- comfort cooling
- air conditioning (which may be 'close control')
- mixed mode systems (i.e. a combination of natural and mechanical ventilation).

The selection of a strategy is affected by, amongst other factors, location, plan depth, heat gains, internal and external pollutant sources, economics, energy and environmental concerns and internal layout. Ultimately it is the use and occupancy of a space that determines the ventilation needs. There is no universal economic solution, although there are some best practice indicators that are considered in subsequent sections. Each ventilation system design must be evaluated on its merits, to suit the particular circumstances.

Excessive air infiltration can destroy the performance of a ventilation strategy[1], hence good ventilation system design should be combined with optimum air tightness to achieve energy efficient ventilation. Inclusion of a requirement for air tightness in a specification does not lead to the choice of a particular design strategy. For example, mechanical ventilation is not necessarily the inevitable consequence of requiring that a building be airtight. Applying the axiom 'build tight, ventilate right', ventilation via natural openings may be suitable. However, it cannot be assumed that a mechanically ventilated building is suitably airtight.

The client needs to understand and accept the ramifications of the selected strategy. Figure 2.3[21] illustrates a typical, broad-brush decision-making process, while Table 2.5 shows the limits of natural ventilation. However, Figure 2.3 and Table 2.5 are of particular reference to office environments and are not necessarily appropriate for other building types. See section 4 for details of individual strategies and further guidance on their selection.

2.5.1 Natural ventilation

Natural ventilation may be defined as ventilation that relies on moving air through a building under the natural forces of wind and buoyancy.

Wind driven ventilation can be single sided (through a single or double opening) or cross flow, which is more effective. Buoyancy driven ventilation can be assisted by stacks, wind towers, atria rooflights, conservatories, or by the façade itself.

Natural ventilation is generally applicable in many building types (industrial buildings being a possible exception) of up to 15 m depth, or greater if designed appropriately. The effective depth over which particular

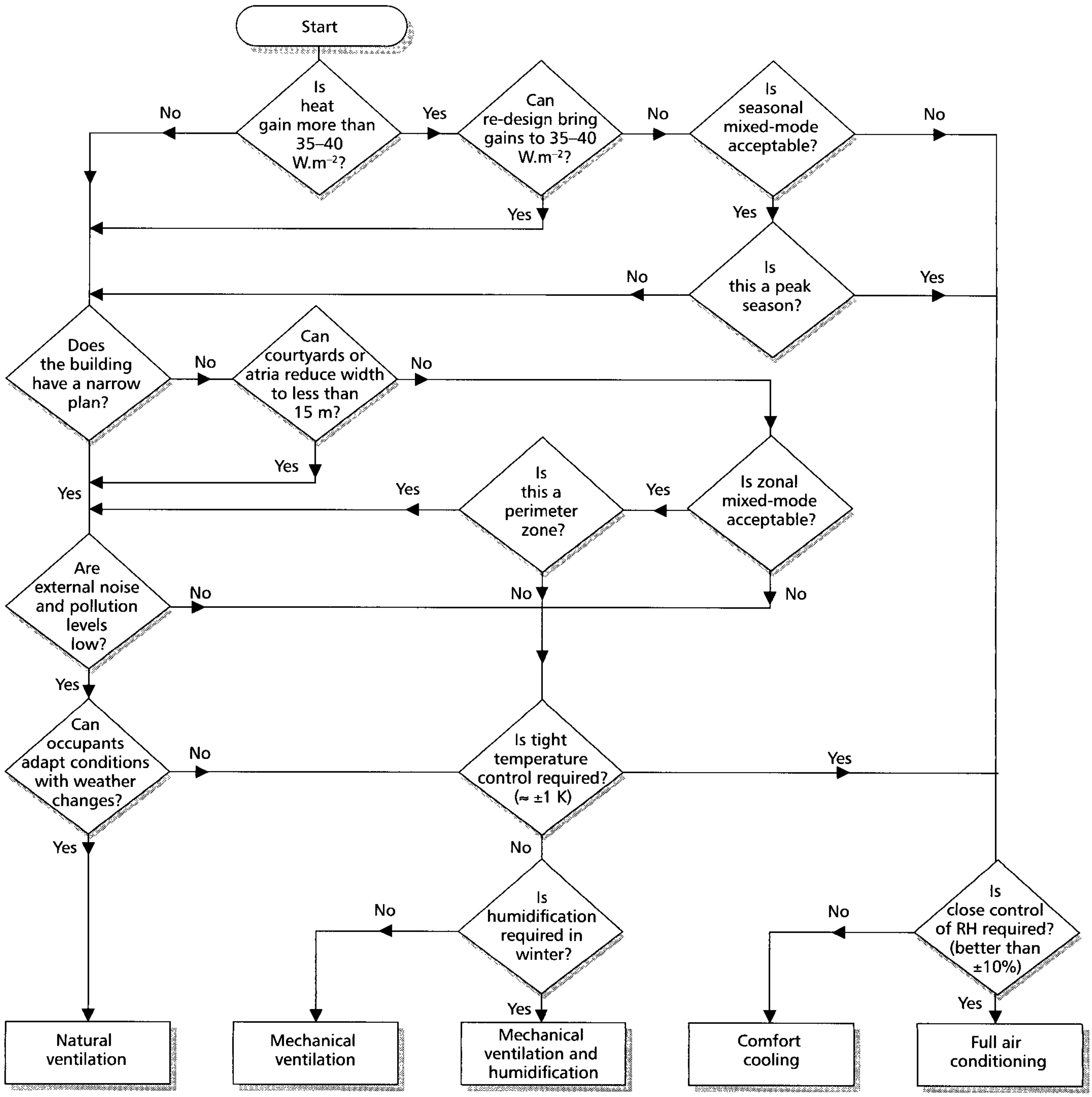


Figure 2.3 Selecting a ventilation strategy[21]

options are viable is the key limitation, see Table 2.5. However, this cannot be applied universally since few data exist for buildings with floor-to-ceiling heights greater than 3.5 m[21].

Table 2.5 Natural ventilation options and their effective depth

Strategy	Effective depth relative to office floor-to-ceiling height
Single sided, single opening	2 *x* floor-to-ceiling height
Single sided, double opening	2.5 *x* floor-to-ceiling height
Cross flow	5 *x* floor-to-ceiling height
Stack ventilation	5 *x* floor-to-ceiling height
Atria	10 *x* floor-to-ceiling if centrally located

If the internal heat gains rise above 40 W.m^{-2}, natural ventilation by itself may be inadequate and a strategy involving mechanical assistance will be required[21]. Table 2.6 illustrates how the design of the building fabric might be adapted to help meet this target[21]. However, this table is indicative of scale only and will vary depending on the characteristics of the particular building and on the freedom or otherwise allowed to the designer by such characteristics .

Further details of natural ventilation systems, including key components and design methods, are given in section 4.3.

Key characteristics of natural ventilation to be borne in mind include:

- *Risk of draught:* this can be difficult to control, especially in an open plan space. Attention must be paid to the size and location of openings and their control.
- *User control:* this can be dependent upon the proximity to a window. Users are reported to favour access to openable windows when external conditions are suitable.
- *Closeness of control:* close control over temperature and humidity is not possible.
- *Capital costs:* costs are heavily influenced by the complexity of the window or ventilator design and by the building form necessary to achieve effective natural ventilation.
- *Running costs:* maintenance costs will be incurred for a natural ventilation system if it is subject to automated window opening.
- *Flexibility:* this is difficult .to achieve if extensive partitioning is introduced. Natural ventilation may reach its limits if heat gains increase.
- *Predictability:* performance can be modelled in theory, but in practice is subject to variation in the motivating forces of wind and temperature difference between inside and outside the building.
- *Noise:* there are no problems with plant noise but there may be an issue with external noise depending upon location, or transmission of internal noise.
- *Ability to deal with polluted environments:* filtration is very difficult due to the pressure drops involved.
- *Winter ventilation:* needs careful design in areas of high occupancy.

Table 2.6 Relationship between design features and heat gains

Design features	Total heat gains* / W.m^{-2} floor area			
	10	20	30	40
	Minimum room height / m			
	2.5	2.7	2.9	3.1
Controllable window opening (to 10 mm)	Essential	Essential	Essential	Essential
Trickle vents for winter	Essential	Essential	Essential	Essential
Control of indoor air quality	May be required	May be required	Essential	Essential
Design for daylight to reduce gains	May be required	Essential	Essential	Essential
Daylight control of electric lighting	May be required	May be required	Essential	Essential
100% shading from direct sun	May be required	Essential	Essential	Essential
Cooling by daytime ventilation only	Essential	Essential	Problem	Problem
Cooling by day and night ventilation	Not necessary	May be required	Essential	Essential
Exposed thermal mass	Not necessary	Not necessary	Essential	Essential

* i.e. people + lights + office equipment + solar gain

2.5.2 Mechanical ventilation

Mechanical ventilation may be defined as the movement of air through a building using fan power; filtration and heating of the air may also take place.

The most often used strategy is 'balanced supply and extract'. Other options are mechanical supply and natural extract, natural supply and mechanical extract, natural extract and mixed supply (as used in some industrial buildings).

The ventilation delivery method may be either displacement (laminar or piston flow) ventilation, Or mixing (turbulent) ventilation systems. The former introduces 'fresh' air gently, normally at low level, at a temperature close to that of the room air. Warm polluted air is extracted at ceiling height. In the latter system type air entering the space is thoroughly mixed with the air within the space. The distribution mechanisms can be via a floor, ceiling or wall supply.

The main roles for mechanical ventilation without the use of mechanical cooling are:

- to provide adequate background fresh air ventilation or compensate for natural means when they are inadequate for occupant well being

- to provide adequate fresh air ventilation for fume control, when a fixed rate would normally be applied
- to cool the building when the outside air is at an appropriate temperature.

There is a considerable difference in the supply air rates for each role. Therefore if a single system is required to combine these roles it would need to be capable of variable volume flow. The typical supply air rates for background ventilation are between 1 and 2 air changes per hour (ACH) and the rate to achieve adequate cooling by ventilation is of the order 5-10 ACH, see section 4.2.

For further details of mechanical ventilation systems refer to section 4.4. Where mechanical systems are combined with openable windows, this is known as a 'mixed mode' approach, see section 4.5.

2.5.2.1 Key performance characteristics

Key characteristics of mechanical ventilation to be borne in mind include:

- *Risk of draught:* in theory, the draught risk is controllable provided that the system is appropriately designed, commissioned and integrated with the space layout.
- *User control:* control can be provided at an individual level, regardless of location. However there is a cost penalty.
- *Closeness of control:* close control over temperature and humidity is possible (subject to air being at a suitable temperature), but with penalties in energy use.
- *Capital costs:* costs are heavily influenced by the amount of mechanical plant required and whether the façade is sealed; alternatively a mixed mode approach requires openings in the fabric.
- *Running costs:* maintenance costs depend on the quantity of plant. Energy costs depend on the fan pressure drop of the mechanical system and the efficiency of heat recovery (if any).
- *Flexibility:* flexibility can be achieved but with cost penalties.
- *Predictability:* performance can be accurately predicted subject to appropriate commissioning and subsequent maintenance.
- *Noise:* external noise and fan noise can be reduced through attenuation, a space allowance will be required.
- *Ability to deal with polluted environments:* filtration is possible in harsh external and internal environments.

2.5.2.2 'Free cooling' versus mechanical cooling

Using mechanical ventilation for cooling (i.e. 'free cooling') requires careful consideration. The energy used to transport the air can be greater than the delivered cooling energy. At worst, the work involved in moving the air (both supply and recirculated) will raise its temperature, resulting in warming of the building. Therefore there is an energy balance to be struck between moving small amounts of cold air and large amounts of tempered or ambient air.

An obvious problem with using outside air ventilation without mechanical cooling as a means of cooling a building is that the temperature of the outside air is generally higher than the inside temperature at the times when cooling is most necessary. This can partly be remedied by using overnight cooling, see section 4.7. However, this is less energy efficient than daytime cooling, and the benefits of natural as opposed to mechanical night cooling would need to be considered.

2.5.3 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 full 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, in the context of the suitability of natural ventilation, see Figure 2.3, 'tight' temperature control is defined as ± 1 K (air temperature) and 'close' control of humidity as better than ± 10%. specific circumstances may require more precise control, e.g. 25% RH and ± 1 K, or ± 2% RH and ±0.5 K in critical areas. It is therefore important for the client and designer to have agreed these parameters.

Various options are available for both the generation and distribution/delivery of cooling. Traditional mechanical refrigeration and alternative means of generating cooling are considered in sections 4.6 to 4.22, which also provide design information for the full range of distribution systems. Key distribution system components are discussed in section 5.

A broad categorisation of heating, ventilation and air conditioning (HVAC) systems is given in Table 2.7. However, 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
- include 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, it may also be affected by the existing building services. Table 2.8 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.

Table 2.7 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, with fine-tuning of the supply temperature or volume occurring at the terminals	VAV and its variants, dual-duct, hot deck-cold deck
Air/water or air/refrigerant	Usually employing central plant to provide fresh air only, terminals being used to mix re-circulated air with primary air and to provide fine-tuning of the room temperature	Fan coils, 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

2.5.4 Mixed mode systems

Mixed mode may be defined as the strategic combination of natural and mechanical ventilation and/or cooling systems.

Sub-classifications of mixed mode systems are as follows[29]:

- ***Contingency designs:*** these are usually naturally ventilated buildings which have been carefully planned to permit the selective addition of mechanical ventilation and cooling systems where this is needed at a subsequent date. However the converse has also occurred.
- ***Complementary systems:*** natural and mechanical systems are both present and are designed for integrated operation. This is the most common variety of mixed mode. Systems can operate in a concurrent manner, i.e. simultaneously, in a changeover manner, i.e. on a relatively frequent basis, or alternately, i.e. a less frequent version of changeover.
- ***Zoned systems:*** these allow for differing servicing strategies to occur in different parts of the building. The zoned approach works best where the areas are functionally different, or where the systems are seamlessly blended.

A flow chart illustrating the selection process is provided in Figure 2.4.

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. Some selection issues are raised below, for further details see section 4.5.

Selection issues to be considered include:

- ***Costs:*** both capital and operating costs are highly variable. A balancing factor is the degree to which supplementary mechanical systems have been installed.
- ***Maintenance:*** requirements for maintenance, and its complexity, are dependent upon the approach chosen. Poor designs could result in an excessive burden due to the need to maintain both mechanical and, possibly, automated natural ventilation systems.

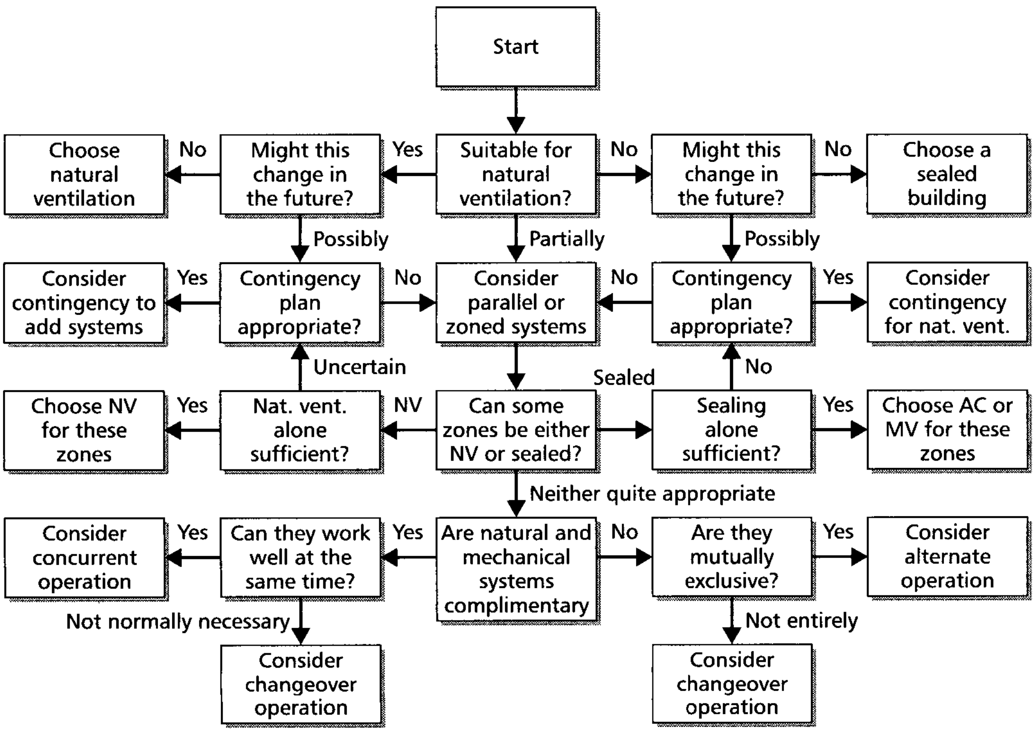


Figure 2.4 Mixed mode selection chart

Table 2.8 Possible system assessment criteria

Criterion	Comments
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
Ventilation and cooling performance	Ability to be zoned Risk of draughts Noise generation Maximum cooling load that can be handled Ability to cope with frequent variations in load Ability to cope with semi-permanent variations in load Potential for use in mixed mode systems
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 ventilation 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

- ***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 window design to be less complicated and more robust than in buildings designed to rely upon natural ventilation alone.
- ***Energy efficiency:*** in relation to fully air conditioned buildings, mixed mode systems should use less energy for fans, pumps and cooling. However this is dependent upon the savings in mechanical plant that have been attained.
- ***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, i.e. occupants can adjust the windows and the engineering systems can be managed.

References

1 ***Testing buildings for air leakage*** CIBSE TM23 (London: Chartered Institution of Building Services Engineers) (2000)

2 ***Environmental design*** CIBSE Guide A (London: Chartered Institution of Building Services Engineers) **(1999)**

3 ***Project management handbook for building services*** BSRIA Applications Guide **11/98** (Bracknell: Building services Research and Information Association) **(1998)**

4 ***Flexible building services for office based environments - Principles for designers*** CIBSE TM27 (London: Chartered Institution of Building Services Engineers) (2001) (available to CIBSE members only; this publication is included on the CIBSE members' CD-ROM)

5 ***Whole Life Costing - A Clients' Guide*** Construction Clients Forum 2000 (London: Confederation of Construction Clients) (2000)

6 BS ISO 15686-1: *Buildings and constructed assets. Service life planning:* Part 1 : 2000: *General principles* (London: British Standards Institution) (2000)

7 *Whole life costs* Construction Procurement Guide No. 7 (London: Office of Government Commerce) (available in electronic format only from www.ogc.gov.uk)

8 *Flexible buildingservicesfor office based environment - Principles for clients* CIBSE Briefing Note (London: Chartered Institution of Building Services Engineers) (2001)

9 Nanayakkara R and Fitzsimmons J *Cost benchmarks for the installation of building services* BSRIA Applications Guide 20-1/99CD (Bracknell: Building Services Research and Information Association) (1999)

10 Hayden G W and Parsloe C J *Value engineering of building services* BSRIA Applications Guide AG15/96 (Bracknell: Building services Research and Information Association) (1996)

11 Parsloe C *Design formaintainability* BSRIA Applications Guide AG 11/92 (Bracknell: Building services Research and Information Association) (1992)

12 Crozier B *Enhancing the performance on oversized plant* BSRIA Applications Guide AG1/2000, (Bracknell: Building services Research and Information Association) (2000)

13 *Energy Assessment and Reporting Methodology* CIBSE TM22 (London: Chartered Institution of Building Services Engineers) (1999) (this publication is included on the CIBSE members' CD-ROM)

14 *Energy use in offices* (ECON 19); *Energy efficiency in hotels* (ECON 36); *Energy consumption in hospitals* (ECON 72); *Energy consumption guide for nursing and residential homes* (ECON 57) DETR Energy Consumption Guides (Garston: Building Research Energy Conservation Support Unit) (various dates) (these publications are included on the CIBSE members' CD-ROM)

15 *Energy benchmarking in the retail sector* BMI Index Special Report 281 (London: Building Maintenance Information) (July 1999)

16 *BREEAM 98for offices - an environmental assessment methodfor offices* BREEAM 98 (1998); *New industrial units - an environmentalassessmentfornewindustrial, warehousingandnon-food retail units series* BREEAM 5/93 (1993); *An environmental assessmentfor new superstores and new supermarkets* BREEAM 2/91 (1991) (Garston: Building Research Establishment) (dates as indicated)

17 *Environmental code ofpractice for buildings and their services* BSRIA COP 6/99 (Bracknell: Building services Research and Information Association) (1999)

18 Bordass W T, Bromley A K R and Leaman A J *Comfort, control and energy efficiency in offices* BRE Information Paper IP3/95 (Garston: Building Research Establishment) (1995)

19 Bordass W T, Bunn R, Cohen R and Leaman A J The Probe Project: technical lessons from PROBE 2 *Proc. CZBSE Nat. Conf, Harrogate, October 1999* (London: Chartered Institution of Building Services Engineers) (1999) (PROBE studies are published occasionally in *Building Services Journal)* (PROBE studies are included on the CIBSE members' CD-ROM)

20 *Energy efficiency in buildings* CIBSE Guide F (London: Chartered Institution of Building Services Engineers) (1999) (this publication is included on the CIBSE members' CD-ROM)

21 *Natural ventilation in non-domestic buildings* CIBSE AM10 (London: Chartered Institution of Building Services Engineers) (1997)

22 *Minimising pollution at air intakes* CIBSE TM21 (London: Chartered Institution of Building Services Engineers) (1999)

23 *Daylighting and window design* CIBSE Lighting Guide LGlO (London: Chartered Institution of Building Services Engineers) (1999)

24 Baker N V and Steemers K *The LT Method 2.0: an energy design tool for non-domestic buildings* (Cambridge: Cambridge Architectural Research) (1994)

25 *Code for interior lighting* (London: Chartered Institution of Building Services Engineers) (1994)

26 Slater A I, Bordass W T and Heesman T A *People and lighting controls* BRE Information Paper 6/96 (Garston: Building Research Establishment) (1996)

27 Braham D, Barnard N and Jaunzens D *Thermal mass in office buildings* BRE Digest 454 (Garston: Building Research Establishment) (2001)

28 BS 5250: 1989: *Code ofpracticefor the control ofcondensation in buildings* (London: British Standards Institution) (1989) (under revision)

29 *The Building Regulations 2000 Approved Document L: Conservation of fuel and power* (interim draft) (London: Department of Transport, Local Government and the Regions) (March 2001)

30 *Mixed mode ventilation* CIBSE Applications Manual AM13 (London: Chartered Institution of Building Services Engineers) (2000)

3 Requirements

3.1 Introduction

A strategic consideration of requirements as part of an integrated approach to design is outlined in section 2. Section 3 provides a more detailed discussion of the development of ventilation requirements for office environments. Differences to this 'standard' approach are then outlined for other building sectors. More specifically it addresses:

— general functional requirements for the application of ventilation and air conditioning in offices to provide a safe and healthy working environment (section 3.2.1)

— general functional requirements for the application of ventilation and air conditioning in offices to provide a comfortable working environment (section 3.2.2)

— general functional requirements for the application of ventilation and air conditioning systems in terms of protecting the building fabric (section 3.2.3) and energy use (section 3.2.4)

— specific requirements for other building sectors (sections 3.3 to 3.24), see Table 3.1.

3.2 Offices

The following requirements apply to offices and to a wide range of other buildings. Requirements specific to other types of buildings are given in sections 3.3 to 3.24.

3.2.1 Indoor air quality: basic requirements for health and safety

The issue of improving air quality in offices (and buildings in general) has previously been mainly related to sick building syndrome (SBS). However recent work[1] has suggested that SBS is not linked to the type of ventilation or air conditioning system used but is more likely to be a function of how well systems are installed, managed and operated. It suggested that workspaces conforming to CIBSE guidelines on temperature and air movement should not suffer from SBS, unless there are aggravating work-related factors or extreme levels of pollution.

In designing any ventilation system it is necessary to understand the functions required of it. For office environments these are:

— to supply sufficient fresh air

— to provide adequate indoor air quality by removing and or diluting pollutants from occupied spaces

— to provide a heat transport mechanism.

The last has been considered on a strategic basis in section 2 and is considered in terms of thermal comfort in section 3.2.2. This section examines the function of maintaining indoor air quality in order to:

— support human respiration

— remove body odour

— remove tobacco smoke

— remove emissions from building materials and furnishings e.g. volatile organic compounds (vocs)

— prevent radon gas entering a space via foundations and air intakes

— support safe and efficient operation of combustion appliances

— allow smoke clearance in the event of fire.

3.2.1.1 Basis of requirements

Ventilation may be used to dilute or displace and remove airborne contaminants released in a space and which would otherwise rise to unacceptable concentrations. Within the Building Regulations, guidance on achieving compliance of relevance to the designer of ventilation systems includes:

— *Approved Document F: Ventilation*[2]*;* Part F1: *Means of ventilation;* Part F2: *Condensation in roofs*

— *Approved Document J: Heat producing appliances*[3]

— *Approved Document B: Fire safety*[4]

— *Approved Document L: Conservation of fuel and power*[5]

Note that if dilution is the main basis of control then the ventilation system should be designed to produce good mixing of the incoming air with the contaminant within the space. In situations where the contaminant release is from a fixed source then it is preferable to arrange the extract location as close to the source as possible so that direct removal is achieved. Requirements will also be affected by the ventilation efficiency, i.e. whether all the fresh air supplied is used or whether some is extracted prematurely. See section 4.2.2 for further consideration.

Section 1 of CIBSE Guide A: *Environmental design*[6] should be consulted for the definition of, and requirements for achieving, suitable indoor air quality standards. It describes two methods for determining suitable outdoor air ventilation rates:

— a prescriptive method

— a calculation method for the control of a single known pollutant being released into the space at a known rate.

Table 3.1 Summary of recommendations

Building sector	Section number	Recommendation
Animal husbandry	3.24.1	See Table 3.20
Assembly halls	3.3	See Table 3.6
Atria	3.4	See section 3.4.3
Broadcasting studios	3.5	6-10 ACH (but heat gain should be assessed)
Call centres	3.24.2	4-6 ACH (but heat gain should be assessed)
Catering (inc. commercial kitchens)	3.6	30—40 ACH
Cleanrooms	3.7	See Tables 3.11 and 3.12
Communal residential buildings	3.8	0.5-1 ACH
Computer rooms	3.9	See Table 3.13
Court rooms	3.24.3	As for typical naturally ventilated buildings
Darkrooms (photographic)	3.24.4	6-8 ACH (but heat gain should be assessed)
Dealing rooms	3.24.5	As offices for ventilation (but heat gain should be assessed)
Dwellings (inc. high-rise dwellings)	3.10	0.5-1 ACH
Factories and warehouses	3.11	See 3.11.1 for regulatory requirements
High-rise (non-domestic) buildings	3.12	4-6 ACH for office areas; up to 10 ACH for meeting spaces
Horticulture	3.24.6	30-50 $litre.s^{-1}.m^{-2}$ for greenhouses (45-60 ACH)
Hospitals and health care buildings	3.13	See Table 3.15
Hotels	3.14	10-15 ACH minimum for guest rooms with en-suite bathrooms
Industrial ventilation	3.15	Sufficient to minimise airborne contamination
Laboratories	3.16	6-15 ACH (allowance must be made for fume cupboards)
Museums, libraries and art galleries	3.17	Depends on nature of exhibits
Offices	3.2	See Tables 3.2 and 3.3
Plant rooms	3.18	Specific regulations apply, see section 3.18
Schools and educational buildings	3.19	See Table 3.18
Shops and retail premises	3.20	5-8 $litre.s^{-1}$ per person
Sports centres (inc. swimming pools)	3.21	See Table 3.19
Standards rooms	3.24.7	45-60 ACH
Toilets	3.22	Building Regulations apply; opening windows of area l/20th. of floor area or mechanical ventilation at 6 $litre.s^{-1}$ per wc or 3 ACH minimum for non-domestic buildings; opening windows of area 1/20 th. of floor area (l/30th. in Scotland) or mechanical extract at 6 $litres^{-1}$ (3 ACH in Scotland) minimum for dwellings
Transportation buildings (inc. car parks)	3.23	6 ACH for car parks (normal operation) 10 ACH (fire conditions)

These are summarised below. A third method has been suggested[7,8] for use where pollution sources are known but not their emission rates or limiting concentrations or consequences.

Prescribed outdoor air supply rate

The prescriptive method is based on chamber studies where tobacco smoke and body odour were considered to be the only pollutant sources. This may result in an underestimate of requirements if other pollutants are present.

Calculation method for control of a specific pollutant

Where a single known pollutant is being released into the space at a known rate, the calculation is based on risk assessments under the Control of Substances Hazardous to Health (COSHH) Regulations[9]. The Health and Safety Executive (HSE) publishes annual guidance[10] on the limits to which exposure to hazardous airborne substances should be controlled in workplaces. This is in the form of occupational exposure limits (OELS) for long-term (8 hour) and short-term (10 minute) exposures. OELS are available for a large number of substances. While these concentration limits must not be exceeded, it is recommended that exposure should be kept as low as is reasonably practical. Compliance with these limits is a fundamental requirement of the COSHH Regulations.

For situations where exposure may be longer than 8 hours a day, or where more susceptible members of the general population, such as the elderly, the young and those prone to ill-health, are involved values lower than the OELS should be applied. It has been suggested that one fifth of the OEL might be an acceptable standard, although limited information is available. Further guidance on pollutant levels for the general population is also available from the World Health Organisation[11].

For a single contaminant under steady conditions, equation 3.1 may be applied to determine the flow of outdoor air that, with good mixing, would maintain the contaminant concentration at a specified level.

$$Q = \frac{q\,(10^6 - C_i)}{(C_i - C_o)} \qquad (3.1)$$

where Q is the outdoor air supply rate (litres^{-1})} q is the pollutant emission rate (litre.s^{-1}), C_o is the concentration of pollutant in the outdoor air (ppm) and C_i is the limit of concentration of pollutant in the indoor air (ppm).

This equation can be adapted for:

— pollutant thresholds quoted in mg.m^{-3} and situations where C_i is small or the incoming air is free of the pollutant in question, see CIBSE Guide A[6], section 1.7.3.1

— situations where the ventilation results in a non-uniform concentration so that a higher than average concentrations exist in the occupied zone and the outdoor air supply rate requires to be increased, see CIBSE Guide A[6], sections 1.7.3.1 and 1.7.4

— non-steady state conditions that might allow the outdoor air supply rate to be reduced, see CIBSE Guide A[6], section 1.7.3.2.

A more comprehensive analysis of the relationship between contaminant concentration and ventilation rate is given in BS 5925[12].

Note that the existing guidelines for the calculations of outside air ventilation rates are based on the assumptions that the air outside the building is 'fresh' and that the pollutant load is inside the building. For buildings in city areas or adjacent to busy roads the quality of the outside air needs to be assessed, as this can also be a source of pollutants. Where specific problems are anticipated, an air quality survey should be undertaken. This should include measurements at likely times of peak pollution (e.g. the rush hour).

The use of natural ventilation means that it is much more difficult to clean the air entering the building. Mechanical ventilation and air conditioning systems can filter the incoming air to remove dust and dirt, but only specialised air treatment can remove gaseous pollutants (e.g. oxides of carbon and nitrogen from traffic fumes). In all building types, gaseous pollutants can be minimised by careful siting of ventilation inlets, see section 4.3 and CIBSE TM 21[13].

In mechanically ventilated buildings, effective air filtration relies on good maintenance[14]. Poor filtration performance can allow dirt and dust to accumulate within a ductwork system, reducing the efficiency of heat exchange equipment and providing potential sites for microbiological activity. Spores and bacteria can then be released into the occupied space, causing potential comfort and health problems. Natural ventilation systems, on the other hand, are generally more accessible for cleaning and maintenance, and there are no components subject to high humidity, such as cooling coils, or humidifiers, which can harbour biological growth.

As well as assessing external air quality, the sources of internal pollution should also be reviewed so that their effect can be minimised or even eliminated. Ventilation should not be used in place of source control to minimise pollutant concentrations in a space.

Calculation method for control of multiple pollutants

There is no accepted approach for the derivation of exposure limits for mixtures of contaminants, although some guidance is given in EH40[10]. In such cases it is recommended that specialist assistance be sought from occupational hygienists or toxicologists. Likewise, guidance currently only exists for a small number of substances in terms of acceptable limits to avoid sensory, as opposed to health, effects[11]. In practice, the exposure of workers in non-industrial environments to these same concentrations of contaminants would not be acceptable and a multiplying factor of 0.1 has been suggested.

A method to deal with the dilution of pollution from non-human sources has been suggested[7,8], see equation 3.2:

$$Q_c = \frac{10\,G}{E_v\,(P_i - P_o)} \qquad (3.2)$$

where Q_c is the outdoor air supply rate to account for the total contaminant load (litre.s^{-1}), G is the sensory pollution load (olf), E_v is the ventilation effectiveness, P_i is the design perceived indoor air quality (decipol) and P_o is the perceived outdoor air quality (decipol). These units are defined elsewhere[7,8].

However, this proposal is still subject to discussion and has not yet gained international acceptance.

3.2.1.2 Human respiration

Carbon dioxide is a dense odourless gas produced by combustion and respiration. The rate of ventilation required for the supply of oxygen for breathing is far outweighed by any requirement for the dilution of exhaled carbon dioxide (CO_2). A build-up of this gas in a room leads to a feeling of stuffiness and can impair concentration. Elevated levels of CO_2 in the body cause an increase in the rate of respiration. Slightly deeper breathing begins to occur when the atmospheric concentration exceeds 9000 mg.m^{-3} or 5000 ppm (0.5% by volume). This is the maximum allowable concentration of CO_2 for 8-hour exposures by healthy adults[10]. In the USA, one half of this limit (0.25%) has been taken as appropriate for general building environments[15].

These figures are based on sedentary occupations; minimum ventilation rates for various activity levels to prevent these limits being exceeded are given in Table 3.2.

For most applications involving human occupancy, the CO_2 limits shown in Table 3.2 are not usually taken as a design criterion as much more air needs to be provided to

Table 3.2 Ventilation rates required to limit CO_2 concentration for differing activity levels

Activity	Minimum ventilation requirement / (litre.s^{-1} per person)	
	0.5% CO_2 limit	0.25% CO_2 limit
Seated quietly	0.8	1.8
Light work	1.3-2.6	2.8-5.6
Moderate work	2.6-3.9	N/A
Heavy work	3.9-5.3	N/A
Very heavy work	5.3-6.4	N/A

meet other criteria such as the dilution of body odours or tobacco smoke.

Within the UK, a CO_2 figure of 800-1000 ppm is often used as an indicator that the ventilation rate in a building is adequate. One thousand parts per million would appear to equate to a 'fresh air' ventilation rate of about 8 litre.s^{-1} per person. In Sweden, the equivalent indicator is 1000 ppm, with a desired level of 600-800 ppm. Note that as outside air itself contains carbon dioxide (approx. 350 ppm), a 50% reduction in internal levels from 1600 ppm to 800 ppm requires a four-fold increase in ventilation rate.

3.2.1.3 Body odour

The ventilation rate required depends on whether the criterion is (a) acceptability to the occupants or *(b)* acceptability to visitors entering the occupied space. In studies on auditoria[16], it was found that that the occupants themselves were insensitive to changes in ventilation over the range 5-15 litre.s^{-1} per person, although there were always nearly 10% of the occupants dissatisfied with the odour level.

Similarly, it has been shown that an outdoor flow rate of 7 to 8 litre.s^{-1} per person is required to restrict the level of body odour so that no more than 20% of the entrants to the occupied space were dissatisfied. The sensitivity was such that halving the ventilation rate increased the proportion dissatisfied to 30%, while more than three times the ventilation rate was required before the proportion decreased to 10%.

Therefore in the absence of further information, it is recommended that 8 litre.s^{-1} per person should be taken as the minimum ventilation rate to control body odour levels in rooms with sedentary occupants. There is evidently a relationship between CO_2 concentration and body odour intensity in occupied rooms. Thus for intermittent or varying occupancy, the control of ventilation rates by CO_2 concentration monitoring can be effective in matching the supply of air supply to the changing requirements.

3.2.1.4 Tobacco smoke

The suggested outdoor air supply rate of 8 litre.s^{-1} is based on sedentary occupants and the absence of any other requirements, e.g. the removal of moisture. This is consistent with the requirements for the removal of body odour but assumes the absence of any smoking. There are no definitive criteria for the required dilution of tobacco smoke. Uncertainties relate particularly to the respirable paniculate component (see section 3.2.1.6). Evidence suggests that particle removal by filtration is necessary to avoid excessively high ventilation rates.

Table 3.3 Recommended outdoor air supply rates for sedentary occupants[6]

Level of smoking	Proportion of occupants that smoke / %	Outdoor air supply rate / (litre.s^{-1} per person)
No smoking	0	8
Some smoking	25	16
Heavy smoking	45	24
Very heavy smoking	75	36

Smoking also produces undesirable odours, particularly to non-smokers. One study[17] has shown that filtration of the smoke particles did not alleviate the odour nuisance, indicating that much higher rates of ventilation are now required to avoid dissatisfaction of more than 20% of visitors to a room occupied by cigarette smokers. Ventilation rates for smokers of 4 or 5 times that required for non-smokers have been suggested although, allowing for the fact that a minority of the occupants may be smokers, the overall ventilation rate may be only twice that needed for non-smoking situations.

If smoking is prohibited, then the rate for 'no smoking' may be used, see Table 3.3. For the other situations described in the table, it has been assumed that each smoker present consumes an average of 1.3 cigarettes per hour. It should be noted that, regardless of the ventilation rate used, the health risks of cigarette smoke cannot be completely eliminated. It is recommended that designers consult current guidelines, such as those issued by the Health and Safety Executive[18], and ensure that clients are made aware of any risks involved in the chosen design strategy. Legal advice may also be advisable.

3.2.1.5 Volatile organic compounds (vocs)

vocs cover a wide range of compounds having boiling points in the range of 50-260 °C and hence existing in vapour form at room temperature. They are particularly prevalent in new and recently refurbished buildings, coming from a variety of sources including:

- people, animals, plants
- consumer products (cleaning agents, paints, glues, solvents etc.)
- building materials and treatment (damp-proofing, furnishings etc.)
- building services and other equipment
- outdoor air.

Analysis is normally restricted to measuring the total voc content in air. ASHRAE Standard 62[19] suggests that complaints are unlikely to arise for total voc concentrations below 300 mg.m^{-3}, whereas above 3000 mg.m^{-3} complaints are likely. Details of the appropriate ventilation provision can be found in section 1 of CIBSE Guide A[6].

3.2.1.6 Respirable particles (PM_{10})

Respirable particles are those constituents of the air that are not in purely gaseous form. They can be ingested into the lungs while breathing and cause a wide range of health problems. The most potentially dangerous particulates are asbestos fibres but there are concerns about other 'man-made mineral fibres' (MMMF) which are widely used for insulation within buildings. Particulate matter is monitored in the UK as PM_{10}, i.e. particles generally less than 10 microns in diameter. A large number of epidemiological studies have shown that day-to-day variations in concentrations of particles are associated with adverse effects on health from heart and lung disorders, and a worsening of the condition of those with asthma.

Details of the appropriate ventilation provision can be found in section 1 of CIBSE Guide A[6].

3.2.1.7 Radon

Radon is a colourless and odourless radioactive gas. It comes from the radioactive decay of radium, which in turn comes from the decay of uranium. Radon is emitted from uranium-bearing soils and emission rates therefore vary depending on the geological conditions of the location. Radon is implicated in the cause of lung cancer. Protection from exposure to radon at work is specified in the Ionising Radiation Regulations[20], made under the Health and Safety at Work etc. Act[21]. A limit for radon in non-domestic buildings has been set at 400 $Bq.m^{-3}$, above which action must be taken to reduce the concentration. Guidance on appropriate action can be found in BRE report BR 293[22].

3.2.1.8 Combustion appliances and products

Adequate fresh air must be supplied to meet the requirements for combustion in fuel burning appliances. Details of these requirements are laid down in BS 6798[23], BS 5410[24] and BS 5440[25]. Part J of the Building Regulations, with its associated Approved Document[3], also governs flues from gas fired combustion appliances of up to 60 kW and from solid fuel and oil burning appliances of up to 45 kW. For guidance on how to ventilate larger installations, i.e. boiler houses and plant rooms, refer to section 3.18.

Guideline values for concentrations of combustion products are given in CIBSE Guide A, section 1, Table 1.8[6]. The most common are nitrogen dioxide (NO_2), sulphur dioxide (SO_2), and carbon monoxide (CO). These may either be created within the occupied space or may re-enter buildings, e.g. from chimney smoke or from the exhausts of cars through windows overlooking car parks.

3.2.1.9 Gas and refrigerant detection methods

Gas detection methods are dealt with in section 3.18. Refrigerant detection methods are also considered in that section, with further guidance in the case of split systems in section 4.21.

3.2.1.10 Smoke control and clearance

Ventilation for the control of smoke in the event of a fire, and its subsequent clearance, is a specialist subject. Guidance is given in CIBSE Guide E: *Fire engineering*[26]. If natural ventilation is to be achieved by means of an atrium, guidance is also available in BRE Report BR 375[27].

3.2.2 Ventilation for internal comfort

3.2.2.1 Temperature

CIBSE Guide A[6], Table 1.1, gives recommended summer and winter dry resultant temperatures corresponding to a mean predicted vote of ±0.25 for a range of building types. However, as noted in Guide A, control within an air conditioned building is normally based on a response to internal air temperatures. In a standard office environment this corresponds to 22-24 °C and 21-23 °C where comfort cooling or air conditioning, respectively, are available. In a naturally ventilated environment, the acceptable dry resultant temperature range is less well defined and various approaches have been suggested, see Table 3.4.

Section 1.4.3 of CIBSE Guide A[6] considers factors that influence the criteria for comfort cooled or air conditioned spaces. A summary of the factors most related to the design of the ventilation or air conditioning systems is given in Table 3.5. However, CIBSE Guide A should be consulted for detailed guidance.

Table 3.4 Alternate approaches to design criteria for naturally ventilated offices

Criterion	Source
Mean temperature during occupied periods with acceptable deviation, e.g. mean summer dry resultant temperature of 23±2 °C in an office with a formal dress code, and 25 ±2 °C in an office with an informal dress code	BRE *Environmental design guide*[28]
Thresholds never to be exceeded, e.g. *(a)* a maximum temperature of 27 °C; *(b)* the internal temperature is never to exceed the external temperature.	CIBSE AM10[29]
A threshold that can be exceeded for a specified period, e.g. *(a)* dry resultant temperature not to exceed 25 °C for more than 5% of the occupied period; *(b)* dry resultant temperature not to exceed 25 °C for more than 5% of the occupied period or 28 °C for more than 1% of the occupied period.	CIBSE Guide A[6]; BRE *Energy Efficient Office of the Future specification*[30]

3.2.2.2 Humidity

The role of humidity in maintaining comfortable conditions is discussed in section 1.5 of CIBSE Guide A[6]. An acceptable range of 40-70% RH is suggested. However, to minimise the risk of mould growth or condensation and maintain comfortable conditions, a maximum design figure of 60% RH is suggested for the design of air conditioning systems. Within naturally ventilated buildings, humidity levels as low as 30% RH (or lower) may be acceptable for short periods of time, but care is needed to restrict airborne irritants such as dust or tobacco smoke. Precautions should also be taken to avoid shocks due to static electricity through the specification of equipment and materials, e.g. carpets.

3.2.2.3 Internal gains

In the absence of information from the client, the British Council for Offices recommends the following allowances for internal gains when specifying ventilation systems[31]:

— solar gains not to exceed 60-90 $W.m^{-2}$ depending upon façade orientation

— occupancy based upon 1 person per 12 m^2, but diversified wherever possible to 1 person per 14 m^2 at the central plant

— lighting gains of not more than 12 $W.m^{-2}$ at the central plant

Table 3.5 Factors in office environments influencing thermal comfort relating to ventilation or air conditioning system design[6]

Factor	Issues to be considered	Guide A section number
Humidity	Little effect on feelings of warmth for sedentary, lightly clothed people at dry resultant temperatures of 23 °C and below. If room humidity is greater than 70% the risk of condensation and microbiological growth may be increased. Dust mite levels may also increase with high humidity.	1.4.3.1
Clothing	The insulation value of clothing (i.e. clo value) can influence the acceptable dry resultant temperature for *sedentary* occupants, e.g. in the case of a thick pullover, a reduction of 2.1 K.	1.4.3.2
Activity	The metabolic rate (hence heat generated) is affected by activity. For people dressed in normal casual clothing (clo = 0.5-1.0), an increase of 0.1 met corresponds to a possible reduction of 0.6 K in the recommended dry resultant temperature.	1.4.3.3
Temperature changes	A smooth change in dry resultant temperature should be aimed at to avoid discomfort.	1.4.3.4
Adaptation and climate	The theory of adaptive thermal comfort, i.e. that the preferred internal temperature is affected by the prevailing external conditions, is still being debated.	1.4.3.5
Age	The requirements of older people for higher temperatures are thought to be associated with their generally lower activity levels.	1.4.3.6
Gender	The requirements of women for slightly higher temperatures are though to be related to their generally lower clo values.	1.4.3.7
Occupants' state of health, disability, and physical condition	Little is know about this factor, although higher temperatures are usually required for bed-ridden or immobilised people due to their lower met and clo values.	1.4.3.9
Draughts	The influence of mean relative air speed on the thermal comfort of occupants is dependent partly upon the temperature of the moving air (see predicted mean vote (PMV), Guide A[6], section 1.4.2.2), the air flow rate, and its direction. An excessive air flow rate can give rise to complaints of draughts, especially in winter. The back of the neck is particularly susceptible. If the room air speed exceeds 0.15 $m.s^{-1}$ the dry resultant temperature should be increased from its still value. An air speed of >0.3 $m.s^{-1}$ is not recommended, unless it is in a naturally ventilated building where it is specifically for cooling. Dissatisfaction with draughts is also affected by fluctuations in air speed. These are defined by the turbulence intensity (TI) and consequently a calculated draught rating (DR), which should not exceed 15%. The relative air speed over a body's surface increases with activity. If activity levels exceed 1 met, 0.3 $m.s^{-1}$ should be added to the air speed relative to a stationary point	1.4.3.10
Vertical air temperature differences	The gradient in either direction (floor to ceiling and vice versa) should be no more than 3 K in the occupied zone. If air velocities are higher at floor level than across the upper part of the body, then a maximum gradient of 2 $K.m^{-1}$ is recommended.	1.4.3.11
Asymmetric thermal radiation	This is affected by the proximity to adjacent cold surfaces e.g. single glazed windows, adjacent hot surfaces e.g. overhead radiant heaters and the intrusion of short wavelength radiation e.g. solar radiation through glazing.	1.4.3.14

— office equipment gains of not more than 15 $W.m^{-2}$ when diversified and measured over an area of 1000 m^2 or more, but with an ability to upgrade to 25 $W.m^{-2}$. Local workstation levels are quoted as typically 20-25 $W.m^{-2}$.

3.2.3 Ventilation of building fabric to avoid interstitial condensation

Many structures are vulnerable to interstitial condensation, which can cause rotting of wood-based components, corrosion of metals and reduction in the performance of thermal insulation. Condensed water can also run or drip back into the building causing staining to internal finishes or damage to fittings and equipment. The traditional view has been that these problems are caused by water vapour generated in the building diffusing into the structure. Avoidance measures have therefore concentrated on the inclusion of a vapour control layer on the warm side of the structure, appropriate placing of insulation, or ventilating the structure to intercept the water vapour before it can condense. Ventilation is specifically required in cavities above the insulation in cold pitched and flat roofs, behind the cladding of framed walls and below timber floors.

Many problems can occur from water entrapped within materials, moving within the structure under diurnal temperature cycles. Under these circumstances it is helpful to distinguish between 'ventilated' and 'vented' air spaces. A ventilated space is designed to ensure a through flow of air, driven by wind or stack pressures whereas a vented space has openings to the outside air that allow some limited, but not necessarily through, flow of air. As the air in the space expands and contracts under diurnal temperature cycles, water vapour will be 'breathed' out of the structure. This mechanism can be very effective in large span structures where it can be very difficult to ensure effect through ventilation of small cavities.

Detailed design guidance for the provision of ventilation within structures is available in various publications including CIBSE Guide A[(6)], BS 5250[(32)] and BRE Report BR 262[(33)].

3.2.4 Energy use

Energy use in offices has risen in recent years because of the growth in information technology, air conditioning (sometimes specified when not required), and intensity of use.

However, this trend is offset by considerable improvements in insulation, plant, lighting and controls. The Energy Efficiency Best Practice programme has produced ECON 19: *Energy use in offices*[(34)]. This provides benchmarks, based on data gathered in the 1990s, which take account of increasing levels of IT provision for four types of office buildings:

— naturally ventilated cellular

— naturally ventilated open-plan

— standard air conditioned

— prestige air conditioned.

Despite perceptions to the contrary, energy-efficient offices are not expensive to build, difficult to manage or inflexible in their operation. Nor do they provide low levels of comfort or productivity. Energy-efficient techniques that work well tend to be reliable, straightforward, and compatible with the needs of the building operator and occupants. Capital costs are often similar to those for normal offices, although budgets may be spent differently; for example, on measures to reduce cooling loads rather than on air conditioning.

Further opportunities for improving energy efficiency should be sought when other changes occur, e.g. refurbishment, fit-out, alteration, and plant replacement. Forthcoming Building Regulations[(35)] and the associated Approved Document L[(36)] will require much greater attention to energy issues during refurbishment, as the scope of the regulations in England and Wales has been widened to bring such activity within the meaning of controlled work and material change. The Scottish regulations are currently being revised and it is anticipated that they will adopt a similar approach. Best results in terms of energy efficiency are obtained when there is a good brief, good design with attention to detail, sound workmanship and commissioning, and good control and management.

At the time of writing (July 2001), office fuel bills can range from about £4 to £30 per year per square metre of treated floor area, excluding VAT and Climate Change Levy. Good practice in energy efficient office design can reduce energy costs by a factor of two. ECON 19[(34)] gives details of the characteristics of best practice energy efficient design, as well as details of the benchmarks for the four office types. Careful attention to energy efficiency should be a constant theme of the design of the ventilation and air-conditioning of a building.

3.3 Assembly halls and auditoria

3.3.1 General[(37,38)]

Assembly halls and auditoria, e.g. theatres, concert halls, conference centres, places of worship, are generally characterised by large but variable occupancy levels, relatively high floor to ceiling heights, sedentary occupation, and stringent acoustic requirements. Whilst being characterised by these criteria, places of worship tend to be serviced with a low cost, simple solutions, often naturally ventilated with a simple background heating system.

Specific issues that need to be addressed for assembly halls and auditoria include the following:

— flexibility of the space being served and if the seating is fixed or removable

— acoustic control measures including plant location, vibration, noise break-out, fan noise, silencers, flexible connections, duct linings, etc.

— integration of relatively large air handling plant and distribution ductwork

— occupancy patterns and part load operation
— viability of heat recovery devices and possible variable speed operation
— zoning of the plant (for large auditoria)
— treatment and integration of builders' work plenums (including control and zoning)
— air terminal device selection, integration with seats, control of draughts and noise regeneration
— stage ventilation and cooling and assessment of lighting heat gains
— temperature control at rear of auditorium due to reduced height
— background heating, and heating for non-occupied hours
— cooling and ventilation to control rooms, translation booths, etc.

3.3.2 Design requirements

Normal design requirements for buildings are shown in Table 3.6. Mechanical ventilation systems for assembly halls and auditoria need to be designed to meet the sound control requirements of CIBSE Guide B5: *Sound control* [39]

Table 3.6 Design requirements: assembly halls and auditona

Parameter	Design requirement
Fresh air ventilation rates	To suit occupancy levels
Air change rate	3-4 air changes per hour for displacement strategy
	6-10 air changes per hour for high level mechanical strategy
Temperature and humidity:	
— heating only	20 °C; 40% RH (minimum)
— with cooling	20-24 °C; 40-70% RH

3.3.3 Strategies

3.3.3.1 Mechanical ventilation — low level supply, high level extract

Low level supply is often via a plenum beneath the seating. Air is extracted at high level, returned to the central plant for heat recovery or exhausted to atmosphere. This approach is suitable for raked fixed seating halls and auditoria. Displacement-type room air distribution strategies are often used. The advantages are that only the occupied zone is conditioned, not the entire space and the potential for 'free cooling' is maximised as supply air temperatures are usually 19-20 °C. Air volumes and energy consumption and maintenance costs are usually less when compared with high level supply, although central plant sizes are normally similar.

3.3.3.2 Mechanical ventilation (high level supply and extract)

This system is usually selected where a flexible space is required, seating is removable, or where it is not feasible or prohibitive in terms of cost to provide under-seat plenums.

3.3.3.3 Natural ventilation

Supply is by attenuated inlet builders' work ducts at low level and high level. Extract is by attenuated outlets at high level, relying on stack effect to ventilate and cool the area. This approach has potentially the lowest running costs but may require a number of provisions to ensure adequate airflow rate and to limit peak temperatures in summer. Particular considerations include providing suitable air paths, inlets and exhaust positions, solar protection, mass exposure and night cooling.

3.3.3.4 Ventilation control

Options for ventilation control strategies include:

— demand controlled ventilation and cooling depending upon *(a)* return air carbon dioxide levels[40], *(b)* occupancy levels
— space temperature and humidity
— time control
— night-time purging of the space and possible pre-cooling of structure.

3.3.4 Further reading

Displacement ventilation BSRIA TM2/90 (Bracknell: Building Services Research and Information Association) (1990)

Demand controlled ventilation BSRIA LB30/93 (Bracknell: Building Services Research and Information Association) (1993)

Carbon-dioxide controlled mechanical ventilation systems BSRIA TN12/94.1 (Bracknell: Building Services Research and Information Association) (1994)

Mansson L G and Svennberg S A *Demand controlled ventilation systems — source book* IEA Annex 18 Report (Paris: International Energy Agency) (1992)

3.4 Atria

3.4.1 General

The incorporation of an atrium into a building will not automatically lead to energy savings, especially if the atrium requires artificial lighting and air conditioning (often for the health of the planting as much as the occupants)[41]. However, if well designed, an atrium can bring the advantages of:

— enhanced opportunities for natural ventilation by stack effect and allowing air to be drawn from both sides of the building towards a central extract point
— preheating of ventilation air
— additional working space.

3.4.2 Requirements

3.4.2.1 Environmental conditions

Environmental conditions within an atrium are dependent upon the degree of comfort required. Saxon[42] defines four categories of atrium:

— simple unenclosed canopy or enclosure without comfort control

— basic buffer space with partial control to assist plants

— tempered buffer space with partial control to assist in achieving some degree of human comfort

— full comfort atrium.

3.4.2.2 Buoyancy driven ventilation (mixed and displacement)

Many atria are sealed and mechanically ventilated and, sometimes, mechanically cooled. However, natural ventilation can provide high rates of air change and also induce cross ventilation of the surrounding office areas. Natural ventilation is driven by wind pressure and thermal buoyancy. The limiting case is likely to be buoyancy alone, i.e. when there is no breeze.

There are two kinds of buoyancy driven ventilation, defined by the position of the openings[43]:

— mixing ventilation

— displacement ventilation.

In mixing ventilation, openings are placed at the top of the atrium only; warm air leaves the atrium reducing the pressure and allowing cool air to enter via the same opening. The cool, dense air falls to the floor mixing with the warm air as it falls. This results in the air temperature at floor level being above ambient by an amount depending on the size of the opening; the larger the opening the smaller the difference between the inside and outside temperatures. Mixing ventilation leads to a relatively uniform vertical temperature distribution.

In displacement ventilation, openings are placed at the top and bottom of the atrium; warm air leaves the upper opening and cooler air enters the lower opening. Assuming a steady input, equilibrium is reached where a stationary boundary exists between the warm air at high level and the cool air at lower level. Reducing the size of the openings lowers the position of this boundary and increases the temperature of the upper zone but the temperature of the lower zone remains at, or close to, the ambient temperature.

In many situations displacement ventilation is appropriate for summer conditions. To promote ventilation the air in the atrium should be as warm as possible over the greatest proportion of the atrium height. In most atria occupation occurs at floor level, excluding galleries and staircases. Therefore it is important to keep the temperature at floor level as low as possible. However if the atrium is open to the surrounding space, or if it provides high level walkways, the high temperatures in these occupied spaces might become unacceptable. The design strategy should therefore be based on the absorption of solar radiation by surfaces above the occupied space. The position of the stationary boundary is important; ideally the hot layer will be confined to a level above adjacent occupied spaces. This suggests that atria should have sufficient height to ensure that this will occur.

Displacement ventilation can be used to reject heat when the outside temperature is below the atrium temperature. At night, heat retained in the massive elements of the atrium will generate stack effect to provide useful night cooling. However, it is possible for the temperature in the atrium to fall below the ambient temperature and thereby cause a reversal of the stack effect.

3.4.2.3 Atrium openings

For displacement ventilation driven by the stack effect, openings will be required at the top and bottom of the atrium of between 5 and 10% of the roof glazing area[43]. For atria with large areas of vertical glazing facing between south and west, the openable areas should be a similar percentage glazing area. The more shading that can be provided, the smaller the openings need to be for a given thermal performance.

3.4.2.4 Roof vents

Roof vents must be carefully positioned within the form of the roof so that positive wind pressures do not act on the outlets causing reverse flow[29]. It is normally possible to arrange the outlets such that they are always in a negative pressure zone. This may be achieved by:

— designing the roof profile so that for all wind angles the openings are in a negative pressure zone

— using multiple vents that are automatically controlled to close on the windward side and open on the leeward side.

3.4.2.5 Ventilation enhancement and fire safety

On hot, still days natural ventilation can be supplemented by extract fans in the atrium roof. Subject to fire office approval, a combination of natural and powered ventilation can also form part of the smoke control or clearance system. It is essential that fire conditions be considered at an early stage so that the possibility and benefits of a dual-purpose system can be evaluated. Guidance on fire safety and atria is available elsewhere[4,26,44,45].

3.4.2.6 Flexibility

The designer should be aware of any intention to use the atrium area for other purposes, e.g. evening concerts or the provision of catering, when selecting the ventilation strategy.

3.4.3 Strategies

3.4.3.1 Types of atrium

Saxon[42] defines three types of atrium with regards to their thermal properties, see Figure 3.1. These are:

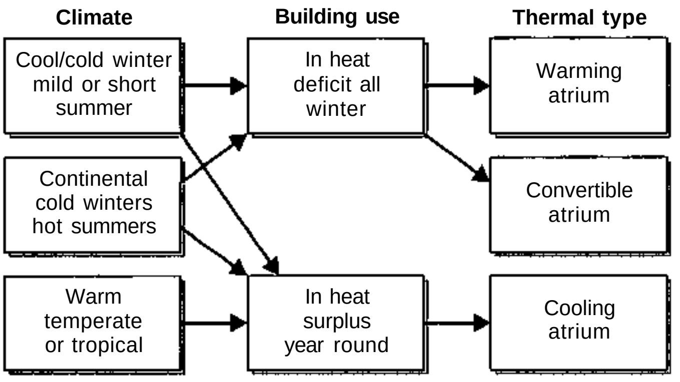


Figure 3.1 Selection of thermal type of atrium

— *warming atrium:* which normally collects heat

— *cooling atrium:* which normally rejects heat

— *convertible atrium:* which changes mode according to the season.

The purpose of the atrium will be affected by climate and building use as this impacts on internal heat gains within the adjacent accommodation.

Warming atrium in winter

A warming buffer atrium is normally designed to admit heat freely (from solar gain or the surrounding accommodation) and will therefore tend to be at higher than ambient temperatures. Even if the atrium is unheated, its temperature in winter will be above the ambient. This may be used as a means of pre-warming ventilation air. Unless the atrium has a low protectivity[43] (i.e. the ratio of separating wall area to the atrium external envelope area), the temperature at night should be maintained above the night set-back, given the flow of heat from the building to the space and stored heat in walls and the floor. The chosen ventilation strategy will affect the heating energy consumption.

Air circulation is desirable, even in winter, to avoid cold air stratifying at the ground level where people pass through. Additional heat can be gained within the atrium by coupling its ventilation system with that of the accommodation, air being discharged into the atrium after heat recovery. If full comfort is sought then coupling becomes even more advantageous. This can be achieved by using the atrium as a return air plenum. This allows solar gains to be collected and food smells to be contained.

Warming atrium in summer

The main concern in summer is to prevent overheating. The primary means of achieving this is through shading[46]. External shading is more effective than internal shading and movable devices can prevent the loss of useful daylight. The stack effect can be used to induce ventilation either of the atrium alone or of the whole building.

Cooling atrium

The function of the atrium is to provide a source of cooling for the surrounding accommodation. This cooling can be as a result of night cooling within the atrium creating a thermal buffer zone. More commonly the atrium is used as a supply or return air plenum.

Convertible atrium

A convertible atrium will function in a similar manner to the warming atrium in winter, but require more protection against overheating in summer to avoid the need for impractical ventilation rates. Pre-cooling of the atrium space may also be employed to reduce the temperature of any radiant surfaces.

3.4.3.2 Ventilation modes

Saxon[42] defines five possible ventilation modes:

— complete separation of ventilation for the atrium and that for the occupied space

— intake of primary air via the atrium and the rest separate

— exhaust of used clean air into the atrium, the rest separate

— use of the atrium as a supply air plenum to occupied spaces

— use of the atrium as a return air plenum.

The advantages and disadvantages of these modes, with regard to the chosen degree of comfort, are shown in Figure 3.2.

3.4.4 Calculation of atrium performance

A suitable choice of thermal model should be made. Guidance on the prediction of winter conditions in atria at an early stage of the design is available[43]. Models for calculating ventilation flow rates such as SERI-RES, DOE 2 and BREEZE are listed in the *European Passive Solar Handbook*[46].

	Comfort type				
	Canopy	Buffer	Temp. buffer	Full comfort	Thermal type
No vent. relationship	Normal	Normal	Possible	NA Behaves as separate room	Warming Convertible Cooling
Intake via atrium	No effect	OK summer NA OK winter	OK summer NA OK winter	NA	Warming Convertible Cooling
Exhaust to atrium	Slight effect	Useful	Useful	NA	Warming Convertible Cooling
Atrium as supply plenum	NA	NA	NA	Possible Useful	Warming Convertible Cooling
Atrium as return plenum	NA	NA	Collects solar useful	Useful	Warming Convertible Cooling

Figure 3.2 Selection of ventilation mode

3.5 Broadcasting studios (radio and television)

3.5.1 General

The general requirement is to provide a comfortable environment within the constraints imposed by the production of television programmes. Specific issues that need to be addressed include:

- high lighting loads in studios
- high occupancies for shows with audiences
- rapid changes in load
- variable operation times and periods
- sensitivity to air movement and noise
- high equipment loads in technical areas
- critical areas requiring a high degree of reliability
- 24-hour operation
- multiplicity of studio arrangements
- adaptability to respond to changing technological and business requirements.

3.5.2 Design requirements

Tables 3.7 and 3.8 provide some typical design requirements. The loads given apply to the working area only. The areas identified provide typical examples and not intended to be exhaustive. Arenas are often used for bigger shows. Small presentation studios may be used for linking programmes; these are subject to similar loads but of short duration and are normally occupied all day by a presenter.

Mechanical ventilation systems for broadcasting studios need to provide a high level of reliability, as the system is critical to the proper functioning of the building and the business conducted within it. Consequential losses arising from failure can be very significant in this type of building.

Television studios may have 2-3 lighting rigs to suit different requirements. For programmes such as news and current affairs, which have less need to create visual interest and that run all day, fluorescent lighting may be used in addition to tungsten, thereby reducing the lighting load.

Heat loads can be highly intermittent. Game shows last for only ½ to 1 hour. During this time the lighting may be brought up and down on the audience. For drama studios, only one of a number of sets may be fully lit at a time. Setting up studios for shows can take several days, during which time loads will be low.

Within the occupied zone near floor level environmental conditions should be 21 ±1 °C, rising to 23 ±1 °C at times of peak load. Relative humidity should ideally be between 40% and 60%. Humidity control is not normally required in the UK. Achieving good humidity control in studios can be problematic due to the rapid load changes. Close control of conditions may be required in tape storage areas to reduce deterioration. It is preferable that tapes are stored under the same environmental conditions as the

Table 3.7 Typical design requirements: broadcasting studios — general areas

Description	Size / m^2	Occupancy	Noise level	Heat loads / $W.m^{-2}$	
				Lighting	Equipment
Flexible studio (light entertainment)	Up to 2000 (typically 400;)	10 crew 50-100 audience	NR20-NR25 NR30 with audience	500,200 over seating	100
Drama studio	150-2000	4-10	NR15	500 over rds. of floor area at any one time	100
Fixed rig studio (e.g. news and current affairs)	150	4-10	NR20-NR25	200	100
Radio studio	5-30	1-10	NR15	20	70

Table 3.8 Typical design requirements: broadcasting studios — technical areas

Description	Size / m^2	Occupancy	Noise level	Equipment heat load
Control room:				
— production*	50	8-10	NR20-NR30	6-8 kW; 70 $W.m^{-2}$
— vision*	50	8-10	NR20-NR30	4-8 kW; 70 $W.m^{-2}$
— sound	16	2-3	As for studio	2-4 kW
Voice-over booth	2.5	1	NR15-NR20	2-4 kW
Editing room:				
— equipment outside room	—	1-4	NR20-NR30	2-6 kW
— equipment within room		1-4	NR35-NR40	1-8 kW
Central apparatus room	†	N/A	NR45	Typically 750-1000 $W.m^{-2}$
Transmission room	†	N/A	NR30	Up to 500 $W.m^{-2}$

* These areas may be combined
† Dependant on size of facility

room in which the tapes are to be used; this minimises sticking and problems due to static electricity.

Air speeds in television studios should be in the order of 0.2 $m.s^{-1}$, but not higher than 0.3 $m.s^{-1}$ in order to avoid visual disturbance of hair, clothing, scenery drapes and dry ice, and noise in microphones. Air movement is critical for drama studios.

Mechanical ventilation systems for broadcasting studios need to be designed to meet the sound control requirements of CIBSE Guide B5: *Sound control*[39]. Typical noise level criteria are given in Tables 3.6 and 3.7. Reference should also be made to the noise criteria established in the BBC's *Guide to acoustic practice*[47]. Noise is particularly critical in drama studios and 'quality' radio studios. As in other applications, background noise from a ducted air system provides a degree of masking of extraneous noise from adjacent areas. If the background noise level is substantially lower than the criterion set, then the extraneous noise normally masked by the ventilation may become apparent.

3.5.3 Strategies

Systems need to be able to cope with high loads and to the rapid changes in load. Air based systems are often preferred due to concerns over water within the space. Central plant may be preferred to local units due to restrictions on access for maintenance.

Variable air volume systems may provide an energy efficient solution for television studios. Constant volume systems provide an even airflow at a constant noise level, which may be important for technical reasons, but can be wasteful of energy in large installations.

Blow-through coils with airside damper control may be preferred to waterside control to respond to rapid load changes. Overcooling can be a problem if response is too slow. Steam injection may be used for fast response to meet humidity requirements.

High reliability for critical areas is normally provided by redundancy on individual units and/or in the number of units provided. Dual power supplies and generator back-up are also generally provided. High loads can lead to rapid temperature rises (that may, in turn, activate sprinklers). Systems should also be designed so that they can be readily adapted to respond to changing requirements.

To separate audience and performance areas for control purposes, studios may be zoned into quartiles by multiple damper assemblies.

Attenuation should be provided to reduce ingress of noise from outside and from central plant. Noise from balancing dampers can be a particular problem and should be avoided if possible. Air speeds inside the studio are critical with regard to noise, see section 3.5.2. Particular problems can arise with boom microphones located close to high-level supply diffusers, both due to noise from the diffuser and wind-generated noise from excessive air movement.

False floors are normally provided in studios but are not generally used for air supply since they are normally filled with cabling, including PVC cables. Acoustic and fire-break issues also need to be addressed.

Equipment heat gains in technical areas may be treated directly by providing dedicated supply and/or extract ducts to the equipment cabinets.

Radio studios are the most critical areas with regard to noise levels. Constant volume systems are preferred while the studio is in use. Where cooling loads are relatively low, cooling systems such as displacement ventilation and chilled ceilings may be used.

Where areas are occupied 24-hours a day, consideration must be given to how the systems will be maintained without loss of cooling or ventilation while the studio is in use.

3.6 Catering and food processing

3.6.1 Kitchens

3.6.1.1 General

Adequate ventilation in catering premises is required for the following purposes:

— To introduce sufficient clean, cool air and remove excess hot air in order for the occupants to breathe and remain healthy and comfortable. Often it is not possible to achieve normal comfort conditions in kitchens because of the difficulties of counteracting the heat released from appliances. Under these circumstances care should be taken to ensure that acceptable working conditions are not breached.

— Provide sufficient air for complete combustion in appliances to prevent carbon monoxide levels exceeding 300 ppm for 10 minutes[9] or 10 ppm as an average over 8 hours[11], and to dilute and remove combustion products.

— Dilute and remove odours, vapours and steam from the cooking process

Local ventilation must be kept clean from fat residues to avoid loss of efficiency and minimising the risk of fire.

Research by the HSE on exposure of kitchen and factory workers to cooking fumes reinforces the importance of providing and maintaining good ventilation in catering kitchens and industrial cooking areas, particularly where meat, fish and cooking oils are directly heated. A fundamental requirement of the Control of Substances Hazardous to Health (COSHH) Regulations[9] is that employers should prevent the exposure of their employees to hazardous substances or, where that is not reasonably practicable, ensure that there is adequate control of hazardous substances. The fumes generated by directly heating foods during frying, grilling and stir-frying have been identified as containing small quantities of carcinogens. Although deemed to be adequate, available information on this issue is limited at the time of writing, making it impossible to state conclusively that no risk exists with current controls. It is therefore important that fume extraction systems are provided and maintained to current standards. Designers should ensure that they are aware of latest revisions to any related guidance.

3.6.1.2 Requirements

Canopy extract

Air needs to be removed from cooking and subsidiary areas at a constant rate to take away combustion fumes and cooking odours as close to the source as possible. It is advisable that the bulk of extraction from the kitchen is via hoods above gas-fired and all other appliances capable of generating heat, water vapour, fumes and odours.

It is recommended that the plan dimensions of the canopy exceed the plan area of cooking appliances. An overhang of 250-300 mm all round is normally adequate for island canopies. Wall-mounted canopies normally have a overhang of 250 mm at the front and 150 mm at the sides. Greater overhangs may be required at some appliances.

Canopies and ductwork need to be constructed from non-combustible materials and fabricated so as not to encourage the accumulations of dirt or grease, nor to allow condensation to drip from the canopy. The ductwork needs suitable access for cleaning and grease filters need to be readily removable for cleaning or replacement.

The amount of air extracted via the canopies should be calculated from the information supplied with the particular appliances, and not based simply on general advice or overall air change rate. Where details of the equipment are known, HVCA specification DW 171[48] describes a method for calculating the ventilation requirement whereby each cooking appliance is allocated a thermal convection coefficient. This is the recommended volume of air to be extracted in $m^3 \cdot s^{-1}$ per m^2 of surface area of the appliance. The area of each appliance is multiplied by the coefficient for that appliance and the values for each item of equipment under the canopy are added together to determine the total volume to be extracted. The factor will vary depending on whether the appliance is fired by gas or electricity.

Where the ventilation requirements of the individual cooking appliances are not available, an approximate air flow rate can be calculated from the total hood size, canopy area and hood face velocity, as follows:

$$Q_{\mathrm{hood}} = 1000 \times A_{\mathrm{hood}} \times V_{\mathrm{hood}} \quad (3.3)$$

where Q_{hood} is the approximate hood air flow rate (litre.s^{-1}), A_{hood} is the canopy area (m^2) and V_{hood} is the hood face velocity (m.s^{-1}). Table 3.9 provides typical hood face velocities.

Table 3.9 Hood face velocities

Cooking duty	Hood face velocity (m.s^{-1})
Light	0.25
Medium	0.4
Heavy	0.5

Ventilated ceiling extract

Where ventilated ceilings are used in place of canopies, the ventilation rates should be calculated taking into account room size and function. As a guide, a ventilation rate of not less than 17.5 litre.s^{-1} per m^2 of floor area and not less than 30 air changes per hour (ACH) is advisable. A lower air change rate may be needed to avoid discomfort from draughts where the kitchen is divided into separate rooms. The Heating and Ventilating Contractors' Association recommends that a general ventilation rate of 40 ACH be used in areas of larger kitchens not treated by canopies.

Replacement air

If the kitchen is in a sealed area (i.e. not adjacent to dining areas) replacement air should comprise typically 85% supplied by mechanical ventilation and 15% by ingress of air from the surrounding areas. This ensures that the kitchen is maintained under a negative pressure to prevent the escape of cooking odours. In basement areas containing kitchens and restaurants, the supply plant to the restaurant areas should be sufficient to offset the down-draught from street level in addition to supplying air to the kitchens.

If non-air conditioned, properly ventilated restaurants adjoin the kitchens, the majority of air may be drawn from the dining area. If the restaurant is air conditioned, air may be drawn from it at a maximum of 7 litre.s^{-1} per person. The difference between the extract and replacement air should be provided by a separate kitchen supply system.

Air drawn from adjacent areas should be clean. It is not advisable to draw make-up air from rooms where smoking is allowed. Where make-up air is drawn via serving hatches or counters it is recommended that air velocities do not exceed 0.25 m.s^{-1} to avoid complaints of draughts. However, higher velocities may be tolerated or desirable at hot serving counters. The make-up air can be drawn in through permanent grilles if the serving hatches are small or likely to be closed for long periods. These should be sized on the basis of 1.0-1.5 m.s^{-1} airflow velocity.

The incoming air from the ventilation system needs to be arranged so as not to affect adversely the performance of flues associated with open-flued gas appliances[49].

In smaller kitchens sufficient replacement air may be drawn in naturally via ventilation grilles in walls, doors or windows. Provision should be made to prevent pest entry by using a fine mesh in the grille; however, it may be necessary to compensate for restrictions in the airflow by increasing the size of the grille.

Cooling air

The effective balancing of incoming and extracted air, together with removal at source of hot vapours, should prevent the kitchen from becoming too hot. The air inlets from any mechanical ventilation systems can be positioned to provide cooling air over hot work positions. Extra provision may be required, either by an overhead outlet discharging cool air or by air conditioning. Local free standing fans are not recommended due to health and safety considerations and their effect on the efficiency of the designed ventilation systems.

Discharge

High level discharge of extracted air, with discharge velocities of about 15 $m.s^{-1}$, are often needed to prevent nuisance to neighbouring properties. The design of the discharge stack should prevent down-draughts and re-entry of fumes into the building.

3.6.2 Food processing

3.6.2.1 General

Food processing covers cooking, preservation and packing. Normally, mechanical ventilation, and sometimes air conditioning, will be required.

3.6.2.2 Requirements

The designer should take into account the heat dissipation based on the energy used in the production process and should make an approximate heat balance for the calculation of air quantities. The ventilation of special food manufacturing processes will need detailed consideration in consultation with food production specialists/managers. Plant may need to be designed to meet individual requirements; for example, a fairly closely controlled temperature is necessary in sweet and chocolate manufacture and local cooling is an essential part of the manufacturing process.

In cooking areas the general guidance given in section 3.6.1 applies. In addition to local ventilation, general ventilation will be necessary. It is preferable to supply air over working areas and extract over cooking equipment or other high heat dissipation areas, but care must be taken to avoid local excess cooling of the processes.

Regular maintenance of kitchen ductwork is essential to reduce the risk of fire(48). Ductwork should be routed in a manner that will enable routine cleaning to be carried out. Drains may be necessary in some cooling processes, as may fire dampers and grease filters.

3.6.3 Further reading

The main health and safety law applicable to catering HSE Catering Information Sheet No 11 (Bootle: Health and Safety Executive) (2000)

Douglas P *Kitchen Planning and Design — Theory* (London: Blandford) (1979)

3.7 Cleanrooms

3.7.1 General

A cleanroom is a room in which the concentration of airborne particles is controlled to specified limits and which is constructed and used in a manner to minimise the introduction, generation and retention of particles within the room. Cleanrooms are classified according to the maximum permitted number of particles of a certain size. Commonly used classifications are given in BS EN ISO 14644-1(50) and FS209E(50), see Table 3.10. The appropriate classification must suit the work that is to be undertaken and it is often the nature of the work that will dictate the arrangement of the ventilation systems.

The Medicines Control Agency (MCA), which publishes the *Rules and Guidance for Pharmaceutical Manufacturers and Distributors*(52) (known as the 'orange book'), uses the FS209E classifications and, in addition, sets limits for microbiological contamination. Classifications may also relate to 'as built', 'at rest' and 'in operation' states.

The appropriate classification must be agreed with the client as the cleanroom suites will often require validation in terms or air change rates, particle counts and other environmental criteria.

Information on the design of cleanrooms is available within the series of *Baseline Guides* produced by the International Society for Pharmaceutical Engineering*.

Table 3.10 Comparison of cleanroom classifications

USA Federal Standard 209E(51)	BS EN ISO 14644-1(50)	MCA(52) ('at rest')
—	1	—
	2	
1	3	—
10	4	—
100	5	A or B
1000	6	—
10000	7	C
100000	8	D

3.7.2 Design requirements and strategies

Generally, the design of the ventilation systems must take account of the following factors, which will need to be agreed with the client:

— classification, i.e. 'at rest' or 'in operation'

— nature of work, e.g. semiconductor/electronics or pharmaceutical

— laminar or turbulent flow requirements

— minimum air change rates

— pressure differentials

— room construction, fabric leakage rates and other air paths

— HEPA filtration standards

— room layout, including fixed furniture and equipment

— open or closed door design

— controls and alarms

— validation requirements.

*International Society for Pharmaceutical Engineering, 3816W Linebaugh Avenue, Suite 412, Tampa, Florida 33624, USA (http://www.ispe.org)

Table 3.11 Design guidance for non-laminar-flow clean rooms

Parameter	Value for achievable class (USA Federal Standard 209E)(51)		
	1000	10000	100000
Room pressure differential to adjacent areas	15 Pa	15 Pa	5-10 Pa
Ventilation rate (depending on type of work)	40-120 ACH	20-40 ACH	10-20 ACH
Clean air inlet area as a percentage of ceiling area (typically for 'in operation' status)	20-50	10-20	5-10
Terminal velocity at clean air inlet	0.15-0.45 $m.s^{-1}$	0.15-0.45 $m.s^{-1}$	0.15-0.45 $m.s^{-1}$
Return locations	Low level or floor	Low side wall	Side wall or ceiling
Wall return spacing	Continuous on all four walls	Intermittent on long walls	Non-uniform
Return face velocities	0.5-1 $m.s^{-1}$	1-2.5 $m.s^{-1}$	2.5 $m.s^{-1}$

Note: Air supply may be drawn from outside or recirculated, subject to client requirements

Mechanical ventilation systems for cleanrooms need to provide a high level of reliability, as the system is critical to the proper functioning of the building and the business conducted within it. Consequential losses arising from failure can be very significant in this type of building.

Filters are one of the major influences on the level of cleanliness in cleanrooms, but must not be considered in isolation. The method used to supply air to the room is a crucial factor, along with how the room is used in operation. The location of fixed furniture, equipment and workstations needs to be considered as they affect airflow patterns and create dead zones within the room. Wherever possible the product should be upstream of the operative. The cleanest zone is the area in immediately in front of the HEPA filter and the product should be in this zone if possible. There should special clothing for operatives with changing rooms etc. Variable speed fans should be used to maintain constant airflow when HEPA filters become dirty. Clean benches are frequently used to upgrade a section of the clean room or carry out work in a normal working area.

Air can be supplied by laminar- or non-laminar-flow methods. Airflow patterns may need to be controlled or located so that the cleanest air can be directed across workstations where the tasks are actually performed.

Non-laminar-flow cleanrooms can achieve up to USA Federal Standard 209E class 1000, whilst laminar-flow clean rooms can achieve class 1 in 'in operation' state. Turbulent-flow clean rooms may achieve higher classifications in 'at rest' state. Non-laminar-flow systems can achieve FS209E 'at rest' class 100 (MCA grade B). Such systems are common in pharmaceutical applications.

In non-laminar-flow clean rooms, air is supplied to the room by individually ducted HEPA filter modules or air diffusers in the ceiling. Alternatively, an in line HEPA filter housing installed in the supply duct as close to the room as possible can be used. The grade of HEPA filter specified will need to suit the room classification. Air should be exhausted through grilles in the walls near the floor as there is no requirement on uniformity of airflow patterns.

Air velocities must ideally be between 0.15 and 0.45 $m.s^{-1}$; lower velocities allow contamination to settle out, high velocities allow contamination to agglomerate.

For non-laminar-flow cleanrooms, observation of certain design criteria is essential. Table 3.11 provides general design guidance for non-laminar-flow clean rooms.

In laminar-flow cleanrooms, air enters the room through filters covering the whole ceiling (downflow) or on one wall (crossflow), and is exhausted through the entire opposite surface, with air flowing in parallel lines and at uniform velocity. Thus, air makes only one pass through the room and any contamination created in the room is carried out. Velocities of 0.45 $m.s^{-1}$ are necessary to prevent settling out. Such rooms are costly to construct and it may be appropriate to subdivide the room into areas having different classifications according to the processes being undertaken. Due to the quantities of air being circulated some form of recirculation should be considered to reduce energy costs. Table 3.12 provides general design guidance for laminar-flow clean rooms.

3.7.3 Further reading

Whyte W *Cleanroom Design* (London: Wiley) (1991)

Table 3.12 Design guidance for laminar-flow clean rooms

Parameter	Value for achievable class (USA Federal Standard 209E)(51)	
	1 and 10	100
Room pressure	15 Pa	15 Pa
Ventilation rate	500-600 ACH	500 ACH
Clean air inlet area as a percentage of ceiling area	90-100%	90%
Terminal velocity at clean air inlet	0.15-0.45 $m.s^{-1}$	0.15-0.45 $m.s^{-1}$
Return locations	Perforated wall/floor	Low level or floor

3.8 Communal residential buildings

3.8.1 General[53]

Communal residential properties are buildings containing separate residential units with some degree of communal facilities. For the purposes of this Guide, the following have been considered:

— residential care homes

— student accommodation

— military barracks.

As with domestic properties, effective ventilation is best provided by reducing air leakage, extracting moisture and pollutants at source, and providing occupant controllable ventilation. Natural ventilation is particularly suitable for achieving this.

3.8.2 Requirements

Overall ventilation rates of between 0.5 and 1 air change per hour are generally appropriate.

Wherever possible, residents should be able to maintain autonomy and control over their immediate environment. In the case of student accommodation the emphasis is on dealing with intermittent occupation and appropriate integration with heating system controls. In residential care homes occupancy is less intermittent and control of the heating and ventilation is likely to be more centralised under the control of a warden.

For communally shared facilities within residential care homes and student accommodation, it will be necessary to make different arrangements for areas of higher occupancy (e.g. television rooms) or areas of excessive moisture or odour generation (e.g. laundry rooms, and cafeteria areas) requiring ventilation direct to the outside.

Within residential care homes it may be necessary to service conservatories, which should, if possible, be separated from other living spaces by doors to prevent excessive heat loss in winter. External draught lobbies or revolving doors should be specified for all major entrances/exits.

In both types of accommodation the needs of smokers may affect the chosen system design, in particular the servicing of smoking lounges. However, as stated in section 3.2.1.4, it should be noted that the provision of ventilation cannot completely remove the health risks associated with cigarette smoke.

3.8.3 Strategies

The required ventilation rates can be achieved by using trickle vents with passive stack ventilation (PSV)[54] systems or extract fans in kitchens and bathrooms. Alternatively, whole-building ventilation systems with heat recovery (MVHR) can be used if the building is well sealed. CIBSE TM23: *Testing buildings for air leakage*[61] recommends an air leakage index of 8 $Pa.m^{3}.h^{-1}$ at 50 Pa as good practice for dwellings with balanced whole-house mechanical ventilation and 15 $Pa.m^{3}.h^{-1}$ at 50 Pa for dwellings with mechanical ventilation. Best practice standards for such dwellings are 4 and 8 $Pa.m^{3}.h$ at 50 Pa, respectively.

3.8.4 Further considerations

The maintenance implications of MVHR systems must be considered[55], as must the consequences of system failure if there is no passive ventilation back-up. Guidance on system optimisation is available, see section 4.4 and elsewhere[56].

3.9 Computer rooms

3.9.1 General

Under operational conditions, computer equipment is susceptible to the temperature, humidity and the cleanliness or otherwise of the surrounding environment. Computer rooms have a number of specific characteristics that need to be taken into account when selecting and designing ventilation and air conditioning systems. These generally include:

— 24-hour operation

— high sensible loads (typically 500 to 1000 $W.m^{-2}$)

— low occupancy and latent loads

— close control of temperatures and humidity required

— high levels of reliability required, with some redundancy to ensure 24-hour operation

— deep raised floors to deal with extensive cabling

— noise levels generally above those for offices due to the computer equipment

— capability for expansion to allow for frequent upgrading of computer equipment

— mainframe computers with tight temperature control requirements may require dedicated chilled water systems.

Mechanical ventilation systems for computer rooms need to provide a high level of reliability, as the system is critical to the proper functioning of the building and the business conducted within it. Consequential losses arising from failure can be very significant in this type of building.

It is particularly important to establish the required loading of the space, the specific requirements of any mainframe computer and the capability for expansion as these are subject to wide variations.

To minimise the effect of the external environment, computer suites are generally provided with highly insulated walls, floors and roofs, and no windows. The building structure should be airtight and vapour-sealed to facilitate close control. Air locks may also be provided at entrances. In many instances computer rooms will normally operate with the lighting off for much of the day.

Heating should be provided in critical areas to maintain a suitable minimum temperature under winter conditions during computer shutdown.

Computer rooms can be grouped into three approximate size categories:

— *small:* in offices, typically 1% of area served, often less critical than larger computer rooms; telephone equipment rooms
— *medium:* IT-intensive organisations, such as financial organisations with dealing facilities, typically 1-2% of area served on the floors plus 2-5% for a main computer room
— *large:* stand-alone data centres; switching centres.

3.9.2 Design requirements

Typical design requirements for computer rooms are shown in Table 3.13.

Requirements should be checked with equipment manufacturers as wider control bands and higher temperatures may be permissible.

3.9.3 Strategies

To provide close control of temperature and humidity, specialist computer room air conditioning units are normally provided. These units generally include:

— cooling coil (DX, glycol or chilled water)
— reheat coil (usually electric due to limited use)
— humidifier (typically steam due to straightforward maintenance and health and safety requirements)

Table 3.13 Typical design requirements: computer rooms

Parameter	Requirements
Internal temperature	To suit computer equipment: typically 21 ± 2 °C; rate of change not to exceed 3 $K.h^{-1}$
Internal relative humidity	50 ± 5% RH; rate of change not to exceed 10% in 1 hour
Filtration	To suit computer equipment: typically 60% efficiency to BS EN 779[57]
Noise criteria	NR55 (range NR45-NR65)
External temperatures	Design temperatures based on a 1% failure rate may not be acceptable; heat rejection plant in particular requires careful selection to ensure it can perform in practically all conditions
Internal heat gains	600 $W.m^{-2}$ sensible (range 500-1000 $W.m^{-2}$)
Ventilation	Computer rooms are generally pressurised by oversupply (1 ACH typical) to prevent infiltration gains and local variations in temperature and humidity; otherwise minimum fresh air to suit occupancy.

— filtration (panel filters)
— fans (single or multiple combinations dependent on duty)
— compressors (DX and glycol units only).

The units can be mounted within the computer room or in service corridors adjacent and come in a variety of sizes. Various degrees of sophistication are possible depending on the reliability required from the individual units. The most usual arrangement is a wardrobe-type unit with common fan drives, controls, heater battery, cooling coil and humidifiers. Reliability is then improved by incorporating redundant units. Alternatively 'modular' units can be used with common controls but individual fans, heaters, cooling coils and even humidifiers in each module, so that a module failure has little effect on the overall performance.

To a large extent, the choice of the type of cooling will be determined by the size of the computer room and the availability, or otherwise, of chilled water, DX cooling is generally used in smaller rooms where chilled water is not readily available 24 hours a day. The DX cooling coil rejects heat through external air cooled condensers. On large installations the proliferation of air-cooled condensers tends to present an unacceptable solution.

Glycol systems are based on a DX cooling coil in the room unit with heat rejection into a glycol closed water system. Dry air coolers are used to reject heat from the glycol system either centrally or on an individual unit-by-unit basis. An additional 'free cooling' coil can be added to the room unit to allow it to operate without running the compressors when the external ambient temperature is low. Glycol systems are generally used for large computer rooms where 'free cooling' can save significant amounts of energy.

Chilled water room unit cooling coils fed from a central chilled water system may be used in smaller rooms where chilled water is available 24 hours a day, and in larger rooms where simplicity of the room unit may have a benefit.

A high sensible cooling ratio is an important consideration for any selected unit to minimise the operation of the cooling coil and humidifier together. Elevated chilled water temperatures (e.g. 10-16 °C) may be used for this reason. The higher temperatures also provide the energy benefit of increased central refrigeration plant efficiency.

Common controllers can be provided but it is usual for each unit to be separately controlled to cater for variations in gains across the computer room. Common central monitoring of the alarms is usual.

To improve system redundancy, dual pipework systems may be used. Generator back-up for the cooling system is normally provided in critical applications. This may be a 'no-break' facility where high loads would give an unacceptable temperature rise between power failure and the generators coming on-line.

Air supply is normally through the ceiling or floor. Supplying air at low level and extracting over the computer equipment has the advantage that the heat released upwards from the equipment can more easily be removed without it affecting the occupied areas. High level supply may be through diffusers or a ventilated ceiling.

Consideration should be given to the operating and maintenance requirements of the installation. Temperature and humidity recording/alarm devices may be necessary together with other operational alarms. Locating equipment in an adjacent service corridor may be preferred for critical/sensitive applications as this will reduce maintenance access requirements to the space.

3.9.4 Further reading

Jones W P *Air Conditioning Applications and Design* (London: Edward Arnold)

Data Processing and Electronic Office Areas Chapter 16 in ASHRAE Handbook: *HVAC Applications* (Atlanta GA: American Society of Heating, Refrigerating and Air Conditioning Engineers) (1999)

3.10 Dwellings (including high rise dwellings)

3.10.1 General

Fresh air supplies within dwellings are necessary for:

— the health and safety of the occupants

— the control of condensation, often the dominant pollutant arising from moisture generated by cooking, washing and clothes drying

— the removal of odours

— the removal of pollutants such as vocs

— the removal of allergens arising from dust mites

— the safe and efficient operation of combustion appliances.

As moisture is the most significant pollutant, its control forms the basis of the ventilation strategy. The key is to avoid a situation where the relative humidity exceeds 70% for a prolonged period[58]. This can usually be achieved with a whole house ventilation rate of 0.5 air changes per hour[59]. Alternatively, more rapid extraction in response to moisture release within the dwelling, either by humidity sensors or manually, can be beneficial in removing moisture before it is absorbed by furnishings and/or the fabric of the building itself[60].

In domestic situations, it is particularly important to inform occupants of the intended operation and purpose of the selected ventilation system to ensure that it achieves its intended purpose. This will ensure that they:

— do not tamper with the system in the belief that it is costing them money to run

— do not interfere with the performance of the system through blocking air inlets or extracts, or by altering sensor settings.

3.10.2 Requirements

As with non-domestic buildings, the underlying concept should be to 'build tight, ventilate right'[59]. Detailed guidance on requirements and acceptable ventilation solutions (e.g. trickle ventilators) can be found in Approved Document F[2]. Guidance on achieving an airtight construction can be found in CIBSE and BRE publications[61,62].

Figure 3.3[63] illustrates the impact of uncontrolled air leakage on the ventilation rate. The greater the air leakage the greater the ventilation rate and the more varied and uncontrollable it will be. Air leakage must often be reduced to bring the overall ventilation rate within the prescribed range. The airtightness of UK dwellings can range from 2 ACH to above 30 ACH at an applied pressure of 50 Pa. This equates to an air infiltration rate of 0.1-1.5 ACH, with an average of 0.7 ACH. Target air leakage rates for domestic properties are:

— 5-7 ACH at 50 Pa for dwellings having local extraction and background ventilation

— 4 ACH at 50 Pa for dwellings having whole house ventilation systems.

3.10.3 Strategies

The normal strategy is to extract directly at source from wet zones using mechanical extract ventilation (local or whole-house) or passive stacks. Fresh supply air is brought into the living rooms and bedrooms either by natural ventilation methods or as make-up, either induced by the negative pressure or via a mechanical whole-house ventilation system. Additional ventilation may be necessary if smoking takes place. However it should be noted that, as stated in section 3.2.1.4, the health risks of smoking cannot be completely eliminated by ventilation.

In high radon areas, sealing the foundations, combined with sub-floor venting, may be required. Specialist advice should be sought. Guidance is available from BRE[64].

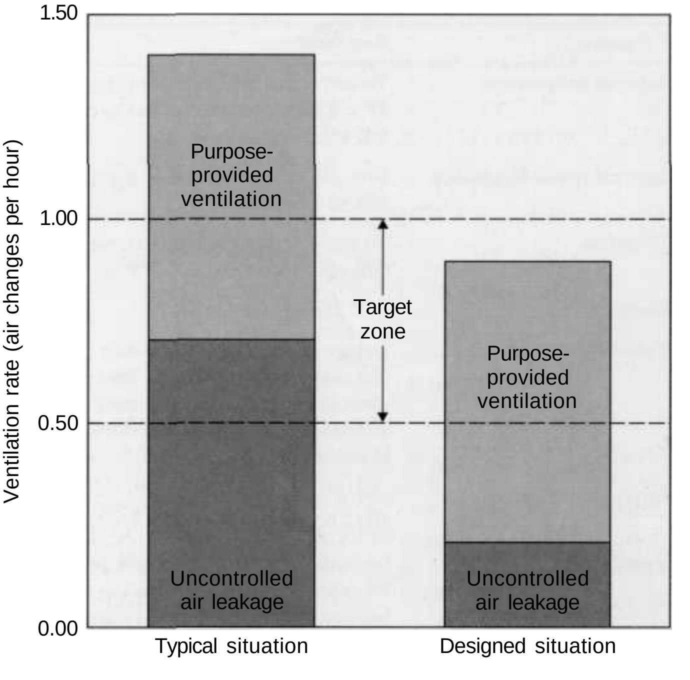


Figure 3.3 Impact of air leakage on ventilation rate

Balanced flue combustion appliances are preferable in dwellings fitted with any form of mechanical ventilation incorporating extraction, as their operation is not affected by pressure differences. Guidance on safety relating to combustion products is provided in BS 5440[25] and BS 5864[65] and Building Regulations Approved Document J[3].

3.10.3.1 Passive stack ventilation[66]

A passive stack system comprises vents located in the kitchen and bathroom connected via individual near-vertical circular or rectangular ducts to ridge or tile terminals. Moist air is drawn up through the ducts by a combination of stack and wind effects. The ducts, which are normally 80-125 mm in diameter[67], should have no more than two bends at greater than 30° to the vertical to minimise the resistance to air flow, and be insulated where they pass through cold spaces to reduce the risk of condensation. Replacement air enters via trickle or similar ventilators located in the 'dry' rooms and via air leakage.

Standard passive stack ventilation (PSV) systems have a simple inlet grille to the duct. Humidity sensitive vents are available that can provide increased flows when humidity is high. Acoustic treatment may be required to reduce ingress of external noise. Fire dampers are required where ducts pass through a fire-separating floor.

PSV systems can be combined with extract fans in hybrid systems, the fan being located in the kitchen.

Advantages

— No direct running costs.

— System will last the lifetime of the building.

— System is silent in operation.

— System requires no electrical connection.

Disadvantages

— Ventilation rate can be highly variable.

— Ventilation rate may be inadequate in poorly ventilated dwellings.

— Existing house layouts may make it difficult to accommodate duct runs.

— Site installation must be of good quality to avoid flow restrictions and excessive pressure drops.

— Uncontrolled systems waste energy due to continuous operation.

3.10.3.2 Local extract fans[60]

These are installed in kitchens and bathrooms to provide rapid extraction (typically 15-60 litre.s^{-1}) of moisture and other pollutants. They normally operate under occupant control or humidity control, or operate in association with door or light switches. Fans can be window, ceiling or wall mounted but are most effectively located at high level away from the source of fresh air, i.e. an internal door or trickle ventilator. In a kitchen they are ideally combined with a cooker hood. Ceiling mounted fans should be ducted to outside; however, it should be noted that duct-work lengths of as little as 1 m can considerably impair performance if an incorrect type of fan has been fitted[68]. Replacement air is provided by trickle ventilators or air leakage.

Fans should be located so as not to produce draughts and so as not to draw combustion products from open-flue appliances[3,65,66]. Note that cooker hoods require permanently open vents as close as possible to the hood. Control can be by manual switching or through being wired into door or light switches. Another option is humidity control with manual override, although the sensor may cause the fan to operate when moisture generation is not taking place, e.g. on warm humid summer days. The sensor needs to be positioned with consideration to where the major source of moisture is located. It may be more suitable to install cowled shutters to avoid noise problems with external gravity back-draught shutters rattling in the wind.

Advantages

— Simple and widely applicable.

— Provides the possibility of rapid extract.

— System is easily understood.

Disadvantages

— System is perceived by occupants to have high running costs and is prone to tampering by occupants.

— Noise can be an issue.

— System requires occasional maintenance.

3.10.3.3 Heat recovery room ventilators[63]

These are a development of the extract fan and are mounted in external walls. They incorporate a heat exchanger that recovers approximately 60% of the heat from the outgoing air. This is passed across to the incoming air to preheat it. The extract fan is often dual speed, providing low speed continuous trickle ventilation or high speed extract. High-speed extract can be under manual or humidity control.

Advantages

— Provides continuous low level ventilation.

— Provides the option of rapid extract.

— Recovers heat energy.

— Allows filtration of the supply air.

— Almost silent in operation at trickle speed.

Disadvantages

— Occupants perceive the systems to have high running costs.

— Regular maintenance is required.

— Some recirculation is possible, due to the close proximity of supply and extract grilles.

3.10.3.4 Mechanical supply ventilation[69]

A fan unit is typically mounted in the roof space and delivers air that has been filtered and tempered by the roofspace into the dwelling. The system works on the principle of continuous dilution, displacement and replacement of air in the dwelling. Air discharge from the dwelling is via purpose provided egress vents and/or leakage paths. Fans typically run continuously at low speed, with manual or humidity controlled boost to a higher speed when required. Temperature controls can incorporate single roofspace sensors or sensors in both the roof and living spaces. The latter system adjusts the flow rate of the unit to suit the temperatures in both spaces, thereby providing the optimum energy benefits for the occupants. Fan units incorporating highly efficient motor technology can provide a significant net energy gain to the dwelling.

Advantages

- Simple and well established as a means of controlling condensation.
- Compatible with open flued appliances.
- Utilises any heat gain in the loft space.
- Allows filtration of the air before it enters the space.

Disadvantages

- Occupants perceive the systems to have high running costs.
- Noise can be an issue.
- Systems are prone to tampering by occupants.
- Regular maintenance is required.
- Limited research has been carried out into system performance.
- Effectiveness depends upon the building shape and layout.

3.10.3.5 Continuous mechanical extract[69]

Continuous mechanical extract ventilation is a simpler alternative to a supply and extract system (see section 3.10.3.6). Further information on design, installation and operation is given in BRE Digest 398[70].

3.10.3.6 Whole-house mechanical ventilation[70]

A whole-house mechanical ventilation system normally combines supply and extract ventilation in one system. A heat exchanger can be incorporated to preheat the incoming air. These systems can be effective at meeting part of the heating load in energy efficient dwellings thereby helping to distribute the heat. Typically, warm moist air is extracted from kitchens, bathrooms, utility rooms and wcs via a system of ducting, and passed across a heat exchanger before being exhausted. Fresh incoming air is preheated and ducted to the living room and other habitable rooms.

Ducts may be circular or rectangular and range in size from 100 to 150 mm in diameter. Air velocities should be kept below 4 $m.s^{-1}$. Vertical exhaust ducts should be fitted with condensate traps, horizontal exhaust ducts should slope away from fans to prevent condensate running back. Both supply and extract grilles should be located at high level as far as practical from internal doors, but at a sufficient distance from each other to avoid 'short circuiting', i.e. a minimum of 2 m. Suitable louvres or cowls should be fitted to prevent ingress of rain, birds or insects.

Such systems can provide the ideal ventilation almost independent of weather conditions. During normal operation the total extract airflow rate will be 0.5-0.7 ACH based on the whole dwelling volume, less an allowance for background natural infiltration if desired. Individual room air change rates will be significantly higher, possibly 2-5 ACH, in rooms with an extract terminal. To be most effective a good standard of air tightness is required, typically better than 4 ACH at 50 Pa. Airflows need to be balanced at the time of installation. Extract rates from bathrooms and kitchens can be boosted during times of high moisture production although care should be taken not to cause draughts. The system can be acoustically treated to reduce noise ingress.

Transfer grilles are necessary only if the system is part of a warm air heating system but may be fitted in other cases, if desired. If the bottom edges of internal doors clear the floor surface by 5-8 mm there is likely to be sufficient opening for air movement. Transfer grilles are usually positioned not more than 450 mm above the floor. If placed higher they may allow the rapid movement of toxic combustion products or facilitate the spread of fire. Fire dampers should be inserted where the ductwork passes through separating walls and floors, and are desirable in kitchens, e.g. cooker hoods.

It is claimed that such systems are effective in reducing condensation due to the controlled ventilation and air-tight structure reducing cold air draughts. Manufacturers also claim that they improve indoor air quality and help in controlling dust mite populations.

Advantages

- Provides controlled preheated fresh air throughout the house.
- Reduces the heating demand in very airtight dwellings.
- Reduces the risk of condensation.

Disadvantages

- Ductwork can be difficult to accommodate.
- Initial costs are high.
- The systems has an ongoing maintenance liability: 6-monthly or annually.
- An adequate level of airtightness must be provided.
- Installation and commissioning is more complex than for other systems.

3.10.3.7 Comfort cooling and air conditioning

Systems are available which incorporate a heat pump into a whole-house mechanical ventilation system. Little information is available on their performance(71); similarly with other proposed systems of domestic comfort cooling or air conditioning(72). The decision to install such systems in domestic properties should not be taken lightly and designers should concentrate on enhancing the fabric performance to eliminate this need. If a comfort cooling and air conditioning system is proposed, key concerns for the occupants would be the ongoing maintenance requirements and acoustic considerations, both internal and external.

3.10.4 High rise dwellings

See section 3.12 for non-domestic high rise buildings.

High rise dwellings pose particular problems because of wind-induced pressures at the higher levels, i.e. above 6 storeys. This requires that special attention be paid to trickle ventilator selection(73-76). Whole-house mechanical ventilation systems, see section 3.10.3.6, are an option(77).

If every dwelling unit comprises a self-contained ventilation system, care must be taken to ensure that inlets to dwellings e.g. windows, trickle ventilators, or mechanical air intakes are not contaminated by ventilation outlets or combustion flue gases from adjacent dwellings. This may encourage the use of centrally ducted ventilation and heating systems(68), particularly in gas or oil heated properties.

The balancing of common toilet and bathroom ducts in high rise buildings is considered in section 3.12.

3.10.5 Further reading

Thermal insulation — avoiding risks BRE Report BR 262 (Garston: Building Research Establishment) (1994)

3.11 Factories and warehouses

This section considers the ventilation of industrial buildings and warehouses; see section 3.15 for ventilation of industrial processes.

3.11.1 General

Minimum ventilation rates are determined by the fresh air requirements for occupants laid down in the Factories Act(78) and Health and Safety at Work etc. Act(78) However these requirements are often exceeded by other criteria such as the ventilation requirements of the particular manufacturing processes.

There is no simple relationship between the building and process energy. The combination can be considered as:

- *Process incidental:* i.e. the process makes few demands on the internal environment. In many ways requirements are similar to office accommodation except that the space may be taller, the systems less sophisticated and environmental conditions often less demanding.
- *Process significant:* i.e. the servicing is dictated primarily by the comfort and performance requirements of the people in the building but affected by the needs of the process, e.g. humidification for textile weaving.
- *Process dominant:* i.e. the process demands very little of the building (e.g. it may be outside) or it may totally dominate the situation, for either quality or health and safety reasons.

Suitable systems will vary depending upon the degree of separation between accommodation types. Within a well-defined office area natural ventilation may suffice. Mechanical ventilation is required where occupancy is dense or where the opening of windows is not desirable. Within the production space, refer to section 3.15.

3.11.2 Requirements

3.11.2.1 Energy use

It is often difficult to distinguish between the energy consequences of the systems required for the industrial processes and those required for the buildings that contain them. However surveys of energy use commissioned under the government's Energy Efficiency Best Practice Programme (EEBPP) have shown that the worst and best performing buildings can differ by more than 100% within a particular industrial sector. For ease of comparison, EEBPP Energy Consumption Guide ECON 18(80) categorises industrial buildings as follows:

- *Storage and distribution buildings:* i.e. warehouses; these are typically 7.5 m high, contain pallet racking, and are naturally ventilated to 16 °C for single shift operation during the day, condensation protection being required at night. Refrigerated warehousing requires specialist treatment.
- *Light manufacturing buildings:* these are typically 5 m high and include areas for offices, storage and dispatch. They are largely naturally ventilated with occasional local mechanical extraction. Shift operation may be longer than for storage buildings only.
- *Factory/office buildings:* these are typically 4 m high, possibly with a suspended ceiling within the office areas, with little other differentiation between production, office and storage spaces. Some local mechanical ventilation or air conditioning may be present.
- *General manufacturing buildings:* these are typically 8 m high to accommodate tall equipment, gantry cranes and local storage racking. Mechanical ventilation may be provided to areas of high heat gain or for the clearance of process contaminants.

Table 3.14 provides energy targets relating to ventilation. However, these figures should be treated with caution, as the industrial building stock is extremely diverse; for example, high bay warehouses of 14 m height are not included in this classification. Further guidance is available on establishing building specific energy targets[81].

3.11.2.2 Air infiltration control

Air infiltration typically accounts for as much as 30% of the heat loss of an industrial building[81]. To minimise air infiltration problems consideration should be given to the following:

— structural integrity should be checked by infra-red thermography

— external windbreaks should be considered on exposed sites

— if a false ceiling has been installed to reduce ceiling heights in office areas, ensure that gaps have been sealed to prevent the leakage of warm air into the ceiling void

— goods doors should not be installed facing the prevailing wind or opposite each other; if this is not possible the goods loading area should:

  (a) be partitioned-off, either internally or externally, with the partitioning insulated to the same level as the external wall

  (b) have rapid closing doors suitable for frequent use, either push-button or automatic, or

  (c) have plastic strip curtains (although these are not a substitute for doors and there are safety considerations), or

  (d) have an air curtain, or

  (e) have a pneumatic seal around vehicle loading bays.

3.11.2.3 Heat recovery

See section 5.6 for details of heat recovery devices. Before considering heat recovery ensure that ventilation rates are minimised and can be adequately controlled. Where the extracted air is contaminated only with particles it may be possible to filter it and return it to the workplace. This eliminates heat losses but will result in more stringent maintenance requirements. If the recycled air is hot it may be discharged back into the workplace at low level during the winter; ductwork should also be provided to allow the hot air to be rejected to outside during the summer. The use of central plant will assist in the installation and economics of heat recovery but may prejudice its controllability.

3.11.2.4 Control

Plant can be controlled by time control or air flow rate control. Larger centralised systems should be zoned. Time control can be by means of:

— manual switching (should be easily accessible, with a well-labelled on/off switch)

— timeswitch

— push button or automatic presence detection allowing pre-set timed operation (useful for intermittently occupied areas)

— electrical interlock to associated production machinery (if local).

Airflow rate control can be achieved by:

— air temperature

— contaminant concentration

— number of machines in operation

— duct pressure (where zone isolation dampers are used on a centralised system).

Two-speed or variable speed motors should be considered. When contemplating reducing airflow rates, designers should be aware that limits may be in place to maintain a minimum duct velocity.

3.11.3 Strategies[81]

3.11.3.1 Natural ventilation

Subject to constraints imposed by industrial processes, natural ventilation can be particularly effective in industrial buildings due to the relatively high ceilings. The most effective ventilation will be obtained by using a combination of low and high level openings (e.g. windows and rooflights). With heat gains up to 20 $W.m^{-2}$, simple systems can be used that may be cheaper to install than those relying on mechanical plant. With heat gains of 20-40 $W.m^{-2}$, more sophisticated natural ventilation strategies may be required which may cost more to install than mechanical ventilation plant. However, life cycle costing could demonstrate the potential for overall savings due to reduced operational costs.

It may be possible to extend the applicability of natural ventilation by grouping process equipment into a few mechanically ventilated areas. For optimum energy efficiency, any natural ventilation should be controllable as natural air change rates in industrial buildings can be quite high (particularly if goods doors are left open). The correct strategy is to design the building to be as airtight

Table 3.14 Building related energy use[80]

Classification	Electricity consumption for fans, pumps, controls / $kWh.m^{-2}$ per year	Total electricity consumption of building / $kW.h.m^{-2}$ per year
Storage and distribution	5	50
Light manufacturing	6	55
Factory-office	10	31
General manufacturing	10	20

as possible and to provide the required amount of ventilation by controllable means. If space is to be subsequently partitioned off for the creation of office accommodation ensure that this will not affect the operation of the ventilation system.

3.11.3.2 Mechanical ventilation

For general factory ventilation consider the use of high level extract fans (either wall or roof mounted). These are effective at removing heat but are ineffective at controlling fumes, see sections 3.2.1 and 3.15. Consider providing all mechanical ventilation systems with back-draught shutters or dampers to prevent air infiltration when the fans are not in use.

Prevent excessive fan power requirements by ensuring that all ductwork is appropriately sized, i.e. pressure drops not more than 1 $Pa.m^{-1}$. This usually equates to an air velocity of about 10 $m.s^{-1}$ in main ducts and 4 $m.s^{-1}$ in branch ducts. Over-sized fans should not be used as they will operate at sub-optimal efficiency and/or may require throttling in order to provide the suction or airflow rates required. Make-up air should be introduced to minimise energy use and discomfort, and to ensure the continued safety of heating appliances.

3.11.3.3 Make-up air

Make-up systems should be specified to provide the optimum building pressure balance. The choice of pressure balance will depend upon the processes taking place within the building, see sections 4.3 and 3.15. Negative pressures may upset heating appliances with traditional flues. Positive pressure may facilitate uniform heating and help prevent the ingress of untreated external air. Direct gas firing is a particularly efficient way of tempering large volumes of fresh air if required as make-up.

3.11.4 Further considerations

3.11.4.1 Automatic doors

These are probably the most energy efficient solution for low traffic situations where it is inconvenient or impracticable to open doors manually. However, they become effectively permanently open doorways when traffic is dense.

3.11.4.2 Air curtains[82,83]

Air curtains condition the incoming air at the entrance in order to minimise cold draughts. They do not act as a physical barrier to prevent the entry of outside air but use heating energy to temper air that enters the doorway. They prevent the natural convection of warm air out of the top of a doorway being replaced by cold air at the bottom.

The heat input of an air curtain must be sufficient to temper the quantity of air coming in at the entrance. An air curtain will not be effective if the velocity of the incoming air is excessive. This can occur as a result of under-pressure within the building from extract systems, stack effect with leaky or tall buildings, or wind effects on an exposed site.

The width of an air curtain discharge grille should be just wider than the doorway opening; an air curtain narrower than the doorway is ineffective. Opening and closing of doors can disrupt the air stream, which takes some time to re-establish. The heating capacity of an air curtain can have an effect on the space temperature within the building entrance and suitable controls need to be fitted to adjust the heat output and air stream characteristics if necessary.

3.11.5 Further reading

An environmental assessment for new industrial, warehousing and non-food retail units: New Industrial units BREEAM 5/93 (Garston: Building Research Establishment) (1994)

3.12 High rise buildings (non-domestic)

This section relates to non-domestic high rise buildings. Domestic high rise buildings are covered in section 3.10.

3.12.1 General

Whilst the aims of the ventilation strategy for high rise buildings (i.e. greater than 20 storeys) do not necessarily differ from those of other buildings, there are specific design issues that need to be taken into consideration when selecting and designing ventilating and air conditioning systems. In particular these include stack effects, high winds and hydraulic pressures.

3.12.2 Stack effect, high winds and hydraulic pressures

Stack effects created by buoyancy pressures are magnified by the height of the building. In cold climates the interior air will usually be warmer than the outside air. Buoyancy forces cause warm air to leak out of the upper part of the building and cold ambient air to leak in at the base of the building. This will have a number of effects including:

— requiring energy to heat infiltrated air
— driving moisture into the envelope assembly, allowing condensation to form and deteriorate the materials and insulation
— creating uncomfortable draughts and possibly annoying whistling noises
— pressure differences between floor space and shafts affecting opening and closing of doors.

In warm climates a negative stack effect occurs with cold air flowing out of the base of the building and infiltration of warm moist air at the top. Moisture condensing in the cool interior environment can cause serious damage to the building materials. Envelope tightness is not usually as carefully controlled in warm climates because leakage is not as apparent; however, the potential damage is greater than that occurring in cold climate.

Features that help combat infiltration due to the stack effect[84,85] and wind pressures include the following:

- revolving doors or vestibules at exterior entrances
- pressurised lobbies
- tight gaskets on stairwell doors leading to the roof
- automatic dampers on elevator shaft vents
- airtight separations in vertical shafts
- tight construction of the exterior skin
- tight closure and seals on all dampers opening to the exterior.

The large stack effect and high winds normally mean that natural ventilation is impracticable and therefore high rise buildings are invariably mechanically ventilated or air conditioned. One possible means of reducing the stack effect is to divide the building into small self-contained units.

Air flows in extract ducts connected to vertical duct shafts in buildings can be unbalanced by stack forces, causing increased flow in some ducts and reduced, or possibly reversed, flow in others[85]. Flow reversal is particularly undesirable on toilet extracts and waste disposal chutes.

A further consideration for system design in high rise buildings is hydraulic system head pressures. Cost, safety and technical limitations relating to maximum head pressure dictate that hydraulic systems are normally split into vertical blocks of 20-25 storeys. There are a number of alternative design solutions for achieving pressure isolation including pressure separating heat exchangers, cascading water upwards to storage tanks, and installing separate systems for vertical zones within the height of the building (though this last solution is complex and costly). For condenser water-type systems an intermediate sump pump could be considered. This should be located as high as possible subject to economic pressure rating. Column pressure is lost above the sump, but retained below providing partial recovery of pump energy.

3.12.3 System considerations

Centralised, floor-by-floor and unitary systems are all potentially suitable for high rise buildings. For centralised systems, the number of floors that can be served is limited to 10 floors above or below (20 floors for an intermediate plant room serving floors both above and below). This is the maximum number of duct take-offs that can readily be balanced. (Note that the static regain method should be considered for ductwork sizing to assist with balancing).

There are a number of issues that will impact on the choice between a centralised, floor-by-floor, or unitary approach including the following:

- tenancy requirements
- floor plate size
- riser and/or plant room space requirements
- maintenance considerations: centralised systems will be subject to large scale disruption due to localised problems or retrofit; unitary systems can require hundreds of units with the attendant management and maintenance difficulties.

3.13 Hospitals and health care buildings

3.13.1 General

The heating and cooling load associated with ventilation plant form the major component of boiler and chiller plant capacity. It is therefore important to determine the ventilation strategy at an early stage of design to ensure that the systems are tailored to the requirements of each area. In practice this means that areas with specific requirements have dedicated air handling systems, and that departments occupied only during office hours are served by plant separate from that serving continuously occupied areas.

In general, separate ventilation systems should be provided for each department or group of similar departments provided that they are closely grouped together.

Each operating theatre suite should ideally be provided with its own plant but it is accepted practice to have a zoned common air handling unit serving two adjacent suites. There are many examples where common air handling plant has been provided for an entire operating department which, in the event of plant failure or maintenance shut-down, will render the whole department inoperative. Also, it means that it would be uneconomic to operate a single theatre for emergency or maternity use out of normal hours.

For health care buildings within the UK, it should not be assumed that the entire building needs to be closely temperature controlled. Ward areas (with the exception of isolation rooms and other special rooms) should be designed for natural ventilation unless situated in a noisy or heavily polluted location. Ancillary areas such as toilets, bathrooms, utility rooms, etc. should be provided with an extract system. It is a general requirement for health care buildings that the building has an overall positive or neutral pressure and the extracted air replaced by treated make-up air supplied to, for example, internal areas, staff base, etc. in ward areas.

3.13.2 Cleanliness and infection control

Ventilation systems should be of the all-fresh air type to minimise risk of infection. In areas such as non-invasive imaging, equipment rooms and staff areas, local recirculatory air systems in the form of fan coil or split air conditioning units may be used, supplemented by primary air.

Air handling plant for all medical areas should be of the 'blow-through' type with only the frost coil and pre-filter upstream of the fan to ensure that there is no inward leakage of air downstream of the coils and main filter.

Ventilation systems should be fully ducted. In the event of contamination only the affected rooms and associated ductwork require cleansing. With a return air ceiling plenum, access to the void above the room would be necessary for cleaning.

3.13.3 Ductwork and distribution

Ductwork systems should be low velocity designs to minimise fan power energy and noise. Attention should be given to eliminate cross-talk in areas where confidentiality is necessary or where patients may be noisy.

Ductwork systems should be cleaned on completion and provided with sufficient access points to ensure that adequate cleaning can be undertaken.

Air terminals should be selected with ease of cleaning as a primary consideration. Internal acoustic linings should be avoided. Room-side supply air attenuators as a minimum should be suitably lined to prevent fibre migration and to facilitate cleaning.

3.13.4 Ventilation system design

There are many mechanically ventilated spaces that do not require close control of temperature and where a summer upper limit of 25 °C will be acceptable. Ventilation systems should be designed with a small temperature difference between supply air temperature and room design temperature to achieve acceptable variation in room temperature for the majority of spaces, without the need for local temperature control.

As a general principle, space heating should be provided independently and not rely on adding heat to the ventilation supply air. However, in theatre suites and high dependency areas such as intensive care, heating requirements would normally be met by the ventilation system.

Most ventilation systems are constant volume type to satisfy pressure regimes or to offset fixed extraction rates. Variable air volume (VAV) systems may be appropriate for areas where cooling loads are variable. They will also be more energy efficient in these situations than constant volume systems.

Mechanical ventilation systems for hospitals and health care buildings need to be designed to meet the sound control requirements of CIBSE Guide B5: *Sound control*[39]

There is often a high proportion of rooms requiring full height partitions for fire compartmentation and acoustic separation and this requires that VAV systems have devices to balance both supply and extract to each area. This means that VAV systems are costly.

An economic case can be made for heat recovery on continuously operating ventilation systems. To avoid the risk of cross infection, air/water heat recovery systems are preferred and air/air systems would be subject to agreement with the infection control officer and would normally exclude dirty extracts.

In hospitals, the patients are dependent to varying degrees on the staff for evacuation in the event of fire. This, combined with various fire risk rooms, results in a higher than normal requirement for sub-compartments and compartmentation of risk rooms. It is therefore important to minimise the number of fire- and smoke-operated dampers by appropriate routing of ducts when compartmentation requirements are determined.

In many departments in hospitals, especially in operating departments and high dependency areas, the ventilation will need to remain operational in the event of fire when other areas would be under firefighters' control. In these circumstances, the ventilation system should shut down only in the event that smoke is detected in the supply air.

Mechanical ventilation systems for hospitals need to provide a high level of reliability, as the system is critical to the proper functioning of the building and the business conducted within it. Consequential losses arising from failure can be very significant in this type of building.

It should be noted that the external design conditions for health care buildings are more onerous than for other building types and summer/winter values are based on those not exceeded for more than 10 hours per year.

For specific ventilation requirements reference should be made to appropriate NHS Health Building Notes and Health Technical Memoranda, with particular reference to HTM 2025[86].

Ventilation rates for typical spaces are given in Table 3.15.

3.13.5 Humidification

It should not be assumed that humidification is required in all areas. The avoidance of infection and, in particular, *Legionellae* is of paramount importance, especially as many patients will have limited resistance. The recommended method of humidifying the supply air is by steam injection from plant steam (clean steam is not required). Electrical generation of steam is low in initial cost but high in running cost and should be avoided. Alternative methods of humidification would normally be subject to agreement with the infection control officer.

Table 3.15 Hospitals and health care buildings: ventilation rates

Space	Ventilation rate (air changes per hour)
Toilets:	
— general	10
— en suite	6
Bathrooms:	
— general	10
— en suite	6
Dirty utility room	10
Changing rooms	5
Isolation rooms	10 (minimum)
Delivery rooms	10 (minimum)
Recovery rooms	15
Treatment rooms	6 (minimum to offset heat gain)

3.13.6 Filtration requirements

Various levels of filtration performance are required, see Table 3.16.

Table 3.16 Filtration requirements

Application	Filter class*
Pre-filters on air handling plant, protection to heat recovery source coils	G3
Final filter for general spaces	F6
Final filter for clinical spaces; protection to HEPA filters	F8
Aseptic suite; sterile services department; operating theatre ultra-clean units	H10-H14

* See Tables 5.10 and 5.11 for details of filter classes

3.13.7 Specialist areas

Certain areas have ventilation requirements that cannot be achieved by normal methods. These include audiology rooms where extremely low background noise levels must be achieved and aseptic suites where low particle counts are necessary. In these instances it is recommended that specialist contractors take responsibility for both the building enclosure and the building engineering services, including ventilation, within the enclosure.

3.13.8 Further reading

Electricity savings in hospitals — a guide for energy and estate managers EEBPP Good Practice Guide GPG 054 (London: Department of Transport, Local Government and the Regions) (1993) (available from http://www.energy-efficiency.gov.uk)

3.14 Hotels

3.14.1 General

Hotels present a number of design challenges. Running costs are usually of high importance to the operator but the control of these should not affect guest comfort levels. Obtaining energy cheaply and using it efficiently are both areas that should be reviewed. Maintenance also needs to be carefully considered as many hotels have limited on-site technical support.

Guests directly paying for a service are reluctant to accept compromises in temperature, service or the quality of the environment that would allow the hotel to reduce its energy consumption. Therefore it is important to avoid waste. To achieve this, systems need to be responsive and readily controllable. Means to turn off, or turn down, systems when they are not required should be provided, but must be straightforward and easily managed by non-technical staff.

The level of service will depending on the type of hotel. Understanding the type and the branding of the hotel is important to choosing the right system. In the UK, standard solutions range from electric heating with natural ventilation to full air conditioning. Many hotel operators will have well-developed standard solutions.

Different types of hotels will also have different occupancy rates and this can have a major impact on sizing of central plant and public space systems. A business hotel will have a full occupancy at between 1.1 and 1.3 persons/room whereas a family or resort hotel will have a much higher occupancy, typically up to at least 2.0 persons/room. A rate of 2.4 persons/room may not be unreasonable for a busy budget hotel near an airport.

3.14.2 Design considerations and strategies

There are three principal areas within a hotel: guest bedrooms (including en-suite bathrooms), public areas and 'back of house' areas. Each of these is serviced in a different manner and requires different operating schedules. The diversities applied to central plant can therefore be quite high and the likely peak loads need to be carefully considered. A spreadsheet showing the combined load at hourly intervals is an effective way of reviewing how the different loads interact and can be used in design discussions with the client. It can also be used as the first step in analysing the potential for a combined heat and power (CHP) approach. Hotels, particularly those with swimming pools, are usually good candidates for CHP.

3.14.2.1 Guest bedrooms

For an air conditioned hotel, a common approach is to employ a four-pipe fan coil unit located above the entrance lobby but it is also possible to locate the unit against the perimeter wall or above the bathroom, provided that adequate access is available for maintenance. The unit should be sized to allow a rapid and individual response to each room. Other common solutions are water source heat pumps and variable refrigerant flow (VRV) systems. These have the advantage of using less riser space but may require more maintenance. Care must also be taken with refrigerant systems to ensure that the effects of a refrigerant gas leak can be dealt with safely[87].

Environmental control needs to be clear and responsive. Controls should be simple to understand and to operate. Acceptable noise levels can also be an issue and need to be agreed with the client. A reasonable standard is to design for an overnight condition of NR30 on low speed and allow higher noise levels to meet the design load. Luxury hotels may require an overnight level of NR25.

Care should be taken not to oversize the selected system while ensuring that the system remains responsive. The peak solar load is unlikely to coincide with the peak internal loads. Depending on occupancy, a peak room cooling load between 1.5 and 2.0 kW will normally be adequate for the UK. For a well constructed and insulated building, the heat gains to a typical bedroom when occupied will offset the heat losses. Therefore heating costs can be low and a design can be developed that will provide the most effective and controllable means of meeting this intermittent low load. Some hotels have adopted electric heating because of the ease of control and the saving on installed cost.

Some hotels choose to limit the energy consumption of the bedroom systems by the use of occupancy detectors, key fobs or central booking systems. These can be used to turn off electrical systems and turn down the air conditioning when the guests are not in the room. This has been shown to make significant energy savings but care needs to be taken to ensure that guest comfort levels are not affected and that critical loads such as the 'minibar' (if present) are not isolated.

For compliance with building regulations, the minimum extract rate for a bathroom is 15 $litre.s^{-1}$ but many hotels use higher values such as 25 $litre.s^{-1}$. This will provide 10 to 15 air changes per hour in the bathroom and balance the supply of fresh air for two occupants in the room. At these higher rates, tempered air is usually supplied directly to the room or to the fan coil unit within the room to avoid large gaps under doors or external air grilles. Some hotels choose even higher values to minimise condensation in bathrooms and improve air quality generally, particularly if smoking is allowed.

The supply location needs to be positioned to reduce the likelihood of draughts over the bed and in areas that may be used by the occupants when walking to and from the bathroom. The fresh air supply should be designed to take account of the fact that many UK guests will turn off the air-conditioning before going to sleep and this should not, ideally, limit the incoming fresh air. It is common to keep the bedroom supply and bathroom extract systems running continuously to maintain room air quality and to ensure adequate extract from the bathroom at unusual hours. Therefore heat recovery should be considered for these systems

3.14.2.2 Public areas

Public areas such as reception, conference, bar and restaurant areas are generally characterised by high, but variable, occupancy levels and lighting loads. The chosen system will need to be responsive and capable of delivering high quantities of fresh air when required to do so. This will often suggest all-air systems but these need to be carefully zoned to allow individual control of spaces. Where possible, separate systems for the different areas are ideal, but multi-zone systems are also used and these are sometimes supplemented with fan coil units to provide more individual control. Constant volume systems with reheat are occasionally used but can be wasteful of energy. VAV systems are also used but should be treated with care to ensure that adequate fresh air is delivered to the space under all conditions.

The design of the systems for the public areas will need to achieve criteria imposed by licensing regulations. The level of occupancy to which the hotel wishes to be licensed should be agreed with the client at an early stage to ensure that the air systems will be capable of delivering the correct fresh air quantities to meet the requirements of the licensing authority. Typical design occupancies range from 1 person per 1.2 m^2 for 'theatre' style conference rooms, to 1 person per 2 m^2 for bars and restaurants and 1 person per 4 m^2 for reception and entrance areas. These figures should be confirmed at an early stage as the operator may wish to have the hotel licensed for higher densities. The fresh air quantities should allow for some smoking but not necessarily at the peak occupancies quoted above.

3.14.2.3 'Back of house' areas

The 'staff only' (or 'back of house') areas will require a variety of systems to suit their different uses. Typically, these areas will include managers offices, kitchens, laundries or linen handling, staff changing, staff dining, training, IT and computer rooms. Reference should be made to the guidance given for kitchens (section 3.6.1) and computer rooms (section 3.9).

The general office areas will normally be treated to the same level as the public spaces, (i.e. for an air conditioned hotel they will be air conditioned). Some hotels believe in extending this to cover further areas, such as the staff dining rooms and this needs to be clarified with the client as early as possible. It is common for the kitchens to be cooled, at least in part, so that salads, pastries and deserts can be well presented and that the general cooking area does not become too uncomfortable.

Many hotels choose to contract-out their laundry requirements, but linen handling space will still be needed. These areas require high air change rates to remove the high levels of dust and lint that will be generated during the sorting and stacking of linen. A design figure is 15 air changes per hour may be considered as reasonable. Linen chutes will also generate high dust levels in the collection room.

Increasingly, hotels have sophisticated IT and billing systems and therefore the computer room housing the central IT equipment is critical to the operation of the hotel and must be properly conditioned.

3.14.3 Further reading

Energy efficiency in hotels — a guide for owners and managers EEBPP Energy Consumption Guide ECG 036 (London: Department of Transport, Local Government and the Regions) (1999) (available from http://www.energy-efficiency.gov.uk)

3.15 Industrial ventilation

3.15.1 General requirements

In an industrial context, ventilation is usually employed to remove airborne contaminants arising from processes or machines. Satisfactory ambient conditions can be achieved by dilution where contaminant sources are weak, of low toxicity, and are either scattered or mobile. However, it is usually more appropriate to remove the contaminant at, or close to, its source by means of local exhaust, e.g. vehicle exhaust removal systems in garages.

Sources of industrial contaminants often require large extract airflow rates to ensure that the released pollutant is effectively captured and conveyed away by the extract system. In such cases, particular attention should be paid to ensuring adequate replacement or make-up air. It may be necessary to directly heat the incoming air in winter or, in order to reduce the resulting high energy consumption, to duct the outdoor air directly to the source location.

For certain processes, such as paint spraying, filtration of the incoming air may be necessary. Similarly it may be necessary to remove the contamination from the exhaust air before it is discharged to outside. Special industrial air cleaning devices are available for this purpose, see section 5.4.

The basic factors that affect the choice between natural and mechanical ventilation are:

— quantity of air required
— quality of air required
— consistency of control required
— isolation required from external environment.

It is almost certain that mechanical ventilation will be necessary given the likelihood of high airflow rates and the need to treat the incoming air, i.e. by heating, cooling, or filtration. Mechanical ventilation systems can be designed to provide constant or variable flow rates distributed as required throughout the building. When a building is located in a noisy environment, it is often impracticable to provide adequate natural ventilation without excessive sound transmission through the openings. In such circumstances, mechanical ventilation systems with appropriate acoustic treatment can be used. Mechanical ventilation can also be designed to control room pressures to prevent the ingress or egress of contaminants.

Ideally, industrial ventilation systems should limit the exposure of workers to airborne contaminants to zero, or as near zero as is practicable. As a minimum, limits should be maintained below the most recently published occupational health limits(10). These are updated annually and it is essential that current information be used.

If extract rates are too low, short term or long term damage to health will occur or, at the very least, serious discomfort will be experienced. If too much air is handled, fan and ductwork costs (both capital and running) are excessive, incoming air treatment costs are high, draughts may be difficult and expensive to prevent, and the industrial processes may be affected by overcooling or costly increases in chemical evaporation rates.

The most effective method of preventing a contaminant from entering the breathing zone of a worker is to isolate the process by total enclosure. This solution is essential where highly toxic substances are involved and may be appropriate for automated processes. Normally some degree of access to the process will be required. It is desirable to limit this access to the minimum necessary for a particular process e.g. access to a low emission chemical process within a fume cupboard via a sliding door, to components to be welded together, or to surfaces to be spray painted. In all cases the contaminant must be drawn away from the breathing zone of the worker.

Guidance on achieving energy efficient ventilation design within industrial buildings is available from BRECSU* and, for industrial processes, by ETSU† under the government's Energy Efficiency Best Practice programme.

3.15.2 Exhaust hood suction dynamics

The velocity of the air induced by suction at an exhaust hood decreases rapidly with distance from the opening. In theory, the velocity at a given distance from an opening can be predicted from an equation of the form:

$$V_x = \frac{Q}{B x^n + A} \qquad (3.4)$$

where V_x is the air velocity at distance x from the opening (m.s^{-1}), Q is the volume flow rate of air (m^3.s^{-1}), x is the distance from the opening (m) and A, B and n are constants depending on the geometry of the opening and the flow characteristics. Values for these constants are usually obtained experimentally.

Figure 3.4 shows solutions of this equation for circular openings having unflanged and flanged edges. Note the improvement in performance when the suction is focussed by the flange. The efficiency of capture can be further improved by side screens which also reduce the influence of cross draughts. The ultimate extension of this principle is to enclose the process completely. Velocities are given as percentages of velocity at the opening V_o. Distances from opening are given as percentages of the diameter, d.

Solutions to equation 3.4 for various types of openings are given section 5.4.

The momentum of the air induced by suction at an opening must be sufficient at the part of the process most

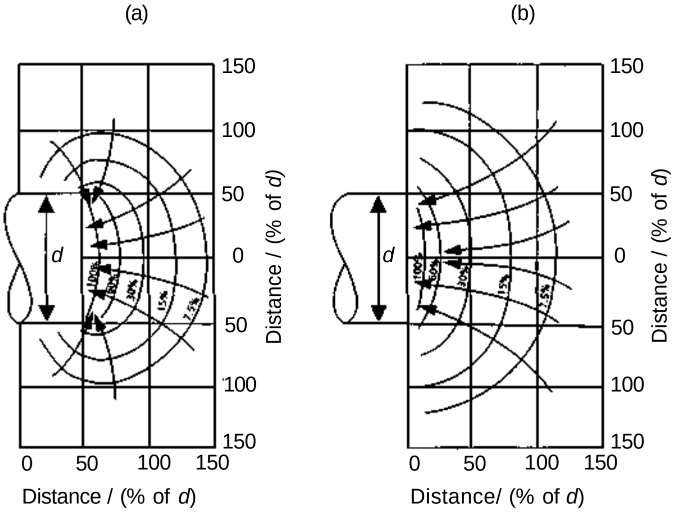


Figure 3.4 Isovels for circular openings; (a) sharp edged opening, (b) flanged opening

*Building Research Energy Conservation Support Unit (BRECSU), Garston, Watford WD2 7JR, UK (http://www.bre.co.uk/brecsu/index.html)

† Energy Technology Support Unit (ETSU), Building 156, AEA Technology plc, Harwell, Didcot, Oxfordshire, 0X11 ORA, UK (http://www.etsu.com/eebpp/home.htm)

remote from the opening to overcome a combination of the following forces:

- *gravitation:* due to the density of the air/contaminant mixture in relation to the surrounding air
- *friction:* to overcome drag on the mixture due to the neighbouring bulk of room air
- *dynamic:* due to the initial momentum of the contaminant on release from source and/or disturbing forces due to movement of room air, e.g. cross draughts.

Gravitational and dynamic forces may be used to assist capture. Heavy dust particles having some momentum should be directed into an opening close to the source and, ideally, should be collected and removed from the exhaust without further transport. Transporting large particles through a duct requires very high velocities.

If emitted into a workspace with low momentum, the concentration of contaminant immediately adjacent to its source will be high but normally complete mixing with workspace air will occur within a short distance from the source. An obstructed bayonet plume from a hot source will entrain and mix with room air thus expanding the plume, but if an opening can be used to contain the plume, induction may prove sufficient to avoid the need for additional fan-induced forces.

3.15.3 System design

Individual exhaust hoods can be either discharged separately to outside via individual fans, or connected via a multi-branch system to central fan(s) depending upon:

- compatibility of substances evolved by different processes; if in doubt, separate exhausts should be used
- access to the outside wall: multiple roof penetration might not be acceptable
- aesthetics of multiple discharges
- potential for air cleaning and recirculation of heat recovery from exhausts (see section 5.4)
- balancing: multi-branch dust handling systems must be self-balancing, obstructions within duct work could create blockages
- process usage pattern: ventilation may need to be isolated when a process is not in use and operation of an isolating damper may upset system balance unless a variable volume fan is used; (the VAV fan would be controlled from a system pressure sensor, which could become blocked if dust is transported within the duct).

If make-up air requirements are small they can be drawn from outside or surrounding areas via cracks or openings in the fabric. However, negative pressure must not be allowed to develop at a level at which swing doors are held open or cold draughts are produced in occupied spaces near doors or windows etc. Careful positioning of perimeter heating will minimise discomfort by warming the incoming air. If make-up is too low, the performance of one hood may be affected by the operation of other hoods.

It is preferable to supply the make-up air via a handling system, which cleans, heats (in winter) and, exceptionally, cools and dehumidifies the air, as appropriate. Large volumes of make-up air may be required. This has considerable implications for energy consumption, therefore consideration must be given to:

- supply of tempered make-up air direct to the process (e.g. by push-pull system)
- partial recirculation of exhaust air after removal of contaminant using high efficiency air cleaning[88], see section 5.7
- recovery of heat from exhaust to incoming make-up air but avoiding transfer of contaminants, see section 5.6.

Make-up air must be supplied into the space in such a way as to avoid causing draughts across the process, which would affect the efficiency of capture.

3.16 Laboratories

3.16.1 General[89]

The design of laboratory projects will generally be biased towards the design of the ventilation systems for (a) fume control, *(b)* containment, or (c) providing specific close environmental conditions for either animal welfare or research processes.

The choice of protection to be provided will need to be identified by the client or end user as part of their safety assessment of the work that is being undertaken. Operator protection may be provided by fume cupboards, micro-biological safety cabinets or other local exhaust ventilation systems.

The design of laboratories will need to take many factors account, including the following:

- number of fume cupboards, their performance criteria and diversity of use
- number of microbiological safety cabinets
- local exhaust ventilation systems
- minimum ventilation rates to dilute odours and contaminants
- pressure differentials or air flow direction with respect to adjacent spaces
- temperature criteria and heat gains
- filtration standards
- standby capacity
- plant space
- fume discharges to atmosphere
- ductwork materials
- running costs.

Mechanical ventilation systems for laboratories need to provide a high level of reliability, as the system is critical to the proper functioning of the building and the business conducted within it. Consequential losses arising from failure can be very significant in this type of building.

Information on the design of laboratories is available within the series of *Baseline Guides* produced by the International Society for Pharmaceutical Engineering*.

3.16.2 Design requirements and strategies

3.16.2.1 Fume cupboard installations

The performance criteria for the fume cupboard will need to be established by the end user and will be a function of face velocity and containment factors. Generally good containment can be achieved at face velocities of 0.5 $m.s^{-1}$ and may still be achieved at lower face velocities depending on the design of the fume cupboard. The face velocity and containment factor are normally specified in accordance with a sash working height of 500 mm. The specification of lower face velocities should be in conjunction with suitable type testing conditions and agreement to containment levels necessary to suit the end user's activities. Higher face velocities may be required for radioactive work but velocities exceeding 0.7 $m.s^{-1}$ can create turbulence around the operator that may affect the containment performance of the fume cupboard.

A minimum air change rate in a mechanically ventilated laboratory may be set between 6 and 15 air changes per hour, depending on the type of work that is being undertaken and the need to remove or dilute odours. Where fume cupboards are installed the face velocity may dictate the amount of air to be extracted and supplied, and this may exceed minimum ventilation requirements.

Where single or a small number of fume cupboards are installed, then constant volume 'face and bypass' fume cupboards may be considered with the fume cupboard acting as the return air path for the room.

Where large numbers of fume cupboards are to be installed then variable volume ventilation systems should be considered. Such systems enable a diversity in use to be applied and hence the size and cost of central plant can be reduced compared with that required for constant volume systems. In addition to the energy savings realised, the increased capital cost of the controls can be offset by the reduced costs of central plant and reduced plant room space requirements. The primary energy saving is achieved by the ability to deliver and extract reduced quantities of air. Central plant diversities of 50-70% can be applied to large installations. The diversity should take into account the number of fume cupboards in the laboratory, the number of users, and the type of work being undertaken. It may be appropriate to undertake studies to this effect, which may lead to lower diversities being applied.

*International Society for Pharmaceutical Engineering, 3816W Linebaugh Avenue, Suite 412, Tampa, Florida 33624, USA (http ://www.ispe.org)

Central extract systems will need to take account of the requirements for discharge of fumes via flue stacks. To achieve suitable dispersal of fumes the discharge velocity should generally not be less than 15 $m.s^{-1}$. With variable volume systems consideration should be given to providing automatic make-up air controls to collector ducts, in order to maintain discharge velocities. Flue stack heights may be in accordance with BS 7258[89] or alternatively can be determined by wind tunnel testing or dilution and dispersal calculations.

The use of individual extract fans may be appropriate if the fume cupboards are dispersed around the building in a way that would preclude the installation of a common collector duct.

3.16.2.2 Microbiological laboratories[90]

The design of laboratories where work on biological agents is undertaken requires the following particular factors to be taken into account:

- containment category
- number, size and class of microbiological safety cabinets
- operational requirements of the laboratory
- standby plant
- pressure differentials
- location and safe change requirement for HEPA filtration
- fumigation and sterilisation procedures
- safe access for maintenance of filters and other areas of potential contaminant concentration.

Guidance from the Advisory Committee on Dangerous Pathogens[91] defines hazard groups and provides recommendations for containment levels for laboratories and animal rooms along with appendices providing useful information and recommendations. Table 3.17 summarises the requirements and recommendations for laboratory containment.

The containment levels are as follows:

- *Containment Level 1*: suitable for work with agents in hazard group 1, which are unlikely to cause disease by infection, some agents in this group are nevertheless hazardous in other ways, i.e. allergenic, toxigenic etc. It is preferable to maintain an inward air flow by extracting room air to atmosphere.
- *Containment Level 2:* suitable for work with biological agents in hazard group 2. Restricted access required. Maintain at a negative air pressure while work is in progress. Doors should be closed whilst work is in progress.
- *Containment Level 3:* suitable for work with biological agents in hazard group 3. The laboratory to be separated from other activities in the same building with access restricted to authorised persons. The laboratory is to be maintained at a negative air pressure generally only when work with biological agents is in progress, although some clients may require pressure differentials to

be maintained continuously. Extract must be HEPA filtered. The laboratory is to be sealed for disinfection, which may require gas-tight shut-off dampers on ductwork systems and sealed fittings and services penetrations. Ventilation systems should also incorporate a means of preventing reverse air flows. Design of systems to achieve the required inward airflow should aim for simplicity.

— *Containment Level 4:* suitable for work with biological agents in hazard group 4. Maintain at a negative air pressure. Input air to be HEPA filtered, extract air to be double HEPA filtered.

Hazardous work within the laboratory will generally be undertaken in microbiological safety cabinets. Safety cabinets provide protection against dangerous pathogens. There are three classes of safety cabinets:

— *Class 1 safely cabinets:* provide user protection. The cabinet has a through flow of air and incorporates an integral HEPA filter. A variable speed fan is provided in the extract ductwork to overcome the changing resistance of the filter. Suitable for use with hazard groups 1, 2 and 3.

— *Class 2 safety cabinets:* protect the operator and the work by recirculating some of the air through a HEPA filter to provide a down-flow over the working area. An integral variable speed fan is provided to overcome the changing resistance of the filters. The main extract fan in the exhaust duct may require to be either variable or constant volume, depending on the manufacturer. Class 2 safety cabinets are divided into two types: high protection, for use with groups 1, 2, and 3; low protection for use with hazard groups 1 and 2.

— *Class 3 safety cabinets:* totally enclosed units designed to provide a high degree of user protection. Air is drawn in and exhausted via HEPA filters. The operator uses gloves to manipulate experiments. Suitable for hazard groups 1 to 4.

3.16.3 Further reading

Croner's Laboratory Manager (Kingston upon Thames: Croner)

Laboratory design issues BOHS Technical Guide No 10 (Derby: British Occupational Hygiene Society) (1992)

Ventilation in healthcare premises: design considerations NHS Estates HTM 2025 (London: The Stationery Office) (1994)

Accommodation for pathology services NHS Estates Health Building Note 15 (London: The Stationery Office) (1991)

3.17 Museums, libraries and art galleries

3.17.1 General

Most buildings control their environment for human health and comfort reasons during periods of occupation. However, buildings used for the display or storage of objects, books and documents requiring long-term preservation must be kept within appropriate relative

Table 3.17 Summary of laboratory containment requirements and recommendations

Measure	Requirement for stated hazard level			
	None	Low	Medium	High
ACDP containment level	1	2	3	4
Isolate from other areas	No	No	Yes/partial	Yes
Air lock	No	No	Optional (self closing)	Yes, via air lock and interlocking outer and inner doors; provide shower
Sealable for decontamination	No	No	Yes	Yes
Inward airflow/negative pressure	Optional	No, unless mechanically ventilated	Yes; -30 Pa in laboratory	Yes; -70 Pa in laboratory; -30 Pa in air lock; alarm system required
Supply filtered	—	Yes	Yes	HEPA filtered
Monitor air pressures	—	No	Yes, on supply	Yes
Effluent treatment	No	No	HEPA filtration of extract air	Double HEPA filtration of extract air, treatment of liquid waste and solid waste
Microbiological safety cabinet/enclosure	No	Yes, where airborne hazard	Yes	Yes
Safety cabinet class (user defined)	—	Class I	Class I, II or III	Class III
Autoclave site	—	In suite	In suite	In laboratory, double ended
Emergency shower	Agree with users	Preferred; agree with users	Yes	Yes

humidity and temperature ranges 24 hours a day so as to minimise damage to the collections they contain.

Historic materials are vulnerable to the following types of damage:

— physical damage, due to expansion, shrinking and cracking

— chemical deterioration, due to corrosion in damp conditions or by pollutants

— bio-deterioration, i.e. destruction by moulds or insects

Damage is caused by the action of atmospheric moisture, heat, direct sunlight, ultraviolet radiation, and external and internal atmospheric contaminants. It is often different combinations of these factors acting together, rather than individual factors acting in isolation, that cause significant damage.

3.17.2 Design considerations

Different materials may have their own distinct requirements. This means that conditions within a building may need to vary in different locations to suit their specific requirements. Since objects, books and archives may be added to, changed or re-organised, it is important that allowance be made in the design for varying the conditions within the space in order to match changing needs. This must be commensurate with sound energy-efficient practice.

The particular physical condition of objects or groups of objects may necessitate different environmental conditions. Therefore specific ranges of relative humidity for the conservation of historic materials can be decided only in discussion with whoever is responsible for their physical well-being, usually a conservator. When this does not take place, the design is often based on idealised ranges that may be inappropriate.

Seasonal differences in the moisture content of fresh air need to be considered when determining the appropriate level of ventilation air; for instance, in winter external air often has a lower moisture content than in summer. While fresh air ventilation is necessary for human respiration, historic materials may also require air to be replenished in order to reduce the concentration of contaminants from off-gassing materials. This needs to be balanced against the potential for transporting harmful external pollutants into the building by ventilation. If mechanical ventilation is fitted, the use of particle and gaseous filtration is recommended for historic materials vulnerable to external pollutants that are likely to be of high concentration in urban locations. It is advisable for mechanical ventilation to be controlled by carbon dioxide sensors in order reduce the fresh air supply to the minimum requirement.

Materials such as paper, parchment, textiles, leather and wood may be kept within the broad range of 40% to 65%. However, the rate of change must be controlled because maintaining a stable relative humidity is more important than an actual set point within the range.

Metals and minerals benefit from an RH level below 50%, while bronze and glass should be kept below 40% RH. In areas where large numbers of people may congregate, it is important to consider that while the human comfort RH range of 40% to 60% may be suitable for most materials, those that require drier conditions may need to be displayed or stored within microclimates.

For room temperature, the range 18-24 °C, which is acceptable for human comfort and commensurate with good energy efficient practice, is also acceptable for historic materials. However, where materials have become acclimatised to a more elevated temperature, active cooling should only be considered after discussion with the conservator and, if appropriate, the conservation architect. Temperatures lower than 16 °C may be desirable for some materials such as photographs and film or where the temperature may be designed to vary in order to maintain a stable relative humidity.

3.17.3 Environmental control

Typical means of achieving controlled conditions in other building types can also be used in museums, libraries and archives. These are close control air conditioning, the use of dessicant or refrigerant rehumidifiers and, where conditions become too dry, humidifiers.

Conservation heating is specific to environmental control in historic buildings. However this strategy is more appropriate to spaces where human comfort conditions are not required throughout the year. Conservation heating consists of control of heating systems with humidity and temperature sensors to provide environmental conditions for long-term conservation of objects, books and documents. Indoor relative humidity may need to be reduced at any time of the year, so the control systems should be set up to operate continuously.

Typically, a conservation heating system will maintain room temperatures 3-5 °C above their 'unheated' level in winter. This is in contrast with domestic winter heat input, which is designed to provide an average temperature increase of 8-10 °C. In good summer weather, there may be no call for corrective action for weeks on end, but weather changes can quickly produce damaging humidity conditions. Monitoring shows that the total heat input during the summer is small but important. This low level of heat input means that energy consumption is significantly lower than that for domestic heating systems. Depending on the size of the space to be controlled, solutions may vary from a single humidistat-controlled electric radiator to full multi-zoned schemes with computer building management systems.

Human beings do not generally notice changes in relative humidity, therefore locations with historic materials should be provided with instrumentation for the monitoring relative humidity and temperature.

Mechanical ventilation systems for libraries need to be designed to meet the sound control requirements of CIBSE Guide B5: *Sound control*[39]

3.17.4 Further reading

Blades N, Oreszczyn T, Bordass B and Cassar M Guidelines on Pollution Control in Museum Buildings *Museum Practice* (November 2000)

Thomson G *The Museum Environment* (London: Butterworth) (1986)

Museums, libraries and archives Chapter 20 in ASHRAE Handbook: *HVAC Applications* (Atlanta GA: American Society of Heating, Refrigerating and Air-Conditioning Engineers) (1999)

BS 5454: 2000: *Recommendations for the storage and exhibition of archive documents* (London: British Standards Institution) (2000)

3.18 Plant rooms

Plant areas should be ventilated as necessary to ensure the correct operation of equipment and the safety, health and comfort of personnel.

3.18.1 Boiler rooms

Boiler rooms and other spaces containing fuel-burning appliances must be supplied with adequate fresh air to meet the requirements for combustion and to prevent overheating of the space. Compliance with the regulations governing the ventilation of such appliances must be maintained. Details of these requirements are given in relevant Building Regulations Part J[3], British Standards e.g. BS 6798[23], BS 5410[24] and BS 5440[25]. Reference should also be made to CIBSE B13[92].

Rooms containing a gas installation should be ventilated to prevent the accumulation of gas such as could occur from minor gas leaks. Ducts containing gas pipework should be ventilated to a safe position, preferably direct to outside air. Measures for routing pipework may include enclosing the pipework in a ventilated gas-tight sleeve ('pipe-in-pipe'). It should be ensured that ventilation arrangements do not impair any provisions for fire/smoke separation. Refer to Gas Safety Regulations[93], Council for Registered Gas Installers (CORGI) and Institution of Gas Engineers[94] for guidance.

Oil tank chambers should be ventilated to the open air to prevent stagnation, independently of any other portion of the premises and preferably by natural means.

3.18.2 Refrigeration plant rooms

Refrigeration plant rooms should be provided with ventilation as required for the safety, health and comfort of personnel and for emergency purposes in the event of a major leak. Reference should be made to BS 4434[93].

3.18.3 Battery rooms

Depending on type of batteries present, ventilation should be provided so that any potentially explosive gaseous mixtures are dispersed safely below non-hazardous levels. Battery life can also be reduced by high continuous space temperatures, e.g. temperatures greater than 25 °C[95].

3.18.4 Electrical plant rooms

Particular care should be taken to ensure adequate ventilation for rooms containing electrical plant to prevent build-up of heat generated by the equipment.

These include the following:

— IT, communications rooms and incoming frame rooms that have active heat generating equipment
— transformer rooms
— electrical switchrooms
— uninterruptable power supply (UPS) rooms.

3.18.5 Water storage areas

Storage temperatures should comply with the requirements of the Water Regulations[96,97] and CIBSE and HSE recommendations concerning the growth of *Legionella*[98,99].

3.18.6 Lift motor rooms

Reference should be made to CIBSE Guide D[100]

3.19 Schools and educational buildings

3.19.1 Schools

3.19.1.1 General

The DfEE's *Guidelines for Environmental Design in Schools*[101] recommend that, as far as possible, schools buildings should be naturally ventilated. Exceptions are wcs, changing rooms, craft design and technology areas, kitchens, laboratories and other special activity areas where contamination or high heat gains might occur that may require local or other mechanical ventilation.

3.19.1.2 Requirements

Table 3.18 lists some required ventilation rates drawn from the Schools Premises Regulations[102].

Airtightness

A level of airtightness for schools is not specified although a maximum of 0.3 ACH has been suggested[103]. This is required to minimise heat losses when the building is unoccupied.

Air movement

Air movement at the level of the occupant must be at a temperature and velocity to ensure comfort. Natural ventilation should therefore be controllable to allow users to adjust the ventilation rate as required. Adjustments should be achieved by the appropriate use of window types and opening sizes, including trickle ventilators.

Ideally openings should be provided in more than one face of each room to maximise cross ventilation. Guidance is available on the passive solar design of schools to facilitate solar-induced stack effect to encourage ventilation on days with little wind[104]. Passive stack enhancement may also be considered.

Particular care should be taken to ensure that any odours arising from the use of volatile organic compounds (vocs) during construction work, or arising from school activities, can be dealt with.

Make-up air

Make-up air may be taken from surrounding spaces if this will not increase ventilation rates in teaching spaces beyond that required, in which case a secondary supply of fresh air may be provided.

Window selection

Sash windows are often used in schools because they provide high and low level openings, thereby giving occupants a considerable amount of control. However, only 50% of their area is available for ventilation. Side-hung casement windows give a greater openable area but care must be taken to ensure that they do not present a safety hazard when fully open[103]. In upper stories, the opening of windows is often restricted to minimise the risk of children falling out of windows.

Atria

Care should be taken with the design of atria within schools premises, which may be provided as low cost teaching space and buffer zones to classrooms. Ventilation provision must be sufficient to prevent overheating without compromising acoustic separation[103].

Draught lobbies

Effective draught lobbies should be specified where possible to minimise the amount of disadvantageous ventilation caused by occupants moving in and out of the building[103].

3.19.1.3 Further considerations

The Department for Education and Employment (DfEE) has embraced the concepts of environmental assessment of its premises. It places emphasis not just on energy use but also on ease of maintenance. Guidance can be found in DfEE publications[101,109]. Designers can gain credits by:

- demonstrating due consideration to the provision of ventilation (including the client and user in the development of the design with regards to risk assessment)
- the timely provision of completed record drawings and operation and maintenance (O&M) manuals
- the provision of training on the operation of any controls to the caretaker.

3.19.2 Higher education premises

3.19.2.1 Residential accommodation

Developments in the UK have demonstrated the potential for low energy residential accommodation, both through high levels of insulation and mechanical ventilation with heat recovery[110] or through passive ventilation via trickle ventilators and local mechanical extract where required[58].

3.19.2.2 Lecture theatres, study areas and design studios

Occupancy patterns can be dense but intermittent, or extended but sparse. Environmental control tends to be remote from the individual occupants. Teaching spaces designed to serve more than 100 people usually require some form of mechanical ventilation, although this may be as part of a mixed mode approach e.g. punkah fans within ventilation stacks. High levels of thermal mass and night cooling can also be effective in reducing energy demand. Control of airflow rates can be achieved through CO_2 sensors to establish a minimum rate. Care must be taken in the case of naturally ventilated solutions to avoid noise problems from external sources.

Table 3.18 Required ventilation rates in schools premises

Area	Ventilation rate	Notes
General teaching areas	3 litre.s^{-1} per person as minimum	
	8 litre.s^{-1} per person for rapid ventilation by opening windows or vents	Ventilation systems, whether natural or mechanical, should be capable of providing approximately 8 litre.s^{-1} per person of fresh air in all teaching areas medical examination or treatment rooms, sleeping and living accommodation
		Adequate measures should be taken to prevent condensation and remove noxious fumes from every kitchen and other room in which there may be steam or fumes.
		Guidance specific to the education sector with regards to health and safety issues as described in the Workplace Regulations[105] has been produced by the HSE[106]
Laboratories	—	To satisfy COSHH requirements[9] and DfEE guidance on fume cupboards[107]
Wash rooms	6 ACH minimum	
Swimming pools	—	Refer to specialist guidance[108] and section 3.21.7

The breadth of design options available for innovative low energy designs is illustrated in case studies by BRECSIP[111-113] and the Higher Education Estates Department*.

3.19.2.3 Specialist areas

Guidance on suitable treatments for other types of space found within higher and further education premises such as laboratories, learning resource centres, swimming pools, catering facilities can be found elsewhere in section 3.

3.20 Shops and retail premises

3.20.1 General

The general aim of the ventilation and air conditioning strategy is to provide a comfortable environment within the occupied zone. This is achieved by providing fresh air for the customers and staff and the removal of the heat from the space which arises from lighting, equipment, solar and occupancy gains.

3.20.2 Design requirements

The temperature within the space will vary according to season but is typically 18-22 °C depending on the requirements of the retailer. The upper limit may be permitted to rise in summer to prevent an unacceptable temperature differential between the retail space and the circulation space outside (i.e. outside or covered mall).

Minimum fresh air should be provided to satisfy occupancy loads based on the client's requirements or Building Regulations Approved Document F[2], whichever is the greater. Fresh air is typically introduced at a minimum rate of 5 $litre.s^{-1}$ per person. This rate is lower than the minimum stated in Building Regulations of 8 $litre.s^{-1}$ per person, which is for an occupiable room that is defined as not including a shop or circulation space. These require a minimum of 1 $litre.s^{-1}.m^{-2}$. The typical minimum fresh air rate is based on a typical occupancy of 1 person per 5 m^2. This fresh air rate is for a retail area in which smoking is not permitted. Minimum fresh air for occupation is supplied to the space via a supply AHU or via an extract fan in conjunction with openings on an external wall.

Heat gains will be a function of the building and the specific application but will often be characterised by one or more of the following:

— transient occupancy with high peak value

— high solar gains local to large areas of glazed shop front

— high lighting gains for display purposes

— localised equipment loads, e.g. hot food counters.

*Estates Team, HEFCE, Northavon House, Coldharbour Lane, Frenchay, Bristol BS16 1QD (http://www.heestates.ac.uk)

Infiltration of air from the outside due to door opening can be a particular concern. The problem may be exacerbated if there are openings on opposite facades of a store encouraging cross-ventilation driven by wind or stack forces (e.g. if opening onto a shopping mall). Locating openings on a single façade will help to balance these forces. Draughts within stores caused by infiltration can be minimised by the sealing of the building structure or the use of lobbies on entrances to deflect/direct airflow.

3.20.3 Strategies

Ventilation and air conditioning of the space can be achieved by various methods using centralised or unitary equipment. The choice of plant is governed by the retailer's particular requirements, the availability of external plant space, the size of the retail space and the availability of services supplied by the lessor.

Systems served by centralised plant can take the form of displacement or constant volume systems, both using recirculation or free cooling to provide the volume necessary to enable distribution of conditioned air at an acceptable temperature. Examples of minimum fresh air systems include unitary cooling split DX air conditioning or 2- or 4-pipe fan coil units.

Consideration should be given to recovery of heating/cooling energy that would normal be rejected. The ventilation system design may allow for the integration of air-to-air heat recovery devices, which transfer heat from the exhaust air stream to allow fresh air inlet. Waste heat from air-cooled condensers used in the refrigeration process may be recycled and utilised to reduce the load on space-heating plant. Cooling recovery at low level using spilled air from display cabinets may be recycled and introduced to cold stores etc, reducing the loads on cooling plant.

It is now common for major outlets to be provided with a water loop for the air conditioning system. This provides users with the flexibility to provide their own heat pumps as necessary to meet their individual requirements. This type of system may also balance well with the diversity of activities undertaken by the occupier, often requiring simultaneous heating and cooling. Water source heat pump systems are well to meet such requirements.

Leakage and build-up of refrigerant in a public space can be a danger to health due to decomposition products from smoking or naked flames in the presence of certain refrigerants. The occupier should prepare an emergency procedure to be followed in the event of leakage occurring. BS EN 378[87] should be consulted for guidance on correct procedures.

Ventilation rates within constant volume systems can be controlled using CO_2 or air quality sensors. Temperature control of central systems should be averaged where possible either using space sensors or a duct-mounted sensor in the extract system. Temperature control of unitary systems should be by individual or group controller, depending on the number of systems.

The building should be maintained under positive pressure by ensuring that the rate of supply exceeds the rate of extract.

Extracts should be positioned in the areas of high heat gain, e.g. lighting displays or hot food counters.

For food stores, the type and performance of refrigerated display cases will influence the design of the ventilation and conditioning system in a number of ways:

— Display cases may require temperature and humidity levels within the space to be maintained below maximum limits.

— Losses from display case will locally cool the space.

— Display cases with integral heat rejection will provide a net heat input to the space.

— The performance of display cases is susceptible to draughts from doors and ventilation systems.

Losses from display cases can vary quite significantly, depending on case design. Refrigerated areas commonly require heating throughout the year. The losses can lead to a 'cold aisle' effect in refrigerated areas of a store. One means of reducing this effect is by recovering some of the cold air spilt from the display cases, which may then by used to cool other areas of the store via the ventilation system.

Some display cases reject heat to the space, rather than to external heat rejection plant via a refrigeration system. Such display cases will impose a net heat gain on the ventilation system.

Internal draughts into cases from the ventilation system are avoided by the careful positioning of supply points from the ventilation and air conditioning equipment.

Smoke extract from retail units may be installed as separate stand-alone systems, which act as additional safety ventilation systems, or be incorporated into the general ventilation systems which serve the retail unit (known as 'dual purpose'). There are three possibilities in smoke extraction design each with a different purpose:

(a) *Life safety:* systems designed to maintain tenable conditions on escape routes and other occupied areas.

(b) *Firefighting access/property protection:* systems designed to increase visibility for, and reduce heat exposure to, trained fire fighters. This allows earlier and less hazardous attack on the fire. Such systems will help to reduce property damage by increasing fire brigade effectiveness.

(c) *Smoke purging:* systems designed to enable smoke to be cleared from a building after a fire has been brought under control.

It is necessary to decide which, or which combination, of these three objectives is to be achieved before commencing a design. BRE Report BR 368[114] should be consulted, in conjunction with BS 5588: Parts 9,10 and 11[115].

3.21 Sports centres

3.21.1 Ventilation requirements

The recommended environmental conditions and ventilation rates for sports centres vary according to the activities being undertaken, see Table 3.19.

Table 3.19 Environmental conditions for sports centres[116]

Facility	Temperature / °C	Ventilation
Multi-purpose centre:		
— sports activities	12-18	8-12 litre.s^{-1} person^{-1}
— sedentary activities	18-21	8-12 litre.s^{-1}-person^{-1}
Fitness centres	16-18	10-12 ACH
Weight training	12-14	10-12 ACH
Squash courts:		
— courts	16-18	4 ACH
— spectators	18	4 ACH
Ancillary halls:		
— sports	15	1.5 ACH
— non-sports	21	3 ACH
Changing rooms	20-25	10 ACH
Reception, administration and circulation spaces	16-20	Up to 3 ACH
Creche	21	Up to 3 ACH
Refreshment and bar areas	18	Not less than 8 litre.s^{-1} person^{-1}*
Swimming pool	27-31†	4-6 ACH 8-10 ACH if extensive water features

* Consult local licensing authority
† At least 1 K above water temperature

3.21.2 Multi-purpose halls/facilities[117]

Ventilation is required to remove players' body heat and odours, supply fresh air, keep spectators cool, maintain comfortable summertime conditions and prevent condensation. If the facility is also to be used for public entertainment, the relative importance of these functions depends on the activities taking place in the hall and the number of people present.

The ventilation system should be designed for controlled ventilation rates that can vary according to the occupants' needs at any given time, without introducing large volumes of cold air into the space that may cause discomfort and high heating loads.

For badminton, a draught-free playing area should be provided with air velocities less than 0.1 m.s^{-1} to prevent deflection of the shuttlecock. The location of inlet and extract grilles and openings must also be considered with regards to the flight paths of the shuttlecocks[118].

3.21.3 Fitness suites and weight training facilities

Effective ventilation is usually the most critical factor because of the metabolic heat gains, body odour and humidity that can rapidly occur in such spaces. Air conditioning is sometimes used but alternative, less energy-intensive approaches should be considered.

Special considerations may need to be made for spas, saunas, and solaria.

3.21.4 Squash courts

Squash courts should be well ventilated to keep walls free from condensation and remove the players' body heat, which can be considerable. Incoming air must not be drawn from changing rooms, bar areas, showers or any other parts of the building with high humidity levels.

In general, each court should have an extract fan centrally placed at high level. Fresh air can be drawn in through airbricks behind the playboard. This should be perforated to provide 10% free area.

Extract fans should over-run for 15 minutes after the courts have been vacated to ensure that all stale air has been removed. Fans should be linked to the court lighting circuit where practicable. The rate of ventilation in the spectator gallery may have to be based on maximum occupancy.

3.21.5 Ancillary halls

Ancillary halls may be used for a variety of both sporting and social activities, including public entertainment. Therefore the range of potential activities should be confirmed with the client prior to finalising the design of the ventilation system. A wide range of air change rates may be required, e.g. to remove smoke and ventilate the space for discos and dances. Consultation with the local licensing authority may also be necessary if the hall is to be used for public entertainment.

3.21.6 Changing rooms

These normally require a mechanical supply and extract system in larger facilities. In small facilities, satisfactory conditions may be achieved with conventional radiators and convectors combined with natural ventilation or local extract fans. The high fresh air requirement offers the opportunity for heat recovery to be cost effective.

3.21.7 Swimming pools

The recommended pool water temperature varies depending upon the activity. For competition swimming the pool is held at 26 °C. For leisure use a temperature range of 28-30 °C is more appropriate; for spas, remedial and other hot pools a pool temperature of 36 °C may be maintained. The air temperature in the pool hall should be at a minimum of 1 K above the pool water temperature. Such environmental conditions tend to create high humidity, therefore ventilation should be provided in order to:

- control humidity
- prevent condensation on the inner surface of the structure
- maintain a satisfactory indoor environment including the prevention of down-draughts
- remove airborne pollutants
- dilute disinfectant fumes.

Humidity levels within the pool hall should be maintained between 50-70% RH. For design purposes, airflow rates of 10 $litre.s^{-1}$ per m^2 of total pool hall area and a minimum of 12 $litre.s^{-1}$ per person of outside air should be provided[119]. Overall air change rates of 4-6 ACH are recommended for standard use or 8-10 ACH where there are extensive water features.

Supply and extract rates should be balanced, or preferably set to maintain a marginally lower pressure in the pool hall than outside or in the adjoining accommodation. This will inhibit the migration of moisture and odour. Although bathers out of the water will be susceptible to draughts, air movement at the pool surface must be sufficient to prevent the accumulation of gases released from the chemically treated water.

Warmed air should be provided to maintain changing rooms at 24 °C, and preferably supplied at low level to assist in floor drying if no provision is made for under floor-heating. Permanent extraction from the clothes storage area should be balanced by an air supply at a rate of 6-10 ACH. A separate extract system should be provided for the wcs.

Ventilation systems for swimming pool halls are either 100% fresh air systems or partial recirculation systems. The latter allow the fresh air supply to be adjusted while maintaining the overall supply volumes to the pool hall, hence maintaining air distribution patterns. However, it is essential that damper positions and control regimes are arranged to ensure adequate introduction of fresh air (30% minimum) and expulsion of contaminated air. Internal accumulations of chlorinous by-products are damaging to the building fabric and potentially dangerous to people. Therefore it is necessary to ensure a minimum ventilation rate at all times when the pool is occupied.

Savings can be made by minimising the intake of outside air for the 100% fresh air system using two-speed or variable speed fans. The impact of any savings is increased if a pool cover is used during periods when the pool is not in use. Both 100% fresh air and recirculation based systems are suitable for installing heat recovery and heat pumps or dehumidification systems.

Extract air from pool halls can be corrosive to the internal surfaces of ventilation systems. Adequate protection should be provided for exposed internal surfaces if maintenance and replacement costs are to be kept to a minimum.

3.21.8 Ancillary areas

Suitable ventilation systems must also be provided for ancillary areas as shown in Table 3.19. Office areas, rest rooms and circulation spaces may be serviced by natural ventilation. Mechanical extract will be required in kitchen areas to ensure that odours do not reach public spaces. There must be adequate ventilation and segregation for smoking.

3.21.9 Operational issues

Ventilation systems can consume nearly half of the energy used in sports centres. Within areas other than swimming pools, more efficient ventilation can be obtained by using the following[120];

— variable speed fans to cope with varying occupancies or activities, linked to modulating dampers using automatic humidity control

— ventilation heat recovery and recirculation, which can reduce running costs for sports centres by 10%.

Maintenance costs represent a significant proportion of the total expenditure on a sports or recreational building over its lifetime. Routine tasks will be made much easier if appropriate space is allocated for plant rooms, voids and distribution routes. Inspection of many mechanical items will need to take place every three months so there should be easy access to:

— dampers

— fans

— filters

— flexible connections

— heat exchangers (plate heat or run-around coils).

3.21.10 Further reading

John G and Campbell K *Handbook ofsports and recreational building design Vol. 2: Indoor sports* (London: Sport England) (1995) ISBN 0 7506 1294 0

John G and Campbell K *Handbook ofsports and recreational building design Vol. 3: Ice rinks and swimming pools* (London: Sport England) (1996) ISBN 0 7506 2256 3

Swimming pools — building services Sports Council Guidance Note 387 (London: Sport England) (1995)

3.22 Toilets

The Building Regulations[2,121] make specific provision for the ventilation of toilets. In England and Wales, for dwellings, one or more ventilation openings must be provided of area 1/20th. of the floor area (some part of which must be at high level, i.e. at least 1.75 m above floor level), or mechanical extract must be provided at a minimum rate of 6 litre.s^{-1}. In non-domestic buildings, sanitary accommodation (which includes washing facilities) again requires either one or more ventilation openings of area 1/20th. of the floor area (some part of which must be at high level) or mechanical ventilation at a minimum rate of 6 litre.s^{-1} per wc or 3 air changes per hour.

In Scotland, for dwellings, a ventilator must be provided of area 1/30th. of the floor area (some part of which must be at least 1.75 m above floor level), or mechanical extract must be provided at a minimum rate of 3 air changes per hour.

Toilets are very often provided with the absolute minimum ventilation to comply with the regulations, in order to achieve very minor cost savings. The result of such economy can be a very unpleasant toilet atmosphere. This unpleasantness is easily avoided at very marginal extra cost by ensuring that the ventilation system exceeds the statutory requirements.

3.23 Transportation buildings and facilities

3.23.1 General

The exhaust gases produced by combustion engines contain toxic components and smoke. Wherever vehicular access is provided it is necessary to consider how ventilation can be provided that will limit the concentrations of dangerous contaminants to permitted and/or acceptable limits.

3.23.2 Tunnels

Road tunnels require ventilation to remove the contaminants produced by vehicle engines in normal use. Ventilation may be provided by natural or mechanical means, or may be traffic induced. Detailed requirements for the ventilation of road tunnels are published by the Highways Agency[122].

Railway tunnels are subject to the requirements of both owners/clients and the Health and Safety Executive's Railways Inspectorate, who should be consulted for detailed design requirements.

3.23.3 Car parks

The general requirement is for engineering systems that will remove the hazards of carbon monoxide from vehicle exhaust emissions and prevent the build up of vapours from fuel leaks etc. The increasing use of diesel engined vehicles also requires control of airborne particles from the burnt fuel.

Above-ground car parks should be provided with natural ventilation openings in the outside walls of at least 5% of the floor area. Openings on opposite sides should be provided to promote ventilation without being adversely effected by wind direction.

Mechanical ventilation is required for car parks that are enclosed or located in basements. The system should be independent of any other systems and provide 6 ACH for normal operation and 10 ACH in a fire condition. Extract points should be placed so as to eliminate pockets of stale air, and be distributed so that 50% of the extract is a high level and 50% at low level, with particular attention at low points and drains. The system should be divided into two parts, each connected to an independent power supply that will continue to operate in the event of mains failure.

Where many vehicle engines are likely to be running simultaneously, e.g. at exit and entrances, consideration should be given increasing the ventilation rates to maintain the acceptable contamination levels based on vehicle emissions. Limiting concentrations of exhaust pollutants are included in the HSE's annual guidance publication EH40: *Occupational Exposure Limits*[10]. If separate from the general car park ventilation system, the ventilation can be controlled using carbon dioxide detectors at appropriate locations.

Manned pay stations may need positive supply air, with the air intake located away from the contaminated roadways.

For further information see ASHRAE Handbook: *HVAC Applications*[123].

3.23.4 Bus terminals

Bus terminals vary considerably in physical configuration. Ideally, buses should be able to drive through a loading platform and not have to manoeuvre within the area.

Naturally ventilated terminals with large open ends may expose passengers to inclement weather and strong winds. Therefore, enclosed platforms with appropriate mechanical ventilation should be considered. Alternatively, enclosed passenger waiting areas can be considered for large terminals with heavy bus traffic. The waiting areas can be pressurised and heated, with normal air volumes depending on the layout and number of boarding gates.

The exhaust gases from diesel engines that affect the ventilation design are carbon monoxide, hydrocarbons, oxides of nitrogen and formaldehydes. Exposure limits are given in HSE EH40[10].

The ventilation rate also needs to provide odour control and visibility, which would generally require a 75:1 dilution rate of outside air to bus exhaust gases. The overall rate of fume emission can be determined from considering the bus operation, terminal configuration and traffic movements. The overall ventilation required can be reduced by removing exhaust gases at the point of discharge.

The guidance given above relates to diesel engined vehicles. However, the use of alternative fuels is increasing and these also need to be considered. For buses fuelled by natural gases, the normal emission rate of unburnt fuel is low. However, if the high pressure gas fuel line were to break, then a large quantity of gas would be released causing a potentially explosive atmosphere. Such a situation would require the prompt use of purging ventilation. Initially, the gas, while cold, will collect at ground level and therefore purging needs to be at this level. However, when warmed, methane tends to rise, as will unburnt methane in the exhaust gases. Therefore, potentially stagnant air zones at high level need to be eliminated.

For further information see ASHRAE Handbook: *HVAC Applications*[123].

3.23.5 Enclosed loading bays

The requirement in ventilating enclosed loading bays is for the dilution of exhaust gases in normal operation and provision for smoke extract under fire conditions.

Consideration should be given to the nature of the loading bay and vehicle movement in order to develop a system that will meet the required standards. Generally, the large entrance door will provide the necessary inlet air and the fume extract can be combined with the smoke extract for general ventilation. As with car parks and enclosed bus terminals, extract should be provided at high and low levels.

3.23.6 Garages (vehicle repair)

In view of the dangerous nature of the accessories to the repair and storage of motor vehicles and the risk of pollution from waste gases and products, the heating, ventilation, fire protection and safety of functional structures is regulated.

Ventilation systems should be designed to limit the contamination levels to acceptable limits[9,106]. Where vehicles are stationary at fixed repair stations, direct exhaust for the emissions should be provided by means of a flexible hose and coupling attached to the tailpipe. The use of such systems will reduce the overall ventilation requirement.

Particular care needs to be taken where inspection and repair pits are present as vehicle and fuel fumes, being heavier than air, will tend to flow into these areas. Therefore a separate extract system is required.

Where garages contain spray booths the relevant codes must be complied with.

For further information see ASHRAE Handbook: *HVAC Applications*[123].

3.23.7 Railway stations/terminals, underground railway stations

Where the railway tracks are enclosed under a canopy or buildings above, it will be necessary to consider how the fumes produced by the locomotives are to be exhausted/diluted.

The design requirements will be similar to those for bus stations, i.e. reduce the level of contaminants and odours to acceptable limits and provide sufficient air circulation to maintain visibility.

For further information see ASHRAE Handbook: *HVAC Applications*[123].

3.23.8 Airport terminals

Airports generally consist of one or more terminal buildings, connected by passageways to departure gates. Many terminals have telescoping loading bridges connecting the departure lounges to the aircraft. These eliminate heating and cooling problems associated with open doorways.

The aim of any ventilation system should be to create a positive internal pressure that will prevent the odour and pollutants from entering the buildings.

Terminal buildings have large open circulation areas, check-in facilities, retail outlets, offices and ancillary areas. As occupancy can vary considerably through the day, it is important that the ventilation/air conditioning system is able to respond to the changes in occupancy. However, due to the large volume of the circulation spaces it is possible to use the building volume to absorb the sudden changes and peak flows. Ventilation systems can be designed with recirculation (to provide heat reclaim), controlled by air quality detectors, thereby automatically reacting to passenger flows.

The system design should also incorporate sufficient zone control to accommodate the widely varying occupancy levels in different parts of the building, or even between adjacent departure gates. If available, histograms on passenger movement for departure and arrival are useful in estimating the design occupancy.

Filtering of the outdoor air with activated carbon filters should be considered to reduce the presence of noxious fumes. However, the siting of air intakes away from the aircraft jet exhausts may obviate the need for filtration and will reduce operating costs. However, since it may be difficult to predict if fumes will affect the air intake location, supply systems should incorporate facilities to enable carbon filters to be added at a later stage, if necessary.

3.24 Miscellaneous sectors

The following information on other sectors has been included mainly to identify specialist sources of information that are available. Material from the previous edition of CIBSE Guide B has been included but not necessarily updated and designers are advised to obtain specialist advice for current guidance.

3.24.1 Animal husbandry[124,125]

3.24.1.1 Farm buildings

Guidance on the housing of animals on farms may be obtained from the Animal Welfare Division of the Department for Environment, Food and Rural Affairs (DEFRA). Reference should also be made to the Welfare of Farm Animals (England) Regulations[126].

Buildings for farm animals fall into two main groups:

— buildings for housing 'hardy stock', such as milking-cows, breeding-pigs and sheep, that do not require any great control of environmental conditions

— buildings such as pig farrowing houses, fattening houses, veal calf houses, laying and broiler poultry houses etc., which require the environmental conditions to be controlled such that the highest possible productivity is obtained at the lowest food and management costs.

Hardy stock require housing only to protect them from extremes of weather, ventilation being provided by low level and ridge ventilators with protection against direct and through draughts. However, care must be taken to ensure adequate ventilation in high density enclosed houses where forced ventilation will be necessary. Humidity is not usually a problem.

For animals requiring close control of conditions, mechanical ventilation is essential, provided by supply and/or extract fans depending on the requirement for positive or negative pressures within the houses. Winter recirculation can be used to conserve heat. Safeguards must be provided against fan failure or livestock will be seriously affected during hot weather. Adequate ventilation will also minimise the occurrence of high humidity. Table 3.20 gives optimum air temperatures and ventilation rates.

Table 3.20 Temperatures and ventilation rates suitable for housed livestock

Animal species	Optimum temperature range /°C	Ventilation rate	
		Winter/ (litre.s^{-1} per kg of body weight)	Summer/ (litres^{-1} per kg of body weight)
Adult cattle	0-20	0.5	0.20-0.38
Calves	10-15	0.10	0.26-0.53
Pigs	5-25	0.10	0.26-0.53
Piglets:			
— at birth	35	0.08	0.08
— after 2 days	28-33	0.06	up to 0.06
Fattening pigs	11-22	0.10	0.26-0.53
Laying poultry	20-25	0.4	1.5-2.6
Broiler chickens	15-25	0.2	0.8-1.3

3.24.1.2 Animal rooms[127-129]

The specification of the design for animal rooms would be undertaken by the holder of the premises certificate, with the approval of the Home Office local inspector. Designers must ensure that all necessary procedures are followed.

The environmental conditions and degree of control required for animal rooms depend on the species and the intended use of the facilities. Tables 3.20 and 3.21 show the conditions required for various animals and for different applications.

For precise experimental work, close control of temperature (±1 K) and relative humidity (±10%) may be required at different conditions within the overall

Table 3.21 Animal room environmental design conditions

Animal	Surface area / m^2	Average metabolic rate a 21°C*/W	Number of animals per 10 m^2 of floor area	Typical animal room gain / $W.m^{-2}$	Recommended temperature range/°C	Relative humidity / %
Mice	0.01	0.5	2000	100	21-23	40-70
Rats (at 60 days)	0.031	1.5	485	73	21-23	40-60
Guinea pigs (at 60 days)	0.07	3.0	400	120	17-20	40-70
Chicken:						
— at 4 weeks	0.04	2.4	230	55	21-23	40-60
— at 24 weeks	0.21	12.0	100	120	16-19	40-60
Rabbits (adult)	0.20	11.0	32	35	16-19	40-60
Cats	0.20	8.0	16	13	18-21	40-60
Dogs:						
— male	0.65	26.0	5	13	12-18	40-70
— female	0.58	22.0	5	11	12-18	40-70

*Based on resting metabolism

Notes:

(1) Assume 35-40% as latent gain.

(2) Figures should be used as a guide only and will vary depending on conditions.

(3) Animal numbers per m^2 based on figures for an average experimental holding room.

operational stage. Uniformity of the environment throughout the space is also important and in some cases the direction of air movement needs to be controlled to minimise, for example, the pollution in the spaces through which the laboratory operatives move.

Requirements may also include standby equipment and/or safety features that are automatically initiated in the event of a failure of the main system.

3.24.2 Call centres

Concern has been expressed regarding employee motivation and stress in telephone call centres. Little guidance has been produced on the ventilation aspects of call centre design, precedence being given to acoustic and lighting related issues. This is partly due to the disparate nature of call centres.

The ideal call centre is characterised by space 15-18 m in depth on a single level, operator teams of up to 12-15 people, large floor-to-ceiling heights, good ventilation, and lighting, and raised floors[130]. However, in reality, call centres are housed in a large variety of building types from converted warehouses to highly specified office buildings.

Space occupancy densities also vary. There may be as little as 6-7 m^2 per person in a centre dealing with simple enquiries or as much as 10-14 m^2 per person allowed in newer, full service centres or where confidentiality is important[131]. The latter figure allows for support areas such as lounges, catering, training facilities and team meeting areas. The subsequent amount and concentration of heat gains will therefore vary and may result in the selection of, for example, VAV, chilled ceilings/beams or fan coils as appropriate.

In selecting a ventilation system the designer should be aware of:

— the intended staffing levels and how these might change in the future (e.g. additional staff) as this will affect cooling loads and the potential requirement for upgrading the system

— the pattern of changing staff levels over the day, week or seasonal basis (e.g. as a consequence of shift patterns); this could affect system zoning and the ability to use night cooling as a pre-cooling strategy, or as a free cooling strategy

— the degree to which staff operate IT equipment, e.g. single or multiple screen systems

— the anticipated importance of 'churn'; e.g. will temporary areas be screened-off for periods within open plan areas, thereby interrupting airflow and causing pockets of stale air?

— the maintenance constraints imposed by the system selection and shift arrangements; will it be necessary to isolate as much plant as possible away from the working space to facilitate ongoing maintenance?

— the potential need to separated off areas within the space (either by full height partitions or screens) to protect the open plan workstations from noise and distraction and to separate support functions and office equipment

— the support features that will be provided and whether or not they will require separate servicing; e.g. is there a need to isolate hot snack areas to prevent odours from drifting?

— the interaction between the ventilation system and individual staff; it is important to ensure a good quality environment across the entire space as staff will be unable to change the position of their work stations or alter the ventilation

— the possibility that the ventilation system will add to background noise levels and thereby affect the ability of staff to deal with incoming calls.

3.24.3 Courtrooms

The Lord Chancellor's Department (LCD) should be consulted for guidance on environmental policy and court room design[132,133].

The LCD's policy is to maximise the use of natural ventilation principles to maintain satisfactory environmental conditions as part of their commitment to provide environmentally friendly buildings.

Mechanical ventilation systems for court rooms need to be designed to meet the sound control requirements of CIBSE Guide B5: *Sound control*[39].

3.24.4 Darkrooms (photographic)

Small darkrooms for occasional use or for purely developing processes may often be ventilated naturally with a suitable light trap, although consideration should be given to providing mechanical extract using an air change rate of 6 to 8 air changes per hour.

For general purpose darkrooms, however, the air change rate should be ascertained from consideration of the heat gain from the enlarger, lights etc. plus the occupants, on the basis of a temperature rise of 5-6 K. In industrial and commercial darkrooms that have machine processing, the machines will very often have their own extract ducting, the air supply being drawn from the room itself. It will usually be necessary to provide a warmed and filtered mechanical inlet in such cases. In special cases, involving extensive washing processes, the humidity gain may be significant and require consideration.

3.24.5 Dealing rooms

Dealing rooms are characterised by much higher heat gains from IT equipment than those for general office areas. Small power requirements are typically in the order of 500 W per trading desk, but can vary between 200 W and 1000 W[134]. Occupation densities can be as high as one trading desk per 7 m^2.

Loads are a function of the IT equipment and are subject to technological developments. Developments may have spatial as well as power implications. These may affect load intensity. For example, flat panel displays (FPD) have lower cooling requirements than cathode ray tubes (SRT) displays, but occupy less space thereby permitting a greater density of occupation[135].

The selection of suitable air conditioning is primarily determined by the high cooling load. Because of the need to minimise disruption, reliability and maintenance requirements are also key considerations. Systems normally incorporate a high degree of redundancy. Risks associated with pipework and condensate leakage should be minimised.

Ceiling mounted system options include fan coils and variable air volume (VAV) systems. However, supplying cooling from above to deal with the heat from the equipment will create a large amount of air movement, thereby increasing the risk of draughts. An alternative approach is to supply cool air directly to desktop computers through the floor void to remove the heat directly, reducing air movement in the occupied space and the risk of draughts. This approach may be used in conjunction with the fan coil units, VAV system or chilled beams/ceilings that deal with the balance of the load.

Mechanical ventilation systems for dealing rooms need to provide a high level of reliability, as the system is critical to the proper functioning of the building and the business conducted within it. Consequential losses arising from failure can be very significant in this type of building.

3.24.6 Horticulture[136]

Guidance on sources of information on the design of horticultural buildings may be obtained from the Commercial Horticultural Association, a trade association based at the National Agricultural Centre*.

Environmental conditions in greenhouses must be favourable to plant growth. This involves heating during cold weather and the limitation of high temperatures due to solar gains in the hot weather. In some cases, carbon dioxide enrichment and humidity restriction will also be required. The internal design temperature should be in the order of 16 °C when the external temperature is -7 to -10 °C.

Greenhouse crops require ventilation to limit the rise in air temperature, provide carbon dioxide for photosynthesis, and restrict the rise in humidity due to transpiration. Automatic ventilators, controlled by an air thermostat, can be opened at a pre-determined temperature (approximately 24 °C). Rates of ventilation of the order of 30 to 50 litre.s^{-1} per m^2 of greenhouse floor area are desirable, which is equal to 45 to 60 air changes per hour for conventional houses. Low level ventilators may be required in addition to the ridge ventilators to increase the stack effect during still conditions.

Propeller extract fans (side wall mounted) with ventilation duties to the rates given have the advantage of positive air movement through crops, thus promoting growth. Inlet air should have a velocity not exceeding 1 m.s^{-1} and be diverted with an upward component, thus preventing cooler air being drawn directly on to the crops. A combination of automatic ventilators and fans will allow for failures of either system.

A complete mechanical ventilation system, using PVC ductwork with air supply discharge holes, can be used for winter heating with heated re-circulated air, and summer cooling with 100% fresh outdoor air. Fans giving a constant 10 to 20 ACH can be supplemented by automatic ventilators or extract fans during hot weather. This type of system has the advantage of even, closely controlled temperatures, with positive air movement throughout the year. However, the initial outlay is likely to be high.

*National Agricultural Centre, Stonleigh Park, Kenilworth, Warwickshire, CV8 2LG, UK (http://www.ukexnet.co.uk/hort/cha/)

Other aspects worth consideration are:

- automatic solar shading equipment
- automatic day and night temperature and lighting sequencing
- evaporative cooling pad air inlet and exhaust fan system
- air purification
- plant cooling by evaporation using overhead spraying
- earth heating plant propagation beds.

3.24.7 Standards rooms

It is usual for standards rooms to be designed to meet the same conditions as those maintained for the manufacturing processes, and reference should be made to the appropriate section. In practice, the environmental conditions within standards rooms may well be more exacting, in order to *(a)* sample and test equipment over varying environmental conditions for set time periods, or *(b)* sample and test equipment manufactured in various areas of the factory maintained at different environmental conditions.

Mechanical ventilation systems for standards rooms need to provide a high level of reliability, as the system is critical to the proper functioning of the building and the business conducted within it. Consequential losses arising from failure can be very significant in this type of building.

References

1 Jones P J, O'Sullivan P E et al. *Internal conditions and the response of office workers* Workplace Comfort Forum (London: Royal Institute of British Architects) (1995)

2 *The Building Regulations 1991 Approved Document F: Ventilation* (London: Department of Transport, Local Government and the Regions/The Stationery Office) (1995)

3 *The Building Regulations 1991 Approved Document J: Heat producing appliances* (London: Department of Transport, Local Government and the Regions/The Stationery Office) (1992)

4 *The Building Regulations 1991 Approved Document B: Fire safety* (London: Department of Transport, Local Government and the Regions/The Stationery Office) (2000)

5 *The Building Regulations 1991 Approved Document L: Conservation of fuel and power* (London: Department of Transport, Local Government and the Regions/The Stationery Office) (1995)

6 *Environmental design* CIBSE Guide A (London: Chartered Institution of Building Services Engineers) (1999)

7 Fanger P O The new comfort equation for indoor air quality *Proc. ASHRAE Conf. IAQ '89: The Human Equation — Health and Comfort, San Diego CA, USA, April 1989* (Atlanta GA: American Society of Heating, Refrigerating and Air Conditioning Engineers) (1989)

8 *Guidelines and Ventilation Requirements in Buildings European Concerted Action: Indoor Air Quality and its Impact on Man* Report No. 11 (Luxembourg: CEC Directorate General Information Market and Innovation) (1992)

9 Control of Substances Hazardous to Health Regulations (COSHH) Statutory Instrument 1999 No. 437 (London: The Stationery Office) (1999)

10 *Occupational exposure limits* EH40 (Bootle: Health and Safety Executive) (published annually)

11 *Air quality guidelines for Europe* (Copenhagen: World Health Organisation) (1998)

12 BS 5925: 1991: *Code of practice for ventilation principles and designing for natural ventilation* (London: British Standards Institution) (1991)

13 *Minimising pollution at air intakes* CIBSE TM21 (London: Chartered Institution of Building Services Engineers) (1999)

14 *Hygienic maintenance of office ventilation ductwork* CIBSE TM26 (London: Chartered Institution of Building Services Engineers) (2000)

15 Jansenn J E Ventilation and acceptable indoor air quality ASHRAE Standard 62-1981 *Proc. Annual American Conference of Government Industrial Hygienists* 10 59-64 (1984)

16 Fanger P O *Body odour and carbon dioxide, Minimum ventilation rates* IEA Energy Conservation in Buildings and Community Systems Programme: Annex IX (final report) (Paris: International Energy Agency) (1986)

17 Leaderer B P and Cain W S Air quality in buildings during smoking and non-smoking occupancy *ASHRAE Trans.* 89 (Part 2B) (1983)

18 *Proposal for an Approved Code of Practice on passive smoking at work* HSE CD151 (London: Her Majesty's Stationery Office) (1999)

19 *Ventilation for acceptable indoor air quality* ASHRAE Standard 62-1999 (Atlanta GA: American Society of Heating, Refrigerating and Air Conditioning Engineers) (1999)

20 Ionising Radiations Regulations 1999 Statutory Instrument 1999 No. 3232 (London: The Stationery Office) (1999)

21 Health and Safety at Work etc. Act (London: Her Majesty's Stationery Office) (1974)

22 Scivyer C R and Gregory T J *Radon in the workplace* BRE Report BR 293 (Garston: Building Research Establishment) (1995)

23 BS 6798: 2000: *Specification for installation of gas-fired boilers of rated input not exceeding 70 kW net* (London: British Standards Institution) (2000)

24 BS 5410: *Code of practice for oil firing:* Part 1: 1998: *Installations up to 45 kW output capacity for space heating and hot water supply purposes;* Part 2: 1978: *Installations of 45 kW and above capacity for space heating, hot water and steam supply purposes;* Part 3: 1976: *Installations for furnaces, kilns, ovens and other industrial purposes* (London: British Standards Institution) (dates as indicated)

25 BS 5440: *Installation and maintenance of flues and ventilation for gas appliances of rated input not exceeding 70 kW net (1st, 2nd and 3rd family gases):* BS 5440-1: 2000: *Specification for installation and maintenance of flues;* BS 5440-2: 2000: *Specification for installation and maintenance of ventilation* (London: British Standards Institution) (2000)

26 *Fire engineering* CIBSE Guide E (London: Chartered Institution of Building Services Engineers) (1997)

27 *Natural ventilation in atria for environment and smoke control: an introductory guide* BRE Report BR 375 (Garston: Building Research Establishment) (1999)

28 Rennie D and Parand F *Environmental design guide for naturally ventilated and daylit offices* BRE Report BR 345 (Garston: Building Research Establishment) (1998)

29 *Natural ventilation in non-domestic buildings* CIBSE AM10 (London: Chartered Institution of Building Services Engineers) (1997)

30 *A performance specification for the Energy Efficient Office of the Future* EEBPP General Information Report GIR 30 (London: Department of Transport, Local Government and the Regions) (1995) (available from http://www.energy-efficiency.gov.uk) (contained on CIBSE Members' CD-ROM)

31 *Best practice in the specification for offices* BCO Guide 2000 (London: British Council for Offices) (2000)

32 BS 5250: 1989: *Code of practice for the control of condensation in buildings* (London: British Standards Institution) (1989) (under revision)

33 *Thermal insulation: avoiding risks* BRE Report BR 262 (Garston: Building Research Establishment) (1994)

34 *Energy use in offices* EEBPP ECON 19 (London: Department of Transport, Local Government and the Regions) (1999) (available from http://www.energy-efficiency.gov.uk) (contained on CIBSE Members' CD-ROM)

35 The Building Regulations 2000 Statutory Instrument 2000 No. 2531 (as amended) (London: Her Majesty's Stationery Office) (2000)

36 *The Building Regulations 2000 Approved Document L: Conservation of fuel and power* (interim draft) (London: Department of Transport, Local Government and the Regions) (March 2001)

37 *Conference centres and lecture theatres* BSRIA Library Bulletin LB4/90 (Bracknell: Building Services Research and Information Association) (1990)

38 *Concert halls and theatres* BSRIA Library Bulletin LB18/93 (Bracknell: Building Services Research and Information Association) (1993)

39 *Sound control* CIBSE Guide B5 (London: Chartered Institution of Building Services Engineers) (in preparation)

40 Potter I N, Booth W B *CO_2 controlled mechanical ventilation systems* (Bracknell: Building Services Research and Information Association) (1994)

41 *Energy efficiency in buildings* CIBSE Guide F (London: Chartered Institution of Building Services Engineers) (1999)

42 Saxon R *Atrium buildings: Development and Design* (London: The Architectural Press) (1983)

43 Baker N V *Energy and environment in non-domestic buildings: a technical design guide* (Cambridge: Architectural Research) (1994)

44 *Design methodologies for smoke and heat exhaust ventilation within atria* BRE Report BR 368 (Garston: Building Research Establishment) (1999)

45 BS 5588: *Fire precautions in the design, construction and use of buildings:* Part 7: 1997: *Code of practice for atrium buildings* (London: British Standards Institution) (1997)

46 Goulding J R, Owen Lewis J and Steemers T C (eds.) *Energy in architecture — the European passive solar handbook* (London: Batsford/Commission for the European Communities) (1992)

47 *Guide to acoustic practice* (2nd edn.) (London: BBC Engineering Information Department) (1990) ISBN 0 563 36079 8

48 *Standard for kitchen ventilation systems* HVCA DW 171 (London: Heating and Ventilating Contractors' Association) (2000)

49 *Ventilation of kitchens in catering establishments* HSE Catering Sheet No. 10 (Bootle: Health and Safety Executive) (1997)

50 BS EN ISO 14644-1: 1999: *Cleanrooms and associated controlled environments. Classification of air cleanliness* (London: British Standards Institution) (1999)

51 *Airborne paniculate cleanliness classes in clean rooms and clean zones* US Federal Standard 209E (Washington DC: Superintendent of Documents) (1992)

52 *Rules and Guidance for Pharmaceutical Manufacturers and Distributors 1997* (London: Medicines Control Agency/The Stationery Office) (1997)

53 *Designing energy efficient multi-residential buildings* EEBPP Good Practice Guide GPG 192 (London: Department of Transport, Local Government and the Regions) (1997) (available from http://www.energy-efficiency.gov.uk)

54 *Energy efficient multi-residential accommodation, Panns Bank, University of Sunderland* BRE New Practice Initial Profile IP91 (Garston: Building Research Establishment) (1995)

55 *Continuous mechanical ventilation in dwellings: design, installation and operation* BRE Digest 398 (Garston: Building Research Establishment) (1994)

56 *Energy-efficient mechanical ventilation systems* EEBPP Good Practice Guide GPG 257 (London: Department of Transport, Local Government and the Regions) (1997) (available from http://www.energy-efficiency.gov.uk)

57 BS EN 779: 1993: *Particulate air filters for general ventilation — Requirements, testing, marking* (London: British Standards Institution) (1993)

58 *Surface condensation and mould growth in traditionally-built dwellings* BRE Digest 297 (Garston: Building Research Establishment) (1985)

59 Perera M D A E S and Parkins L M Build tight — ventilate right *Building Servs.J.* 14 (6) 37 (June 1992)

60 Stephen R K *Humidistat-controlled extract fans: performance in dwellings* BRE Information Paper IP5/99 (Garston: Building Research Establishment) (1999)

61 *Testing buildings for air leakage* CIBSE TM23 (London: Chartered Institution of Building Services Engineers) (1999)

62 Stephen R K *Ainightness in UK dwellings* BRE Information Paper IP1/00 (Garston: Building Research Establishment) (2000)

63 *Energy efficient ventilation in housing* EEBPP Good Practice Guide GPG 268 (London: Department of Transport, Local Government and the Regions) (1999) (available from http://www.energy-efficiency.gov.uk)

64 *Radon: guidance on protective measures for new dwellings* BRE Report BR 211 (Garston: Building Research Establishment) (1991)

65 BS 5864: 1989: *Installation in domestic premises of gas-fired ducted-air heaters of rated input not exceeding 60 k W* (London: British Standards Institution) (1989)

66 *Passive stack ventilation in dwellings* BRE Information Paper IP 13/94 (Garston: Building Research Establishment) (1994)

67 *Energy efficiency primer* EEBPP Good Practice Guide GPG 171 (London: Department of Transport, Local Government and the Regions) (1997) (available from http://www.energy-efficiency.gov.uk)

68 *A guide to energy efficient ventilation* (Coventry: Air Infiltration and Ventilation Centre) (1996)

69 Stephen R K *Positive input ventilation in dwellings* BRE Information Paper IP12/00 (Garston: Building Research Establishment) (2000)

70 *Continuous mechanical ventilation in dwellings* BRE Digest 398 (Garston: Building Research Establishment) (1994)

71 *Heat pumps in the UK - a monitoring report* EEBPP General Information Report GIR 72 (London: Department of Transport, Local Government and the Regions) (2000) (available from http://www.energy-efficiency.gov.uk)

72 Meyer-Holley A et al. *The future market potential for small scale air conditioning in the UK* BSRIA Report 79570/1 (Bracknell: Building Services Research and Information Association) (November 1998)

73 BS 6375: *Performance of windows:* Part 1: 1989: *Classification for weathertightness (including guidance on selection and specification);* Part 2: 1987: *Specification for operation and strength characteristics* (London: British Standards Institution) (dates as indicated)

74 BS EN 12207: 2000: *Windows and doors. Air permeability. Classification* (London: Chartered Institution of Building Services Engineers) (2000)

75 BS EN 12207: 2000: *Windows and doors. Watertightness. Classification* (London: Chartered Institution of Building Services Engineers) (2000)

76 BS EN 12207: 2000: *Windows and doors. Resistance to wind load. Classification* (London: Chartered Institution of Building Services Engineers) (2000)

77 *Energy efficient refurbishment of high rise housing* EEBPP NPFR 84 (London: Department of Transport, Local Government and the Regions) (1995) (available from http://www.energy-efficiency.gov.uk)

78 The Factories Act 1961 (London: Her Majesty's Stationery Office) (1961)

79 The Workplace (Health, Safety and Welfare) Regulations 1992 Statutory Instrument 1992 No. 3004 (London: Her Majesty's Stationery Office) (1992)

80 *Energy efficiency in industrial buildings and sites* Energy Consumption Guide ECON 18 (London: Department of Transport, Local Government and the Regions) (1999)

81 *The designer's guide to energy-efficient buildings for industry* EEBPP Good Practice Guide GPG 303 (London: Department of Transport, Local Government and the Regions) (2000) (available from http://www.energy-efficiency.gov.uk)

82 Alamdari F *Air curtains — commercial applications* BSRIA Applications Guide AG2/97 (Bracknell: Building Services Research and Information Association) (1997)

83 Bos J *Numerical simulation of a three-dimensional comfort air curtain* Master's Thesis (Groningen: University of Groningen) (date unknown)

84 *Commercial and Public Buildings* Chapter 3 in ASHRAE Handbook 1999: *HVAC Applications* (Atlanta GA: American Society of Heating, Refrigerating and Air Conditioning Engineers) (1999)

85 Lovatt J E, Wilson A G Stack effect in tall buildings *ASHRAE Trans.* 3825 (RP-661) 1994 Part 2 420-431 (Atlanta GA: American Society of Heating, Refrigerating and Air Conditioning Engineers) (1994)

86 *Ventilation in healthcare premises: design considerations* Health Technical Memorandum NHS Estates 2025 (London: The Stationery Office) (1994)

87 BS EN 378: *Specification for refrigerating systems and heat pumps. Safety and environmental requirements:* BS EN 378-1: 2000: *Basic requirements, definitions, classification and selection criteria;* BS EN 378-2: 2000: *Design, construction, testing, marking and documentation;* BS EN 378-3: 2000: *Installation site and personal protection;* BS EN 378-4: 2000: *Operation, maintenance, repair and recovery* (London: British Standards Institution) (2000)

88 *Industrial ventilation — a manual of recommended practice* (Lansing, MI: American Conference of Governmental Industrial Hygienists) (2001)

89 BS 7258: *Laboratory fume cupboards:* Part 1: 1994: *Specification for safety and performance;* Part 2: 1994: *Recommendations for exchange of information and recommendations for installation;* Part 3: 1994: *Recommendations for selection, use and maintenance;* Part 4: 1994: *Method for the determination of the containment value of a laboratory fume cupboard* (London: British Standards Institution) (dates as indicated)

90 BS 5726: *Microbiological safety cabinets:* Part 2: 1992: *Recommendations for information to be exchanged between purchaser, vendor and installer and recommendations for installation;* Part 4: 1992: *Recommendations for selection, use and maintenance* (London: British Standards Institution) (dates as indicated)

91 *Categorisation of biological agents according to hazard and categories of containment* (4th edn.) (London: HSE Books/The Stationery Office) (1995)

92 *Combustion systems* CIBSE Guide B13 (London: Chartered Institution of Building Services Engineers) (1986)

93 The Gas Safety (Installation and Use) Regulations 1998 Statutory Instrument 1998 No. 2451 (London: The Stationery Office)(1998)

94 *IGasE Utilization Procedures* (various titles) (London: Institution of Gas Engineers) (various dates)

95 BS EN 50091: *Specification for uninterruptible power supplies (UPS):* Part 1: 1993: *General and safety requirements* (London: British Standards Institution) (1993)

96 The Water Supply (Water Fittings) Regulations 1999 Statutory Instrument 1999 No 1148 (London: The Stationery Office) (1999)

97 The Water Supply (Water Fittings) (Amendment) Regulations 1999 Statutory Instrument 1999 No 1504 (London: The Stationery Office) (1999)

98 *Minimising the risk of Legionnaires' disease* CIBSE TM13 (London: Chartered Institution of Building Services Engineers) (2000)

99 *The control of legionellosis including Legionnaires' disease* HS(G)70 (The Stationery Office) (1993)

100 *Transportation systems in buildings* CIBSE Guide D (London: Chartered Institution of Building Services Engineers) (2000)

101 *Guidelines for environmental design in schools* Building Bulletin 87 (London: Department for Education and Employment/The Stationery Office) (1997)

102 The Education (School Premises) Regulations 1996 Statutory Instrument 1996 No. 360 (London: The Stationery Office) (1996)

103 *Energy efficient design of new buildings and extensions for schools and colleges* EEBPP Good Practice Guide GPG 173 (London: Department of Transport, Local Government and the Regions) (1997) (available from http://www.energy-efficiency.gov.uk)

104 *Passive solar schools — a design guide* DFEE Building Bulletin 79 (London: Department for Education and Employment/The Stationery Office) (1994)

105 Workplace (Health, Safety and Welfare) Regulations 1992 Statutory Instrument 1992 No. 3004 (London: The Stationery Office)(1992)

106 *Workplace (Health, Safety and Welfare) Regulations 1992: Guidance for the education sector* (Bootle: Health and Safety Executive) (1995)

107 *Fume cupboards in schools* DfEE Building Bulletin 88 (London: Department for Education and Employment/The Stationery Office) (1998)

108 *Saving energy in schools swimming pools — a guide to refurbishment and new pool design for head teachers, governors and local authorities* EEBPP Good Practice Guide GPG 56 (London: Department of Transport, Local Government and the Regions) (2000) (available from http://www.energy-efficiency.gov.uk)

109 *Schools' Environmental Assessment Method (SEAM)* DFEE Building Bulletin 83 (London: Department for Education and Employment/The Stationery Office) (1996)

110 *Cost effective low energy buildings in further and higher education* EEBPP Good Practice Guide GPG 207 (London: Department of Transport, Local Government and the Regions) (1997) (http://www.energy-efficiency.gov.uk)

111 *The Queens Building, De Montfort University Anglia —feedback for designers and clients* Probe NPFR 102 (London: Department of Transport, Local Government and the Regions) (1997) (included on CIBSE Members' CD-ROM)

112 *The Elizabeth Fry Building, University of East Anglia —feedback for designers and clients* Probe NPFR 106 (London: Department of Transport, Local Government and the Regions) (1998) (included on CIBSE Members' CD-ROM)

113 *Passive refurbishment at the Open University — Achieving staff comfort through improved natural ventilation* EEBPP GIR 48 (London: Department of Transport, Local Government and the Regions) (1999) (available from http://www.energy-efficiency.gov.uk)

114 *Design methodologies for smoke and heat exhaust ventilation within atria* BRE Report BR 368 (Garston: Building Research Establishment) (1994)

115 BS 5588: *Fire precautions in the design, construction and use of buildings:* Part 9: 1990: *Code of practice for ventilation and air conditioning ductwork;* Part 10: 1991: *Code of practice for shopping complexes;* Part 11: 1999: *Code of practice for shops, offices, industrial, storage and other similar buildings* (London: British Standards Institution) (dates as indicated)

116 *Drawing a winner — Energy efficient design of sports centres* EEBPP Good Practice Guide GPG 211 (London: Department of Transport, Local Government and the Regions) (1998) (available from http://www.energy-efficiency.gov.uk)

117 *Handbook of sports and recreational building design Vol. 2: Indoor sports* (London: Sport England) (1995)

118 *Badminton* Sports Council Guidance Note 357 (London: Sport England) (1995)

119 *Energy efficiency in swimming pools for centre managers and operators* EEBPP Good Practice Guide GPG 219 (London: Department of Transport, Local Government and the Regions) (2000) (available from http://www.energy-efficiency.gov.uk)

120 *Energy efficiency in sports and recreation buildings — technology overview* EEBPP Good Practice Guide GPG 144 (London: Department of Transport, Local Government and the Regions) (1996) (available from http://www.energy-efficiency.gov.uk)

121 *Technical Standards for compliance with the Building Standards (Scotland) Regulations 1990* (London: The Stationery Office) (1999)

122 *Design manual for roads and bridges — Design of road tunnels* BD78 (London: Highways Agency/The Stationery Office) (1999)

123 *Enclosed vehicular facilities* Chapter 12 in ASHRAE Handbook: *HVAC Applications* (Atlanta GA: American Society of Heating, Refrigerating and Air Conditioning Engineers) (1999)

124 Noton N H *Farm buildings* (Reading: College of Estate Management) (1982)

125 Sainsbury D and Sainsbury P *Livestock health and housing* (Reading: Balliere Tindall) (1979)

126 Welfare of Farm Animals (England) Regulations Statutory Instrument 2000 No. 1870 (London: The Stationery Office) (2000)

127 Clough G and Gamble M *Laboratory animal houses — a guide to the design and planning of animal facilities* (Manual Series No. 4) (London: Medical Research Council) (1979)

128 Firman J E Heating and ventilation of laboratory animal accommodation *J. of the Inst. of Animal Technicians* (1966)

129 McSheeny T (ed.) *Control of animal house environment* Laboratory Animal Handbook 7 (London: Laboratory Animals) (1976)

130 Brookes J *The demand for United Kingdom call centres continues* (London: G V A Grimley) (2000) (available from http://www.gvagrimley-callcentres.co.uk)

131 *Chase E Call centres — long term customer solution or late 90s flash in the pan?* (London: Information Facilities Management) (1999) (available from http://www.i-fm.net)

132 *Court standards and design guide* LCD internal document (London: Lord Chancellor's Department) (2000)

133 *A design guide for naturally ventilated courtrooms* (London: Lord Chancellor's Department) (1997)

134 *Net effect — a report on the impact of network computers on trading room design* (London: Pringle Brandon) (1998)

135 *20/20 vision — a report on the impact of flat panel displays on trading room design* (London: Pringle Brandon) (1996)

136 Livingstone F C Heating and ventilation in the service of horticulture *H & V Engineer* 43 (515) 638-644 (June 1970)

4 Systems

4.1 Introduction

Section **4** provides details of the strategies and systems available to deliver the clients' requirements as initially considered in section 2 and defined in section 3. Section **4.2** addresses the fundamentals of room air diffusion and distribution. Sections **4.3** to **4.6** consider the successful application of the chosen ventilation strategy. Sections 4.7 to 4.22 provide design guidelines for the various HVAC systems that can be utilised to achieve this objective. (Details of the constituent items of equipment within HVAC systems are given in section 5.)

The range of systems considered has been expanded significantly to reflect developments in low energy cooling techniques since the previous edition of CIBSE Guide B was published.

The treatment of low energy cooling techniques is not limited to guidance on natural ventilation Strategies, but also extends to elements of mechanical systems design.

It is not intended to provide step-by-step design guidance, but to summarise the 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 Guides A[1] and F[2]. For details of refrigeration methods, see CIBSE Guide B13[3].

4.2 Room air distribution strategies

4.2.1 Room air diffusion — criteria for design

Air diffusion is the main interface between the system and the occupants. If the air diffusion is not well designed the system will fail, no matter how accurately building loads have been modelled and how carefully the plant and equipment have been selected.

The effectiveness of all ventilation and 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
- purity of the supply air
- 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.

As discussed later, 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, but conversely particular room air diffusion requirements should 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)
- the temperature of the air stream in relation to the temperature of the still air adjacent to other parts of the body
- the level of activity taking place
- the occupants' clothing
- the purity of air in the breathing zone
- the individual's susceptibility and acclimatisation
- the appearance and positioning of any ventilation devices or openings
- the noise emitted.

The above are discussed in detail in section 1.4 of CIBSE Guide A[1].

ISO 7730[4] recommends that, during cooling, the mean air velocity should be less than 0.25 $m.s^{-1}$ for moderate thermal environments with light, mainly sedentary, activity and that, in winter, it should be less than 0.15 mv'. 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.

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, the occupied zone is any region where the occupants are likely to linger for significant periods. In the case of desk terminals, this definition does not apply. For desk terminals, mixing occurs over the desk surface and, for seatback terminals, mixing occurs in the regions above and between the seats.

An assessment of predicted percentage dissatisfied (PPD)[4] 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.s^{-1} can be acceptable in offsetting high temperatures. This technique has been applied to the concept of spot cooling in some industrial applications[5] whereby heat stress in the workers is avoided by keeping the local conditions below an agreed value of wet bulb globe temperature.

4.2.2 Ventilation efficiency[6,7,8]

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 **4.2.5).** 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.

4.2.3 Air distribution[9]

Air can be supplied to a space in a number of ways, the principal division being between diffusers and perpendicular jets. Airflow patterns for both types of terminal are strongly dependent upon the presence or absence of the Coanda effect (see section **4.2.3.5).**

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 other types of equipment are considered in section **5.13.** Further guidance can be obtained from HEVAC's *Guide to air distribution technology for the internal environment*[9].

4.2.3.1 Air terminal phenomena[10-19]

Many studies of jets and their effect on room air movement have been undertaken. Figure **4.1** shows the predicted airflow patterns for various types and positions of air terminal device[20].

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- In many 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).

4.2.3.2 Air diffusion terminology

ISO 3258[21] gives 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.s^{-1} isovel.

Normally lower velocities are required for air entering the occupied zone, typically **0.25** m.s^{-1} for cooling, **0.15** m.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.s^{-1} isovel. Note that most manufacturers give the width of the **0.25** m.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.s^{-1} isovel.

4.2.3.3 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.

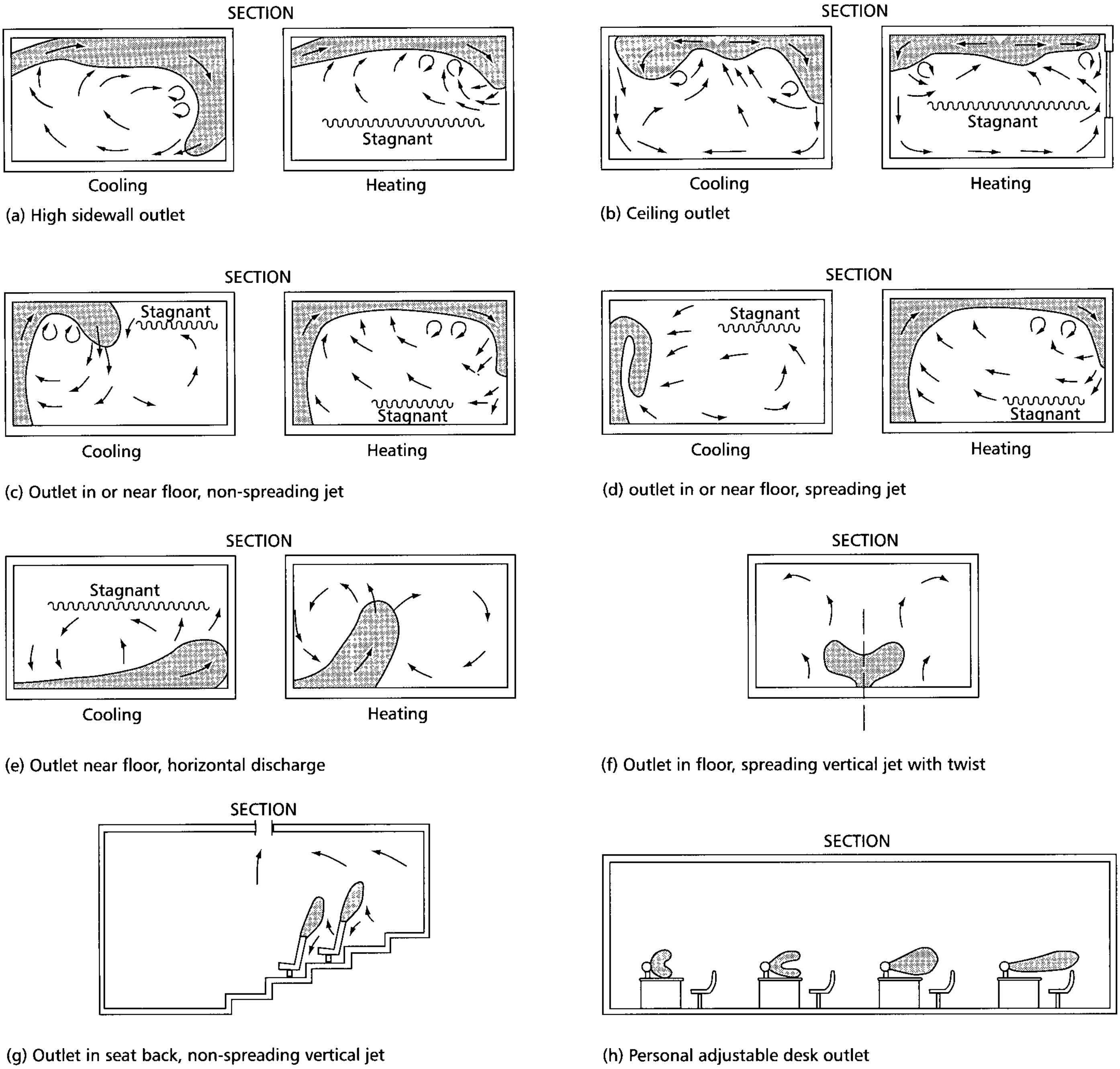


Figure 4.1 Predicted airflow patterns (reproduced from ASHRAE Handbook: *Fundamentals*, by permission of the American Society of Heating, Refrigerating and Air-Conditioning Engineers)

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[9,20].

4.2.3.4 Effect of temperature differential

Figure 4.1 shows that a jet which is not influenced by the proximity of a solid surface follows a path which 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.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:

$$q_s = m\, C_{ph}\, \Delta T \qquad (4.1)$$

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 of the air and water vapour mixture (kJ.kg^{-1}.K^{-1}) and T 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.

For comfort applications, air change rates are unlikely to exceed 10 ACH, 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 either:

- 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 4.1 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 4.1 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[22], see Table 4.2.

Table 4.2 Typical cooling temperature differentials for various applications

Application	Maximum temperature differential / K
High ceiling (large heat gains/low level input)	12
Low ceiling (air handling luminaires/low level input)	10
Low ceiling (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 (see below). 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, see section 4.21.

Effect of room convection currents

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 cultivates 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 4.2. An exception is the case of laminar downflow cleanrooms[23,24] where an even velocity across the full area of 0.4 $m.s^{-1}$ should be maintained from ceiling to floor. However, even in these circumstances, sources of extremely buoyant upflow should be avoided.

4.2.3.5 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.

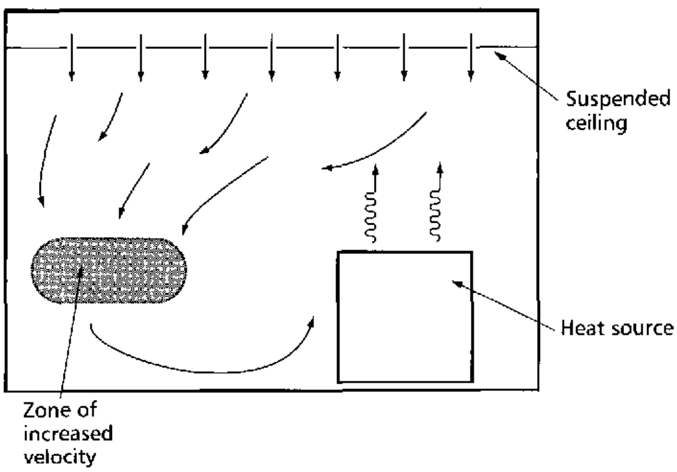


Figure 4.2 Effect of room convection currents

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 which 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%. 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 importance of these influences for side-wall terminals with various aspect ratios, velocities and temperatures differences is discussed elsewhere[17]. The most important factor 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 5.7). Cleanliness of the exhaust air is difficult to control and some staining of surfaces near to exhaust openings is inevitable.

Effect of projections and junctions

Techniques exist[15] 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 4.3 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. A design procedure for selecting optimum supply velocities and temperature is given elsewhere[18].

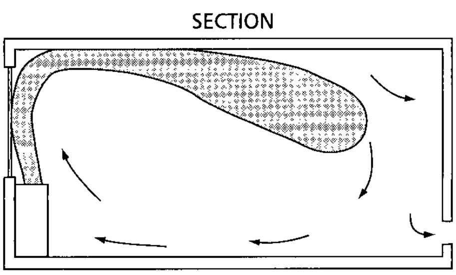
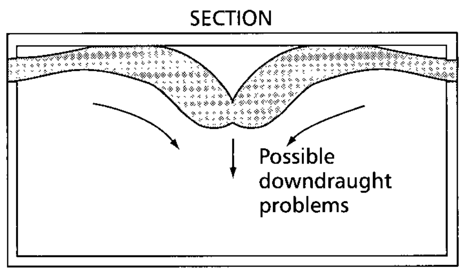


Figure 4.3 Effect of a horizontal surface on a jet

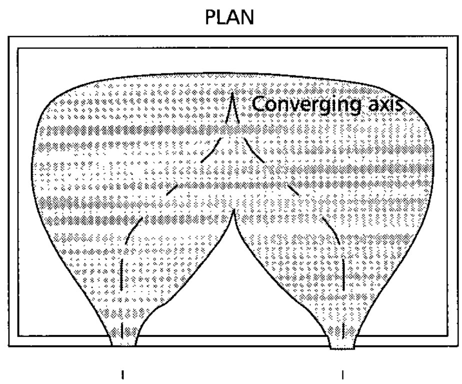
4.2.3.6 Interaction between jets

Figure 4.4(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 air streams must not be greater than the 0.25 $m.s^{-1}$ at the boundary otherwise discomfort may occur due to excessive down-draughts.

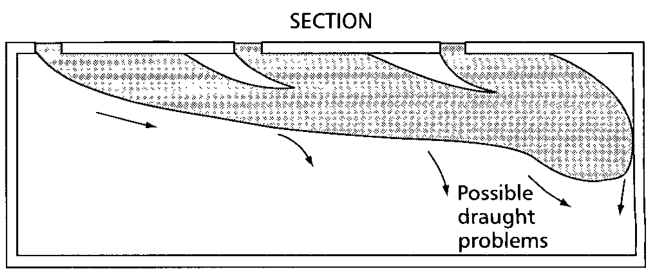
The envelopes of two converging jets may also interfere with each other, combining to form a single, wide jet with a maximum velocity at the new axis between the two jets, see Figure 4.4(b). A similar phenomenon occurs with two jets moving in tandem, see Figure 4.4(c). The downstream jet entrains and 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.



(a) Opposing jets



(b) Converging jets

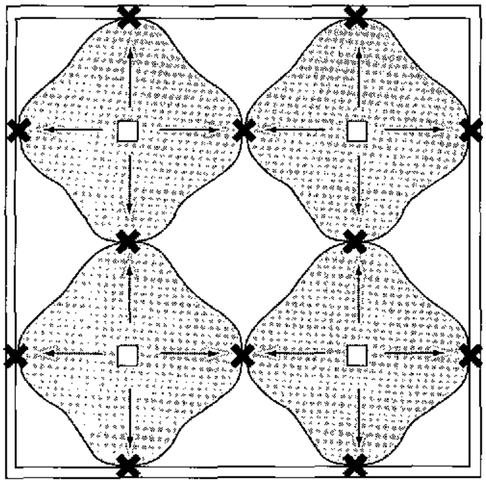


(c) Three jets in series

Figure 4.4 Room air velocity patterns; interactions between jets

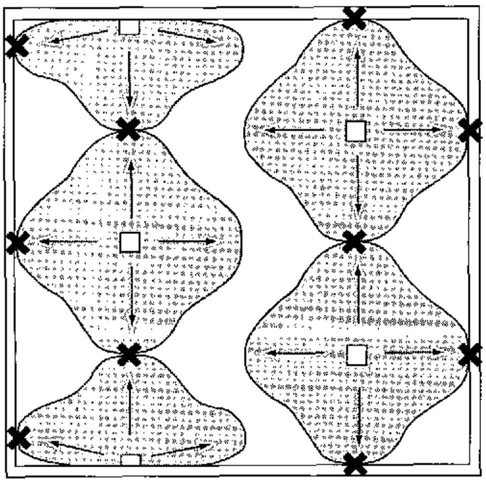
Figure 4.5 shows examples of possible layouts for ceiling diffusers. The main problems likely to be encountered are those described above. Down-draughts may been encountered in areas marked 'X' and this problem may be eliminated by avoiding terminals with excessive throw, particularly in large spaces where stagnation between terminals is unlikely to occur. The layout shown in Figure 4.5(c) may cause convergence problems with long rooms.

PLAN



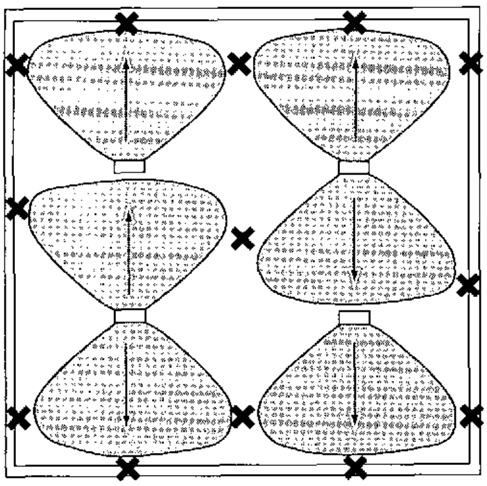
(a) Four-way ceiling diffusers, symmetrical layout

PLAN



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

PLAN



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

Figure 4.5 Supply terminal layouts for open plan spaces

For side-wall applications, the spacing of diffusers should be in accordance with manufacturers' recommendations. However, in the absence of such recommendations, Table 4.3 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 centreline to the wall. Table 4.4[22] indicates typical turndown limits for various types of fixed air terminal device.

Table 4.3 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 in metres where axial velocity has decayed to 0.25 m·s^{-1}; L_y = throw in metres where axial velocity has decayed to 0.5 m·s^{-1}

Table 4.4 Turndown limits for various types of fixed air terminal device[22]

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

4.2.3.7 Exhaust terminals

The positioning of the opening has little influence on the airflow pattern in the space because the zone of localised high velocities associated with exhaust openings is very close to the opening, see section 3.15.

Exhaust terminals may be sited to advantage as follows:

- in a stagnant zone where supply jet influence is limited
- close to a source of unwanted heat and/or contamination, e.g. above a luminaire
- close to an excessively cold surface to increase its surface temperature and thereby reduce radiant losses and cold draughts
- at a point of local low pressure, e.g. the centre of a ceiling diffuser.

The following positions should be avoided:

- within the zone of influence of a supply air terminal since this allows conditioned air to pass directly to exhaust without first having exchanged heat with its surroundings; this results in very low ventilation efficiency
- close to a door or aperture which is frequently opened since this leads to the exhaust handling air from outside the conditioned space
- in a position which causes contaminated room air to be drawn through the occupants' breathing zone.

4.2.4 Duct and plenum design

Air terminal devices will only perform as intended if the approach velocity is even. 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 is difficult or impossible.

Design procedures for duct and plenum connections to various types of air terminal are given elsewhere[9]; also see section 4.4.

If the ceiling 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.

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.

4.2.5 Displacement ventilation[25,26]

In buoyancy-driven displacement-flow ventilation 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 ventilation 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 ventilation and pollutant discharge, the air quality in the occupied zone of a room with displacement ventilation can be higher than that using a mixed-flow ventilation method. In displacement ventilation, 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 displacement ventilation, a vertical temperature gradient is unavoidable. ISO 7730[4] recommends a vertical temperature gradient for sedentary occupants of less than 3 K. This equates to approximately 3 3 K.m^{-1} if workers are assumed to be seated, although a limit of 1.8 or 2 K.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% 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 should not be lower than 18 °C for sedentary occupancy and 16 °C for more active occupancy. 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.s^{-1} may cause discomfort due to cold feet. Occupants should be located a sufficient distance from diffusers to avoid such discomfort. 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 displacement ventilation is therefore limited to 25 W.m^{-2} due to discomfort considerations[27].

Displacement ventilation can be employed for many applications and building types. It is often 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 ventilation strategies. These include[28]:

- where the supply air is warmer than the room air (except under particular circumstances where cold downdraughts exists over the supply position)
- where contaminants are cold and/or more dense than the surrounding air
- where surface temperatures of heat sources are low, e.g. <35 °C

- where ceiling heights are low, i.e. <2.3 m (the preferred height is not less than 3 m)
- where disturbance to room airflows is unusually strong.

4.2.5.1 Displacement ventilation devices[26,28]

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 to maximise the use of outdoor air for free cooling and this may eliminate the need for mechanical cooling (e.g. through groundwater).

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 appear to be less disruptive but 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.

4.2.5.2 Control of displacement ventilation systems[28]

The main forms of control are:

- *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.
- *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.

Temperature sensor location

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 location free from significant draughts.

4.3 Natural ventilation systems design

4.3.1 General

Natural ventilation is the airflow through a building resulting from the provision of specified routes such as openable windows, ventilators, ducts, shafts, etc, driven by wind and density differences. Natural ventilation may be used to provide:

- outside air for ventilation purposes
- cooling for thermal comfort.

Natural ventilation is considered in detail in CIBSE **AM10:** *Natural ventilation in non-domestic buildings*[29].

4.3.2 Strategy

There are a number of strategies that can be adopted. The basic forms are outlined in this section and illustrated schematically in Figure 4.6. The pattern of airflow through the whole building should be considered for all operational regimes - winter and summer, as well as night ventilation, if required. Ventilation strategy should be considered on the basis of the whole building rather than just room-by-room. Circulation areas such as stairwells or corridors can be used as plenums or supply ducts, although care must be taken to avoid these routes acting as 'short circuits'. Consideration should be given to where the fresh air will be brought from, e.g. it may be beneficial to draw the air from one side of the building to:

- avoid noise and traffic fumes from a busy road
- draw cooler air from a shaded side of the building to maximise the cooling.

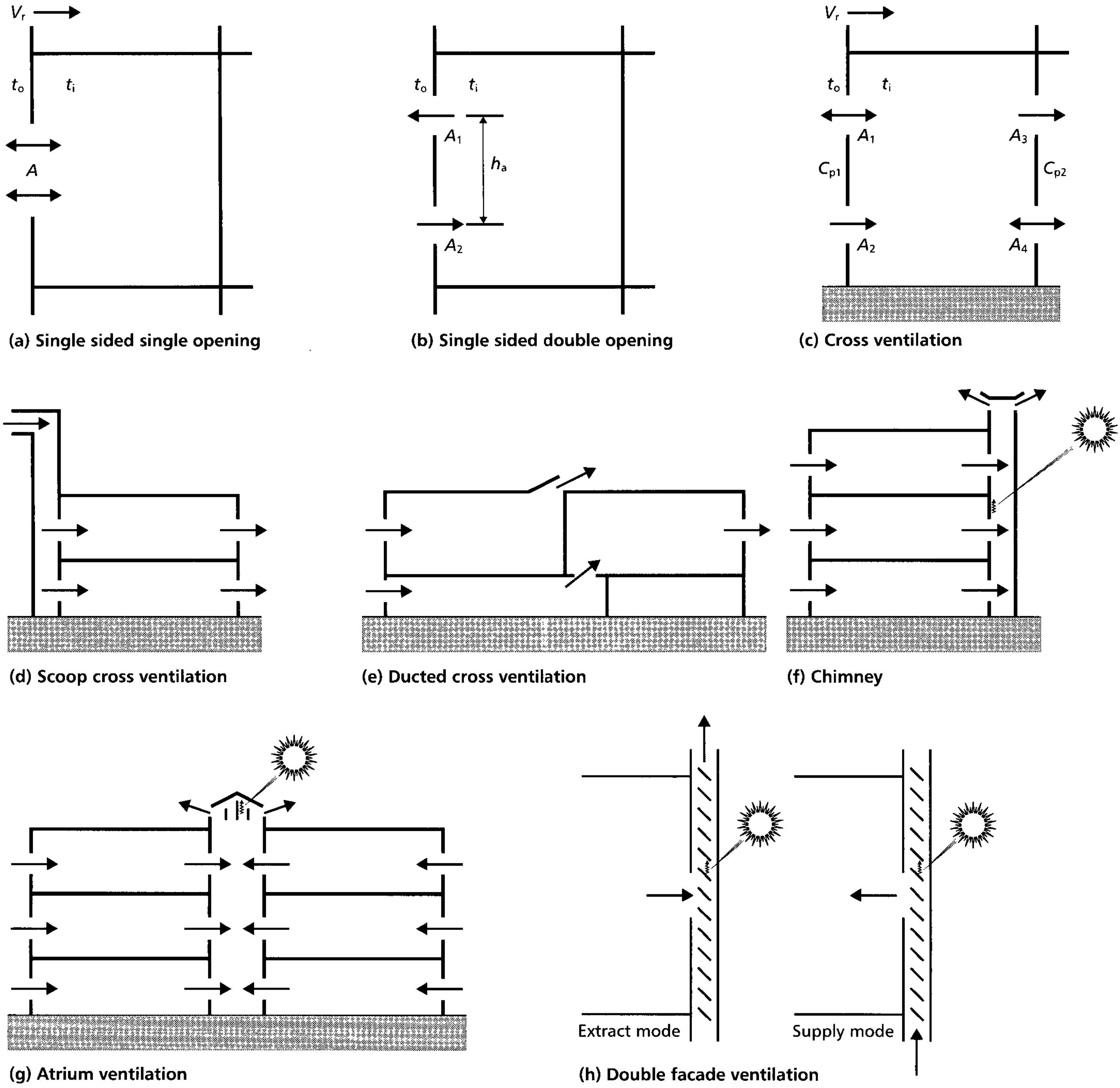


Figure 4.6 Ventilation strategy options

The magnitude and pattern of natural air movement through a building depends on the strength and direction of the natural driving forces and the resistance of the flow path. The driving forces for natural ventilation are wind and density difference.

Wind

Wind driven ventilation, see Figure 4.7, is caused by varying surface pressures acting across the external building envelope. The distribution of pressure depends on:

- the terrain
- local obstructions
- the wind speed and its direction relative to the building
- the shape of the building.

Air will flow through the building from areas of high surface pressure to areas of low pressure. In very general terms, building surfaces facing into the wind will experience positive pressures; leeward surfaces and those at right angles to the wind direction will experience negative pressure (suction). As wind velocity increases with height and the wind pressure increases as the square of wind speed, high rise buildings can experience very large wind pressures.

Density differences (buoyancy)

Warm air is lighter than cold air. If two columns of air at different temperatures are separated by a dividing boundary, a difference in pressure will exist across that boundary due to the different pressure gradients on either side. In the normal situation, where the inside of the building is warmer than outside, the pressure difference acts inwards at the lower levels of the building and

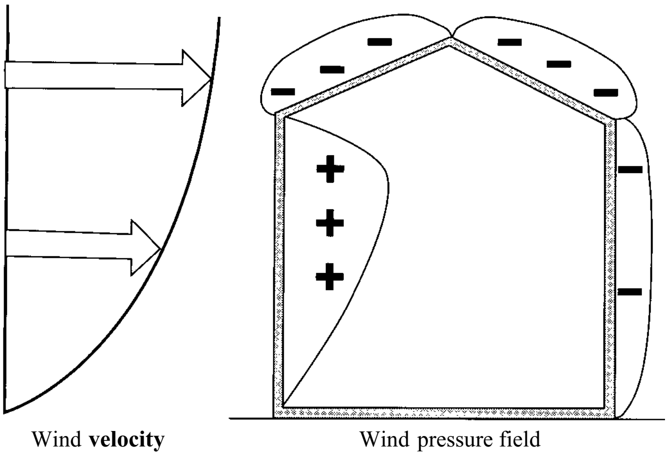


Figure 4.7 Wind driven ventilation

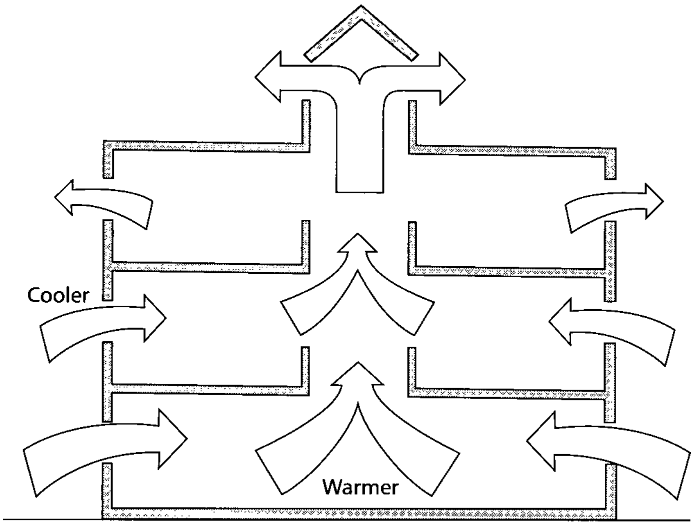


Figure 4.8 Buoyancy driven ventilation

outwards at high level. When openings are placed in the boundary separating the two air columns, an upward airflow will be created through the building, exhausting warm air at high level and replacing it by cooler air at the lower levels, see Figure 4.8. This is known as the stack effect.

Stack effects do not occur just over the whole height of the building. Stack pressures will be exerted over any vertically spaced openings that are inter-connected. For example, in a large window opening, air will tend to flow in at the bottom and out at the top.

For more detailed information on the wind and stack driving forces, refer to section 4 of Guide A(1). The basic forms of natural ventilation strategy are outlined below. The rules of thumb for estimating the effectiveness of natural ventilation given in the following paragraphs are based on section 3 of CIBSE AM10(29) and on BRE Digest 399(30). In certain situations, (primarily if heat gains are low) the limits given in these rules of thumb may be increased. The strategies described are essentially natural, but can use mechanical ventilation to supplement the ventilation during hot, still weather.

4.3.2.1 Single-sided ventilation

Single-sided ventilation relies on opening(s) on one side only of the ventilated enclosure. It is closely approximated in many cellular buildings with opening windows on one side and closed internal doors on the other side. Single-sided ventilation can be applied in offices approaching 12 m in depth, if the windows have sufficiently large openable areas. A limiting depth of 10 m is suggested as a reasonable criterion[31,32].

Single opening

For a single ventilation opening in a room, see Figure 4.6(a), the main driving force for natural ventilation in summer is normally wind turbulence. Relative to the other strategies, lower ventilation rates are generated and the ventilating air penetrates a smaller distance into the space. Single-sided single opening ventilation is effective to a depth of about 2 times the floor-to-ceiling height. The formulae for estimating the airflow rates due to the driving forces of wind and temperature difference are as follows.

(a) Wind only:

$$Q_w = 0.05\, A\, V_r \tag{4.2}$$

(b) Temperature difference only:

$$Q_s = 0.2\, A \left(\Delta t\, h\, g / (\bar{t} + 273)\right)^{0.5} \tag{4.3}$$

(c) Wind and temperature:

$$Q_t = \left(Q_w^2 + Q_s^2\right)^{0.5} \tag{4.4}$$

where Q_w is the airflow rate due to wind alone ($m^3 \cdot s^{-1}$), Q_s is the airflow rate due to temperature difference alone ($m^3 . s^{-1}$), Q_t is the total airflow rate due to temperature difference and wind ($m^3 \cdot s^{-1}$), A is the area of the opening (m^2), V_r is the wind speed at building height ($m.s^{-1}$), t is the inside–outside temperature difference (K), h is the height of the opening (m), g is the acceleration due to gravity ($m.s^2$) and t is the average of inside and outside temperatures (°C).

Double opening

Where multiple ventilation openings are provided at different heights within the facade, then the ventilation rate can be enhanced due to the stack effect, see Figure 4.6(b). The ventilation rate will be further enhanced by any wind pressures that may be acting on the ventilation opening. Single-sided double opening ventilation is effective to a depth of about 2.5 times the floor-to-ceiling height.

The stack-induced flows increase with the vertical separation of the openings and with the inside to outside temperature difference. To maximise the height over which the stack pressures act, it may be necessary to separate the ventilation openings from the window itself. As well as enhancing the ventilation rate, the double opening increases the depth of penetration of the fresh air into the space. Low level inlets should be positioned to minimise the risk of ankle level draughts in cold weather.

The formulae for estimating the airflow rate due to the temperature difference are as follows:

$$Q_s = C_d\ A_w\left(2\ \Delta t\ h_a\ g\,/\,(\bar{t}+273)\right)^{0.5} \tag{4.5}$$

$$\frac{1}{A_w^2} = \frac{1}{A_1^2} + \frac{1}{A_2^2} \tag{4.6}$$

where Q_s is the airflow rate due to temperature difference alone (m^3-s^{-1}), C_d is the discharge coefficient (0.61 for large openings), A_w is the effective area of the combined openings (m^2), A_1 and A_2 are the areas of the upper and lower openings respectively *(m^2),* t is the inside-outside temperature difference (K), h_a is the vertical distance between centres of the openings (m), g is the acceleration due to gravity ($m.s^{-2}$) and t is the average of inside and outside temperatures (°C).

4.3.2.2 Cross ventilation

Cross ventilation occurs where there are ventilation openings on both sides of the space concerned, see Figure 4.6(c), and is usually wind driven. As the air moves across the zone, there will be an increase in temperature and a reduction in air quality as the air picks up heat and pollutants from the occupied space. Consequently there is a limit on the depth of space that can be effectively cross-ventilated. This implies a narrow plan depth for the building, which has the added benefit of enhancing the potential for natural lighting. Cross ventilation is effective up to 5 times the floor-to-ceiling height.

The formulae for estimating the airflow rate are as follows:

(a) Wind only:

$$Q_w = C_d\ A_w\ V_r\ \Delta C_p^{0.5} \tag{4.7}$$

where:

$$\frac{1}{A_w^2} = \frac{1}{\left(A_1+A_2\right)^2} + \frac{1}{\left(A_3+A_4\right)^2} \tag{4.8}$$

(b) Temperature only:

$$Q_s = C_d\ A_b\left(2\ \Delta t\ h_a\ g\,/\,(\bar{t}+273)\right)^{0.5} \tag{4.9}$$

where:

$$\frac{1}{A_b^2} = \frac{1}{\left(A_1+A_3\right)^2} + \frac{1}{\left(A_2+A_4\right)^2} \tag{4.10}$$

(c) Wind and temperature:

For $Q_w > Q_s$:

$$Q_t = Q_w \tag{4.11}$$

For $Q_w < Q_s$:

$$Q_t = Q_s \tag{4.12}$$

where Q_w is the airflow rate due to wind alone ($m^3.s^{-1}$), Q_s is the airflow rate due to temperature difference alone ($m^3.s^{-1}$), Q_t is the total airflow rate due to temperature difference and wind ($m^3.s^{-1}$), C_d is the discharge coefficient (0.61 for large openings), V_r is the wind speed at building height ($m.s^{-1}$), C_p is the difference in pressure coefficient between inlet and outlet, A_w and A_b are the effective areas of the combined openings (m^2), A_1 and A_2 are the areas of the upper and lower openings respectively on the windward side of the building (m^2), A_3 and A_4 are the areas of the upper and lower openings respectively on the leeward side of the building (m^2).

Ideally the form of the building should be such that there is a significant difference in wind pressure coefficient between the inlet and outlet openings. Consideration should also be given to the resistance to airflow. Insufficient flow may be generated, particularly in summer conditions, if openings on one side of the building are closed, or if internal partitions (particularly full height ones) restrict the flow of air across the space. In such situations, the ventilation mechanism will revert to single sided.

In order to improve air distribution into deeper spaces, it is possible to use ducted or underfloor ventilation pathways. This can provide ventilation to internal spaces or a perimeter zone local to a pollution source (e.g. a busy road). Because of the low driving pressures with natural ventilation (<10 Pa), it is important to design the supply duct for very low pressure drops.

The normal approach to cross ventilation is via opening windows, but other approaches have been used with success, particularly in hot desert countries. One example of this approach is the wind scoop.

Windscoops

Wind scoops, see Figure 4.6(d), capture the wind at high level and divert it into the occupied spaces to exhaust on the leeward side. The performance of a wind scoop strategy is enhanced when there is a dominant prevailing wind direction, (e.g. at a coastal site). Where wind direction varies frequently, multiple inlets would be necessary, with automatic control to close the leeward and to open the windward ventilation openings. Since wind speed increases with height, the pressure will be greatest at the top of the structure, thereby generating a positive pressure gradient through the whole building.

When designing a wind scoop, the effect of stack pressures must be considered, since these may act in opposition to the intended direction of flow.

4.3.2.3 Stack ventilation

This term is used to describe those ventilation strategies that utilise driving forces to promote an outflow from the building, thereby drawing fresh cool air in via ventilation openings at a lower level. The approach utilises the density difference between a column of warm air and surrounding cooler air. Stack ventilation can be effective across a width of 5 times the floor-to-ceiling height from air inlet to the stack inlet.

The stack pressures are a function of the temperature difference and the height between inlet and outlet. Therefore the driving force reduces at the higher stories and this needs to be counteracted by providing increased opening areas.

The height up the building where the inflow changes to an outflow is called the 'neutral pressure level'. The position of the neutral pressure level is a function of the density difference of the two air columns and the vertical distribution of the openings. Typically the neutral pressure level is designed to be located above the top floor to avoid recirculation of stale air from the lower floors back through the upper floors. The neutral pressure level can be raised by either increasing the size of the roof vent, or by reducing the size of the openings on the lower floors.

The driving forces for stack ventilation can be enhanced by designing the stack outlet to be in a wind-induced negative pressure region.

At night, as outside temperatures drop, the temperature difference driving the stack ventilation will increase. This enhances the ventilation rates that can be achieved for night cooling using stack ventilation.

If a building makes extensive use of passive cooling by thermal mass, then on the hottest days room temperatures may be below the outside temperature, potentially producing a negative buoyancy effect in the stack. This will reduce ventilation rates, a beneficial effect for thermal comfort if the outside temperature is above the room temperature.

By its nature, the ventilation strategy is essentially cross ventilation, as far as the individual occupied zones are concerned, in that air enters one side of the space and exits via the opposite side. The air may flow across the whole width of the building and exhaust via a chimney, or it may flow from the edges to the middle to be exhausted via a central chimney or atrium.

Chimney ventilation

Chimneys provide a means of generating stack driven ventilation, see Figure 4.6(f). The essential requirement is for the air in the chimney to be warmer than the ambient air. If the chimney has a large surface area exposed to the prevailing weather, this should be well insulated.

Where chimneys provide no functional purpose other than ventilation, they may be sized just to satisfy the pressure drop requirements. Chimneys can also act as light wells, solar collectors, architectural features, locations for weather stations, and (historically) as security aids/watch towers. They can be in the form of a single linear chimney or several smaller chimneys distributed around the building. If the building faces onto a busy road, it would be possible to place the inlets on the facade away from the noise and pollution source with the chimneys on the road side.

It is possible to enhance the stack pressures by means of absorbing solar gain (the so-called 'solar chimney') introduced via glazed elements. Location of the solar chimney on the sunny side of the building in order to capture the solar radiation will generally result in cooler air being drawn in from the opposite shaded side.

Care should be taken to ensure that there is a net heat gain into the chimney during cooler weather i.e. the solar gain must be greater than the conduction loss. In cold weather, the conduction heat loss will result in low surface temperature for the glass that may be sufficient to generate down draughts inhibiting the general upward flow through the chimney. The outlet should be located in a negative wind pressure zone. The wind driving pressures can be enhanced by careful design of the roof profile and/or the chimney outlet configuration.

As a means of providing adequate ventilation on very hot and still days, consideration should be given to installing extract fans in the tower to pull air through the building. The fan should not provide a significant resistance to flow when the chimney is operating in natural draught mode.

Atrium ventilation

An atrium is a variation on the chimney ventilation principle, see Figures 4.7(g) and 4.9. The essential difference is that the atrium serves many more functions than does the chimney; e.g. it provides space for circulation and social interaction. These can restrict the flexibility to locate the atrium to maximum advantage for ventilation purposes. The design of atria is discussed in detail by Saxon[33]; refer also to section 3.4.

The maximum distance from building perimeter to atrium must conform to the cross ventilation limits given earlier (i.e. 5 times the floor-to-ceiling height). With a centrally located atrium, the air can be drawn from both sides of the building, thereby doubling the plan width of the building that can be ventilated effectively by natural means. (Note that the same effect could be achieved by a central spine of chimneys.)

The atrium also provides an opportunity for introducing daylight into the centre of a deep plan building. Because atria are designed to capture natural light, they are by definition solar assisted. To promote natural ventilation, the air temperature in the atrium should be as high as possible over as great a proportion of the atrium height as possible. If the atrium is open to the surrounding space, or if it provides high level walkways between floors, then excess temperatures at occupied levels may be unacceptable. The design should therefore seek to allow solar gain to be absorbed by surfaces such as:

- elements of the structure,
- solar baffles or blinds which act as shading devices.

As with the chimney ventilation strategy, roof vents must be carefully positioned within the form of the roof so that positive wind pressures do not act on the outlets thereby causing reverse flow. This is achieved by:

- designing the roof profile so that the opening is in a negative pressure zone for all wind angles
- using multiple vents which are automatically controlled to close on the windward side and open on the leeward side.

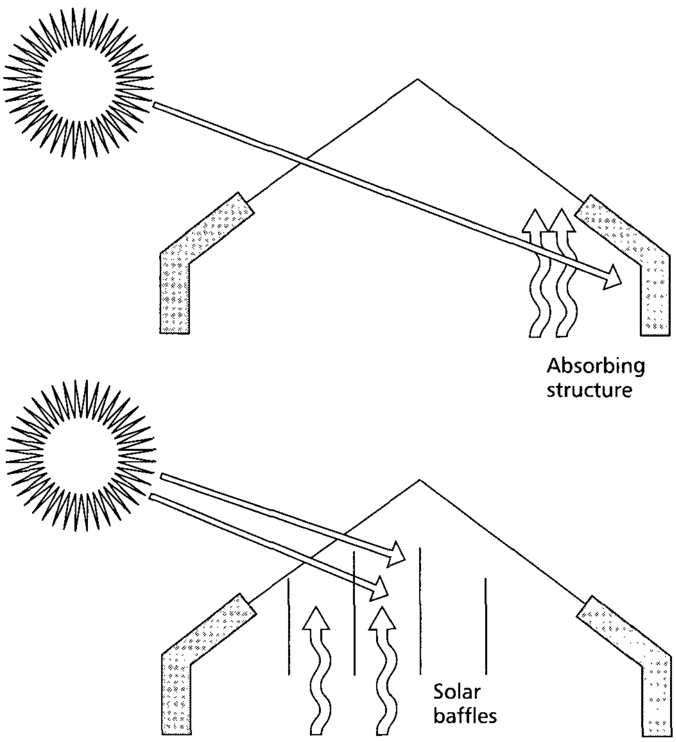


Figure 4.9 Atrium ventilation

Natural ventilation can be supplemented on hot still days by the use of extract fans in the atrium roof. Subject to approval by the fire officer, these can also form part of the smoke control system.

Facade ventilation

The double facade is a special form of solar chimney, where the whole facade acts as an air duct, see Figure 4.6(h). It can act as an extract plenum similar to a solar chimney. In order to provide absorbing surfaces to promote convective flow in the facade, cavity blinds are used. These also prevent direct gain passing through the facade to the occupied space.

Alternatively, the cavity could be used as a supply plenum. Outside air is introduced into the cavity at low level and the cavity acts as a solar collector, pre-heating the outside air. The warmed air is then supplied into the occupied zones via ventilation openings between the cavity and the space. If the air in the cavity is too hot, then it can be exhausted to outside or to a heat recovery device.

The efficiency of the solar collector mode can be significantly reduced if the conduction losses are too high. The possibility of condensation should also be checked based on the conditions of the air entering the cavity and the temperature of the glass.

4.3.3 General system issues

If external pollution levels or heat gains are high, it is unlikely that natural ventilation on its own will be able to maintain air quality and thermal comfort within acceptable limits. A prerequisite for the use of natural ventilation to provide cooling for thermal comfort is control of heat gains into the occupied space.

This section outlines general issues that should be considered when designing a natural ventilation system and the strategies available. More detailed guidance and information is contained in CIBSE Applications Manual AM10[29]. Section 4.7 provides guidance on the use of night cooling to minimise summer overheating. Details of equipment for natural ventilation are given in section 5.3.

It should be recognised that not all parts of a building need to be treated in exactly the same way. Different natural ventilation strategies may be applied to different parts of a building as appropriate. Natural ventilation can also be combined with mechanical ventilation (and/or air conditioning) to those parts of a building with particular environmental requirements in a 'mixed mode' system. Reference should be made to section 4.5 for guidance on mixed mode systems.

The following sections outline general issues that should be taken into account during the selection and development of a natural ventilation strategy or strategies.

4.3.3.1 Building form and fabric

The interaction between building form and ventilation strategy is outlined in section 2.3. Natural ventilation relies on the building envelope (rather than any mechanical system) to provide the primary environmental control. The building form will need to facilitate the airflow strategy. Particular consideration should be given to the following:

- building spacing and orientation and their impact on building shading and wind effects
- plan width/floor-to-ceiling height ratio to achieve effective ventilation; as airflows across the zone sufficient height is required for stratification to lift heat and contaminants above the downstream occupied space
- good solar control by sensible choice of glazing ratios and by shading provision (although a balance must be achieved in allowing appropriate natural lighting levels); buildings with their main facades facing north and south are much easier to protect from excessive solar gain in summer
- openings in the external facades to provide airflow paths
- thermal capacity (exposed soffits etc.) to absorb heat gains; refer to section 4.7 for design details
- airtightness to minimise energy losses and cold draughts in winter and to assist the controllability of natural ventilation; refer to CIBSE TM 23[34] for airtightness targets.

4.3.3.2 Thermal comfort

Section 1 of CIBSE Guide A[1] should be referred to for detailed guidance on thermal comfort. Natural ventilation should for most of the year be able to maintain temperatures within the control bands given in section 1 of Guide A for air conditioned buildings. However, temperatures will inevitably rise during peak summer conditions. Natural ventilation is therefore suited to buildings where an increase in peak summer temperatures is permissible (see section 3.2.2.1 for temperature requirements).

To reduce any overheating, it is essential that the level of both internal and climate induced gains are minimised. Night cooling (ventilation of the building at night when ambient temperatures are lower) is often used to limit temperature rise. The air cools the fabric of the building and the stored cooling is then available the next day to offset heat gains. The thermal capacity of buildings may be increased (commonly by exposing soffits) to increase the amount of cooling that may be stored. Refer to section 4.7 for further guidance and design details.

Although natural ventilation cannot offer control of space humidity, the relative humidity in non-air conditioned buildings will not exceed 70% unless there is a very high level of internal moisture generation. Low internal humidity can be caused by excessive infiltration/ventilation in very cold weather.

The ability to open windows to provide increased air movement can provide a beneficial cooling effect in summer (see CIBSE Guide A[1] section 1). However, the high ventilation flow rates associated with summer conditions can cause nuisance draughts that may disturb papers etc. This can be reduced by specifying that the openable part of the window to be above desk height.

The minimisation of draughts is a particular issue for natural ventilation systems in winter. The potential problem can be reduced in a number of ways:

- provide multiple trickle ventilators (or similar)
- use specially provided ventilation openings positioned so that the air is warmed before reaching the occupied space (e.g. behind a radiator, or in a floor void with a suitable convector heater)
- use a separate mechanical ventilation system which can pre-heat the air.

4.3.3.3 Air quality

Mechanical ventilation and air conditioning systems can filter the incoming air to remove dust and dirt. Gaseous pollutants can be minimised by careful siting of ventilation inlets. However, in a naturally ventilated building, there is usually one inlet per room outlet and the inlets are more evenly distributed over the building facade (both horizontally and vertically). It is therefore more difficult to locate all the inlets away from sources of pollution.

Consideration should be given to source control measures to minimise the internal pollutant load, including elimination of sources of pollution (e.g. by choosing alternative materials) or, if this is impractical, locating the pollution source (e.g. a photocopier) near to a ventilation extract.

Where air is passed from one zone to another the flow of fresh air should be sufficient to provide acceptable air quality in the downstream zone.

The fresh air rate is normally specified in terms of a constant flow rate. Natural ventilation cannot provide a constant flow rate but the important parameter is the time-averaged flow, rather than the instantaneous, flow rate. This means that, within reason, the fresh air rate can vary and there will not be any significant variation in indoor air quality because of the reservoir provided by the volume of the space.

Refer to CIBSE TM21[35] for guidance on the nature and characteristics of pollutants in the outdoor air and their impact on indoor air quality.

4.3.3.4 Heating

In winter, any fresh air over and above that required for controlling indoor air quality represents an energy penalty. If the ventilation is to be provided by opening windows, then the windows should be capable of being well sealed when closed to minimise energy loss due to infiltration.

There is usually a significant difference in the required airflows in summer and winter and precise control of ventilation flows is difficult to achieve with an opening window. Separate ventilation openings, such as trickle ventilators, may be installed to provide the winter ventilation requirement.

The interaction between the ventilation and heating system should also be considered. If an area of the building gets too warm (e.g. due to solar gain through a window), the instinctive reaction of the occupant is likely to be to open the window rather than to turn down the heating. Measures to reduce conflict include:

- localised controls such as thermostatic regulating valves
- interlocks between the heating system and opening windows
- compensated variable temperature heating circuits.

4.3.3.5 Acoustics

External noise should not normally present a significant problem unless opening windows face onto busy main roads or are within 100 m of a railway line. A partially open window typically has a weighted sound reduction index of 10-15 dB compared to 35-40 dB for thermally insulating double glazing[36]. Measures to improve acoustic performance include:

- the use of acoustic baffles
- siting the opening windows on a quiet side of a building
- use of acoustic ventilators (as opposed to windows)
- placing buffer zones (e.g. a circulation space) adjacent to the noise source.

Discomfort can be caused by too little background noise as well as by high noise levels. Background noise levels should generally achieve a reasonable compromise between audibility and privacy. External noise can provide a beneficial masking effect for indoor acoustic privacy.

As well as the ingress of external noise, consideration also needs to be given to internal acoustic design issues including:

- conflict between partitioning for acoustic privacy and provision of air paths
- exposed therma1 mass increases the number of hard surfaces, see section 4.7.

Acoustic are dealt with in detail elsewhere in CIBSE Guide B5: ***Sound control***[37].

4.3.3.6 Flexibility

Flexibility should be provided to cope with changing occupant requirements over the life of the building. Systems can be designed to be capable of relatively simple upgrading (and downgrading) ***so*** that extra cooling systems can be added when and where required. Contingency planning is required at the design stage to provide:

- sufficient space for the subsequent installation of additional equipment
- an adequate floor-to-soffit height to enable additional servicing to be routed through floor or ceiling voids
- breakout floor zones that could form future service risers.

4.3.3.7 Control

The control strategy should consider all normal operational modes (e.g. winter, summer, night cooling) as well as emergency modes such as smoke control. It should also consider how the controls should 'fail-safe' in the event of power failure. Modulation of airflow is normally achieved by regulating the size of the ventilation openings in response to changing demand. This can be carried out automatically or manually.

Typical features for automatic control may include:

- CO, or occupancy sensors to control ventilation rates in heating mode
- internal temperature control of ventilation in cooling mode
- night cooling if the inside or slab temperature is high (refer to section 4.7)
- wind speed sensors to throttle back vent openings at high wind speeds, possibly in combination with rain sensors to indicate potential driving rain problems
- wind direction sensors to open vent on the leeward side
- solar gain sensors for feed forward control to increase ventilation when gains are high.

The positioning of sensors to obtain representative readings is very important. In particular, internal temperature sensors should not be too close to windows as incoming fresh air may not have mixed with the room air and the sensed condition may not therefore be representative. External temperature sensors should not be placed on sunny walls that can absorb solar radiation and elevate the sensor reading throughout the 24-hour cycle.

Automatically controlled openings could be modulated, open/shut, have intermediate fixed positions, or open in sequence where a number of vents serve a common zone. Operation should be a function of prevailing weather conditions as well as the required ventilation rate since these will influence the driving forces. Wind speed override may be required to prevent excessive ventilation under windy conditions.

Manual control is the most common form of control. It provides increased personal control over the environment in their workspace by the occupants, a factor often associated with increased occupant satisfaction. Control should be[38]:

- territorial, positioned locally and, ideally, affect a single person
- intuitive
- accessible.

Problems may arise if a single opening is required to provide ventilation for a group of occupants. This can be minimised if the window unit has high and low level openings for independent control by occupants internally and at the perimeter respectively. This may require actuators on the high level openings operated by a remote controller (that could also be used as part of an automatically controlled night cooling regime).

Intuitive manual control will not necessarily lead to windows being opened at the optimum time of day. The instinctive reaction is to open windows to increase ventilation as indoor temperatures increase later in the day, whereas higher ventilation rates may be more beneficial earlier in the day, when ambient temperatures are lower.

Refer to section 4.7 for night cooling control strategies. If night cooling is under manual control, windows will either be closed or left open for all the unoccupied hours resulting in either:

- inadequate pre-cooling, with overheating the following afternoon or
- overcooling, with cold discomfort problems the next day (or a need for heating).

These problems can be avoided by some form of automatic control of window opening or by provision of a separate mechanical night ventilation system.

4.3.3.8 Energy efficient naturally ventilated buildings

An energy efficient naturally ventilated building will provide the required levels of thermal comfort and acceptable indoor air quality under all seasonal conditions and will also meet acoustic requirements for the internal conditions with a minimum use of energy. For further guidance on the energy efficient application of natural ventilation refer to section 6 of CIBSE Guide F[2].

4.3.3.9 Heat recovery

With improving insulation standards, ventilation heat loss is becoming an increasingly important element of the energy balance, particularly given the trend to greater fresh air rates to improve indoor air quality. A high efficiency of heat recovery is difficult to achieve in a naturally ventilated building, except in very special circumstances such as the double façade 4.3.2.3. The use of high levels of thermal capacity is a way of achieving some energy recovery, since it allows heating in winter (and cooling in summer) to be stored for use at different times of the day (refer to section 4.7). The efficiency of this process is lower than air-to-air heat recovery devices in mechanical ventilation systems but the parasitic energy losses can also be much lower.

4.3.3.10 Security

If a Ventilation Strategy relies on opening windows (especially if they are left open overnight for night ventilation), particular thought needs to be given to the security implications. Movement of ventilation openings at night and entry of birds through openings can also cause problems with movement detection security systems.

4.3.3.11 Rain

The large ventilation openings that may be needed to deliver the required airflow should be designed to avoid rain entering the building, taking account of the effects of driving wind, splashing etc. Particular thought needs to be given for ventilators left unattended during night ventilation.

4.3.3.12 Fire safety

The ventilation strategy may interact with the requirements for fire and smoke control, particularly if the building needs to be subdivided into separate compartments. Ventilation routes that penetrate a fire separation are not allowed to compromise its rating. Fire rated ductwork or fire dampers may be used to maintain the separation[39]. Any ventilation openings penetrating a separation would need to be closed in the event of a fire incident using measures including:

- fire doors held open by magnetic catches that release on a fire or smoke alarm
- fire (or fire/smoke) dampers in ducted ventilation paths or transfer grilles.

Although penetrations may be accepted on partitions along a horizontal means of escape, greater concerns would be expressed for partitions surrounding vertical means of escape such as stairwells. The requirements for atrium buildings with phased evacuation are more restrictive than buildings with single stage evacuation.

The relationship between the escape routes and the normal ventilation flow path should always be considered as part of the overall strategy. For example, in a building with a central atrium, the escape routes should be toward the perimeter, always moving people in the direction of reducing smoke concentration.

Guidance on fire smoke control issues can be found elsewhere including CIBSE Guide E[40], BRE Report BR 368[41], BS 5588[39] and Approved Document B[42].

4.3.3.13 Testing and commissioning

The commissioning (setting to work) of a natural ventilation system is relatively straightforward. However, fine-tuning of the system should be carried out for at least one year after handover. Guidance on initial set-points and fine-tuning is provided in BSRIA Technical Note TN11/95[43].

Commissioning of building management systems (BMS) should be in accordance with CIBSE Commissioning Code C: *Automatic controls*[44].

4.3.3.14 Maintenance

Provision should be made to ensure that equipment associated with natural ventilation systems is accessible for maintenance. This is a particular issue for automatic vents located at high level in atria.

4.3.4 Performance assessment

Analyses will normally first need to determine the airflow rates required to meet the ventilation and/or cooling requirements and then, secondly, to size the components of the natural ventilation system to provide the airflow rates. This section outlines the basic tools available for these steps and describes a number of specific tools for more detailed analysis of issues such as air movement.

4.3.4.1 Assessment of ventilation requirements

See section 3.2.1 for required airflow rates for ventilation purposes. Airflow rates for cooling will normally be based on a summertime temperature prediction using some form of therma1 analysis. An overview of some of the assessment techniques available to determine airflow requirements is given in Appendix 4.A1. These include simple (dynamic) modelling and simulation.

4.3.4.2 Sizing components

Both explicit and implicit calculation methods are available for sizing components. Explicit equations and methods have been developed for calculation in one step. Equations relating to simple strategies and geometries have been given in sections 4.3.2.1 and 4.3.2.2. For analysis of more complex cases, reference should be made to CIBSE AM10[29]. Implicit methods use an iterative process, adjusting the component sizes until the required airflows are achieved. These range from single zone models to more complex multi-zone models (see Appendix 4.A1).

All of the calculation methods require data on component airflow characteristics and the wind and stack pressures driving the ventilation. There is a vast the range of data available on flow characteristics of components[45]. Data provided by the manufacturers is preferred for specific flow components. If these data are not available then, for large openings, the orifice flow equation given in section 4 of CIBSE Guide A[1] should be used, where the area is the openable area of the device. When considering the openable area of a window, this must be the orifice area normal to the airflow, not the facade area of the window unit that is openable. For large openings there can be two-way flow when buoyancy predominates[29].

The wind driving pressures are proportional to the velocity pressure of the wind, which, in turn, is proportional to the square of the wind speed. The factor that relates the surface pressures to the wind velocity pressure is the pressure coefficient. Reference should be made to section 4 of CIBSE Guide A[1] and other sources[45] for pressure coefficient data.

The calculation of stack driving pressures relies on the prediction of the temperature distribution through the building. The external temperature will be defined by the weather data used as the basis of design. The internal temperature for design purposes is normally taken as the air temperature specified to pertain in each of the internal spaces at the design condition.

4.3.4.3 Detailed analysis

Computational fluid dynamics (CFD) and physical models are often used for more detailed analyses of air movement and to provide visualisations of airflow behaviour (refer to Appendix 4.A1).

4.4 Mechanical ventilation systems design

4.4.1 General

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

4.4.2 Mechanical ventilation strategies

There are several possible arrangements for the supply and extraction of air in mechanical ventilation systems[46]. These are described in the following sections.

4.4.2.1 Balanced supply and extract

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. It is also weather independent. However, effective building sealing is required as the system is designed to be pressure neutral. Capital costs are high due to the expense of two separate ductwork systems and increased fan energy requirements. Regular cleaning and maintenance are also necessary.

Sometimes systems may be set up to be slightly unbalanced to maintain the building under a small negative pressure (e.g. for dwellings) or small positive pressure (e.g. for commercial buildings).

4.4.2.2 Mechanical supply and natural extract

Supply air is mechanically introduced into the building, displacing indoor air through purpose provided openings and/or infiltration. A proportion of the air can be recirculated. This is used in situations where positive pressure is required to prevent the inward leakage of air, e.g. clean rooms. It can be used to provide uniform ventilation, or can be set to provide individual airflow rates. The supply air can be treated as required, e.g. heated or filtered, the latter facility making it suitable for allergy control.

Noise may also be an issue. Air intakes must be carefully located to avoid drawing in externa1 pollutants and must not be obstructed or blocked. The removal of pollutants at source is not possible.

4.4.2.3 Mechanical extract and natural supply

A fan is used to extract air from the space and create a negative pressure that draws in an equal mass of fresh air from outside. If the under-pressure is greater than that developed by the wind and temperature differences then the system is weather independent, if not it is dominated by infiltration.

Mechanical extract can be provided on a local basis, either from industrial processes or sources of moisture e.g. bathrooms. It can also be provided by a centralised system on a whole house, or non-domestic environment where a suction pressure is desirable to prevent the egress of contaminants, e.g. chemical laboratories.

Excessive under-pressure must be avoided as it may give rise to back-draughting of combustion products, the ingress of radon or other soil gases, and noise problems. The system cannot easily be adapted to provide individual control.

In terms of delivering the air to the space this can be achieved by either displacement or mixing ventilation.

Displacement ventilation

This is based on the provision of a low-level, low-velocity air supply that is at a temperature just below that of the room. The air then rises due to buoyancy, created by heat sources within the space, to form a concentrated layer of pollutants at the ceiling from whence it is extracted. This system is considered to provide 'less polluted' air within the occupied zone and is 100% fresh air based. It is also thought to be energy efficient in that both fan power and cooling requirements are reduced. There is limited cooling capacity unless it is combined with active cooling systems such as chilled beams. Ideally, a minimum floor-to-ceiling height of 2.7 m is required. Appropriate diffusers must be selected. For further details of displacement ventilation system design refer to section 2.5.

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. The extraction of air then 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.

Floor-based supply

A floor-based supply is usually selected if raised floors are already in place 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.

Ceiling-based supply

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

4.4.3 System considerations[46]

4.4.3.1 Air handling units

The air handling unit should be located as close as possible to the ventilated space, in order to minimise the length of the ductwork run. Guidance on the sizing of plant rooms to allow the safe maintenance of mechanical ventilation plant is available elsewhere[47].

4.4.3.2 Ductwork and system velocities[48]

Ductwork should have as large a cross-sectional area as possible to produce low velocity systems and reduce system pressure drops. Figure 4.10[49] illustrates the running and capital costs for systems having different design air velocities. These figures show how the running costs are reduced for low velocity systems, and how some components become more expensive while others become cheaper. The benefits of the energy efficient (i.e. low velocity) system include a reduction in electricity costs of approximately 70%, while the additional capital cost is recovered in less than five years.

The basis of the comparison is as follows:

- all systems supplying 2 $m^3.s^{-1}$ of air
- all systems supplied by a centrifugal fan operating at an efficiency of 70%
- pulley and motor efficiencies of 90% and 80%, respectively
- electricity cost: 5 pence per kW.h
- annual run time: 3000 hours
- noise levels less than 40 dBA.

In a low velocity system, the air handling unit face velocity would typically be less than 2 $m.s^{-1}$ with the main duct velocity less than 3 $m.s^{-1}$. In a medium velocity system these figures would become 2-3 $m.s^{-1}$ and 5 $m.s^{-1}$ respectively. In a high velocity system the air handling

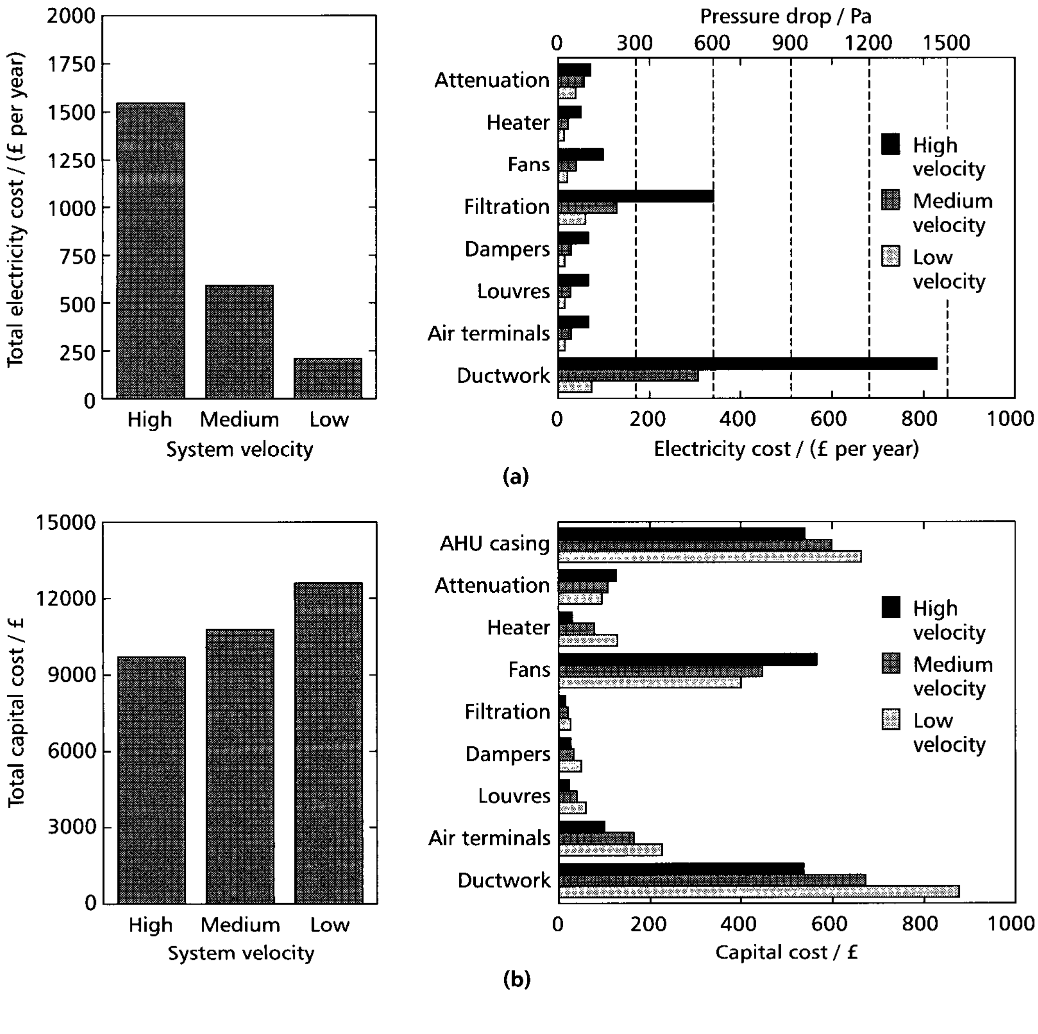


Figure 4.10 Comparison of high, medium and low velocity systems[49]; (a) electricity costs, (b) capital costs

unit velocity would typically be greater than 3 $m.s^{-1}$ with the main duct velocity at 8 $m.s^{-1}$.

Air leakage from ductwork should be minimised to prevent the wastage of fan power. Ductwork should be insulated accordingly and runs through unoccupied spaces should be minimised. Testing of ductwork air tightness should be undertaken(50).

Good duct design should achieve airflow that is as laminar as possible throughout the ductwork run to reduce the pressure drop. **To** achieve this:

- changes to the direction of the flow should be minimised
- where possible 2-3 diameters of ductwork should be allowed either side of components before changing direction
- radius bends should be used in preference to right-angled bends
- Y-junctions should be used in preference to T-junctions
- turning vanes should be used wherever appropriate
- for rectangular ductwork, the aspect ratio should be as close to unity as possible.

4.4.3.3 Noise

Noise should be prevented from getting through to the occupied spaces. Design features in support of this objective, which largely correspond to those required for energy efficiency, include the following:

- a low air velocity in the ductwork
- the use of round ducts
- the use of bends with large internal radii
- smooth transitions and changes in flow direction
- the use of low-noise control vanes
- low air leakage.

4.4.3.4 Ductwork hygiene and filtration

In order to maintain ductwork hygiene, both the supply and recirculated air streams should be clean(51). Access must be available for cleaning to minimise the build up of microbial growth on ductwork, fan blades or coils(52). The latter can result in loss of performance. There is also a need for regular inspections.

To minimise pressure drops caused by filtration, the airflow entering a filter should be laminar, requiring the filter surface to be as large as possible. A manometer should be installed across each filter bank to ascertain when filters need changing and access doors should be provided for ease of filter replacement.

4.4.3.5 Heat recovery

Heat recovery within mechanical ventilation systems becomes economic when the value of the recovered heat or cooling outweighs the increase in fan capital and running costs, as well as those of the heat recovery equipment. The viability of heat recovery increases:

- as the number of air changes per hour increases and the heating/cooling season lengthens
- as the temperature difference between supply and extract air streams increases
- with increased proximity of the supply and extract air streams, although it can still be considered when they are not adjacent through the use of a run-around coil.

Heat recovery can increase the overall pressure drop and subsequent fan power used by 50%, although options such as double accumulators offer high heat recovery efficiencies and lower pressure drops. See section 5.6 for guidance on heat recovery equipment

When heat recovery devices are used in full fresh air systems, parasitic losses should be avoided in summertime operation by the use of a by-pass. Effective damper control for minimum fresh air and free cooling on recirculation systems should be provided through enthalpy control, see section 4.3.2.

4.4.3.6 Fire protection

Ductwork must not contribute to the spread of fire, smoke or gases. Therefore in passing through a fire partition the ductwork must not decrease the fire protection properties of the structure. See CIBSE Guide E(40) for guidance on fire protection.

4.4.3.7 Energy efficient control of mechanical ventilation systems

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

- Select efficient fans (see section 5.11).
- 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 of two-speed or variable speed fans. This can be achieved through variable speed drives or inlet guide vanes. (The latter technique is not recommended due to its relative inefficiency.) Further information on variable speed fans is available in EEBPP General Information Report GIR 41(53).
- 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 a 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.

- Appropriate coverage of a building by mechanical ventilation, i.e. using natural systems where applicable (mixed mode approach), see section 4.5.
- Control fan operation according to occupancy in both 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 which break down unnoticed (or continue to operate when cooling is required); 'free cooling' control systems which introduce the wrong proportions of outside air; and 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 become noticed.

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 in section 4.3.3.7. The most popular options are:

- ***CO, 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.

4.4.4 Performance assessment

At the time of writing (July 2001), mechanical ventilation and air conditioning systems are about to become controlled services for the purposes of the Building Regulations for England and Wales. The provisions of Part L will apply. Within these provisions it is possible to make an assessment of specific fan power (SFP) defined by the Approved Document[54] as:

> 'the sum of the design total circuit watts including all losses through switchgear and controls such as inverters, of all fans that supply air and exhaust it back to outdoors (i.e. the sum of supply and extract fans) divided by the design ventilation rate through the building'

Minimum standards are given within the Building Regulations of a maximum limit of 2.0 W per litre.s^{-1} in new buildings and 3.0 W per litres' for a new system in a refurbished building or where an existing system is being substantially altered. ECON 19[55] currently suggests a single good practice figure of 2 W per litre.s^{-1} based on its benchmark data set. However, it has been suggested within the industry that in new premises it may be possible to attain SFPS of 1.0 W per litre.s^{-1}.

Appendix 4.A1 considers a fuller range of assessment techniques available to calculate ventilation and cooling requirements and to look in more detail at air movement.

4.4.5 Further reading

Guide to Good Practice: Air Handling Units **(Marlow: HEVAC Association) (1991)**

Fan and ductwork installation guide **(Marlow: HEVAC Association) (1993)**

4.5 Mixed mode systems design

4.5.1 Introduction[56,54]

Mixed mode ventilation solutions 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.

4.5.2 Strategy

4.5.2.1 Contingency designs

Contingency designs are usually naturally ventilated buildings that have been designed to permit the selective addition of mechanical ventilation and cooling systems 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 so as to be amenable to subsequent naturally ventilated operation, either in part or in whole. Some 1970s offices have been refurbished in this way. Guidance on refurbishment for natural ventilation has been published by BSRIA[58].

4.5.2.2 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 ventilation, with or without cooling, operates 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 basis of a variety of conditions as suggested in section 4.5.3.6.

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.

4.5.2.3 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 ventilation in core areas. The zoned approach works best where the areas are functionally different, or where the systems are seamlessly blended.

4.5.3 General system issues

This section outlines general issues that should be taken into account during the selection and development of a mixed mode strategy. 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 4.5.2. Furthermore, this section cannot be treated in isolation but read in conjunction with sections 4.3,4.4 and 4.6, which consider the principles of the individual operating modes.

4.5.3.1 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 refrigeration 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.

4.5.3.2 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[56].

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

- zones facing 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 beverage machines, tea rooms, or cleaners' cupboards
- restaurants or kitchens
- areas with dense occupation or high equipment heat loads which may require comfort cooling or close control air conditioning such as meeting rooms, electronic data processing rooms, dealer rooms etc.
- atria.

4.5.3.3 Flexibility

Flexibility is of particular concern with contingency systems where future change is taken into account. This requires the provision of a building fabric with a stated indoor environment control performance and a defined strategy for subsequent adaptation through the addition and omission of either centralised or localised supplementary mechanical systems. 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

It may be possible to include space for plant rooms that can be put to alternate use until it is required for ventilation or cooling purposes, e.g. as storage or car parking. External flat roof and undercroft locations may also be suitable. Plant room locations should preferably allow mechanical plant containers to be installed. A further option is prefabricated plant rooms that can be obtained on hire and 'plugged-in' with minimum site disruption. These can subsequently be disconnected for reuse elsewhere when a tenancy terminates.

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 suspended floor may also allow direct expansion, chilled water and condensate pipes to be routed to any potential 'hot-spot'. With appropriate initial sizing the floor void also has the potential to become a floor supply plenum, from which individual 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, these being provided with isolating valves or proprietary, self-sealing couplings. Where appropriate, these basic systems would need to be tested at initial completion to confirm their integrity.

4.5.3.4 Choice of HVAC system

The choice of HVAC system will depend upon the clients' functional requirements, see section 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
- possible planning restrictions on the use of the façade
- the availability of space for logical horizontal and vertical distribution routes.

4.5.3.5 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, see sections 4.3.3.8,4.4.3.7 and 4.6.3.1. 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:

- Mechanical systems should be used only when and where required. The specific 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. For example, 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.

4.5.3.6 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 control for occupants that is non-presumptive
- simple and effective controls for the building management that are easy to commission and operate on occupation of the building.

The general principles of a good control strategy are given in section 4.3.3.7. In the case of a mixed mode system it is also important to remember that the control characteristics of windows differ from the 'designed' characteristic of HVAC dampers and coils. The control authority of a window is low and non-linear or proportional, hence the use of sophisticated control algorithms will not bring greater accuracy. Given the pulsing effect of the wind or natural ventilation, continuously correcting automatic controls should be avoided and the controls response slowed.

The reactions of the occupants to the control systems must also be allowed for in terms of:

- the provision of intuitive user interfaces and control strategies
- adverse reactions to systems which appear to operate in a capricious manner noticeable by changing noise levels or creating a draught
- giving occupants the ability to manually override automatic controls and the impact on system performance
- providing a rapid response to a requested control action.

Elements of the following control sub-strategies may be included:

- *Normal working day control:* where mechanical cooling is switched on when a pre-determined temperature is exceeded.
- *Seasonal control:* where for example the building is sealed in peak winter and summer conditions under mechanical operation, but runs freely during spring and autumn.
- *Top-up/peak lopping control:* where mechanical cooling is switched on only at times of peak load.
- *Pre/post-occupancy space conditioning:* where selected areas prone to overheating may be cooled outside of working hours to ensure that the space temperature is the minimum acceptable at the start of the working day.
- *Overnight cooling:* where the building thermal mass is utilised either through natural or mechanical means, see section 4.7.
- *Moisture control:* where exposed direct cooling such as chilled ceilings or chilled beams are used.
- *Ventilation control:* where carbon dioxide (CO,) sensors are used as a surrogate indicator of occupancy levels to switch on mechanical ventilation when the level exceeds a pre-set value and occupants have not elected to open windows.

4.5.3.7 Post-handover

Training of occupants in the use of the building control facilities is very beneficial in terms of ensuring energy efficient operation and user satisfaction. This may be achieved through the provision of written statements and demonstrations as part of the handover procedures.

Occupiers and designers should meet regularly for at least one year after initial occupation to review the performance of the building and to identify any alterations and improvements necessary.

4.5.4 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 4.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 4.A1.

4.6 Comfort cooling and air conditioning

4.6.1 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 4.8 to 4.22.

4.6.2 Strategy

General guidance on the relative merits of the most common systems is available from EEBPP Good Practice

Guide GPG 71[59] and on more innovative systems from an IEA Annex 28 publication[60]. CIBSE Guide F[2] adopts the classification system for HVAC systems given in GPG 71[59], see Figure 4.11, and discusses issues relating to the energy efficient design of system families.

4.6.2.1 Centralised all-air systems

These consist of *(a)* constant volume (single or multi-zone), *(b)* dual duct, or *(c)* 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 dual duct systems the heated and cooled air are circulated separately and the two air streams are combined to produce an intermediate comfort temperature. In a variable air volume system, it is the airflow that is altered to meet the requirements of the space.

4.6.2.2 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 VAV with terminal reheat, fan coils, unitary heat pumps and induction units. Both heating and cooling pumped water circuits are normally needed to satisfy varying requirements. Three-pipe systems with a common return are to be avoided as cooling and heating energy are wasted when the return air is mixed.

Tempered fresh air systems limit the humidification and de-humidification capacity. However, this is normally adequate for most applications and discourages attempts at unnecessarily close control of humidity, which is very wasteful of energy. The air handling unit should be sized for minimum 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 reheaters. All partially centralised systems should have local inter-connected 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.

4.6.2.3 Local air conditioning systems

These include 'through the wall' units, split systems, variable refrigerant flow units and individual reversible heat pumps. Local systems can provide filtration, comfort cooling and heating, but not humidification. They are often used as a refurbishment option. Local units may have lower coefficients of performance than centralised plant but can provide energy savings through reduced distribution losses, simpler heat rejection equipment, greater control over operating periods, and their ability to be more readily confined to the areas of greatest need.

4.6.3 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 4.4 on mechanical ventilation and section 4.5 on mixed mode systems. Design guidance for individual HVAC systems is given in sections 4.8 to 4.22.

4.6.3.1 Energy efficient operation of comfort cooling or air conditioning systems[2]

The energy efficient design of comfort cooling and air conditioning systems starts by considering the issues raised in sections 4.4.3.7 and 4.5.3.5. Emphasis is then placed on the cooling and humidification processes, e.g:

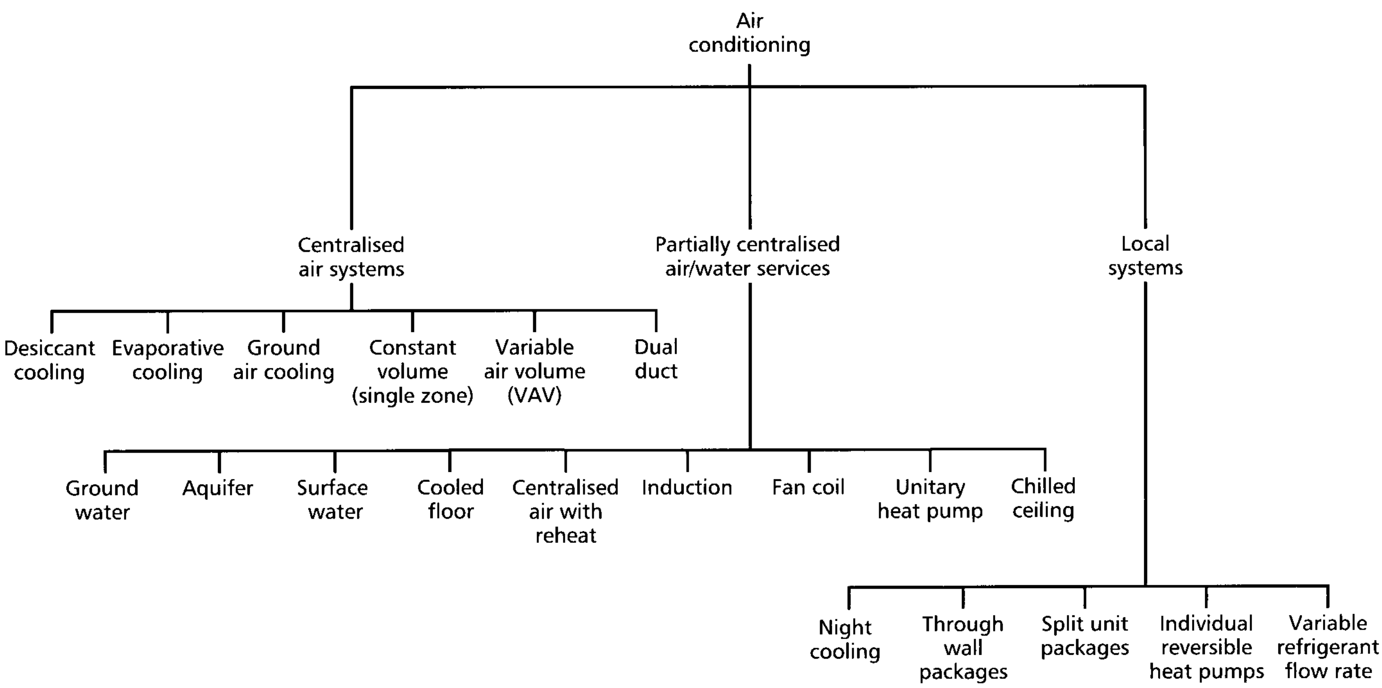


Figure 4.11 Classification of HVAC systems

- Ensure plant is not oversized, see BSRIA Guidance Note GN11/97[61].
- Consider switching off humidifiers when humidity control is not critical. Allow the humidity to drift between 40 and 65%, 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. Gas-fired humidifiers should be considered as an alternative.
- 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 if made of recirculated air and fresh air for free cooling as appropriate, see sections 4.6.1.2 and 5.6.

These issues are considered in more detail below.

4.6.3.2 System control

The general principles of a good control strategy are described in section 4.3.3.7. Figure 4.12 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 and mechanical ventilation systems.

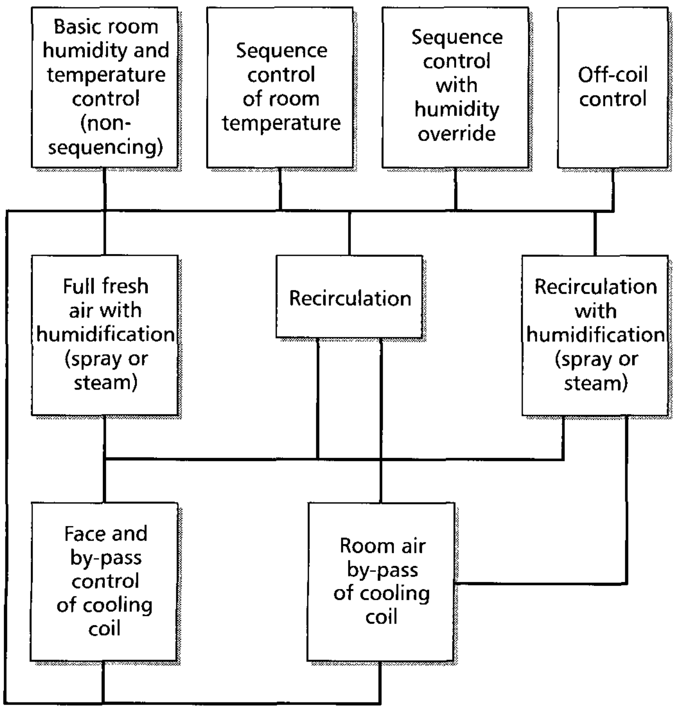


Figure 4.12 System control

'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 4.2) may use higher supply temperatures for summer cooling and can occasionally dispense with the need for mechanical refrigeration by some combination of the following:

- drawing outside air from a shaded 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.

If mechanical refrigeration 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 input, 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. Again, it may be possible to limit the rise in inside temperature by drawing air into the central plant at a lower temperature than the outside dry-bulb temperature.

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[62] (h_r), see Appendix 4.A2, Table 4.8.

'Free cooling' is also available via cooling towers providing cooling water without the need to operate the chillers. Guidance on control strategies can be found in BSRIA publication RR16/96[63].

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 5.5). In these cases, electric or water-fed serpentined coils should be provided, switched at 4-5 °C from a downstream thermostat.

Simultaneous heating and cooling

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 controls, not indicated on the system schematics in sections 4.8 to 4.22, may be required to deal with early morning boost, heat recovery and variable occupancies[62]. 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 section **1** of CIBSE Guide A[1] and section 6 of CIBSE Guide F[2]. Guide A suggests that at design temperatures normally appropriate to sedentary occupancy, the room humidity should, if possible, be above 40%. Lower humidity is often acceptable for short periods. Humidity of 35% 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% is proposed to minimise the risk of mould growth or condensation in areas where moisture is being generated. This can be extended to 70% 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, reheater 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 reheater ***so*** that the sequencing room temperature sensor calls for further cooling and hence dryer air is supplied
- overriding control over the reheater in a variable air volume zonal reheat terminal or the mixing dampers in a dual duct terminal so that the zonal temperature sensor calls for a larger volume of dry air to be supplied to that zone
- resetting the set point of an off-coil 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 is dealt with CIBSE Guide H[62].

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 section 4.19, Figure 4.46)
- 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 4.48)
- 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.

4.6.3.3 Fan position

The systems schematics that follow mainly indicate a 'draw-through' arrangement for the supply air handling plant, with separate extract fan, see Figure 4.13(a). Alternative arrangements include 'blow-through' and combined supply/extract fan. The former is the normal configuration for dual duct systems.

'Blow-through' central plant

The main advantages of positioning the fan upstream 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 4.13(b))
- the cooling coil condensate drain will be under positive pressure, which reduces the chances of drawing airborne contaminants from the drainage system or plant room into the system.

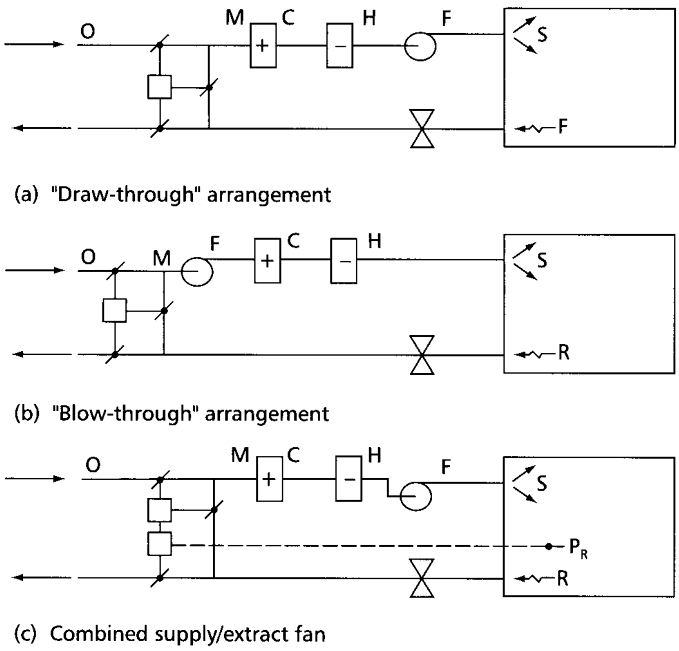


Figure 4.13 Supply air handling plant; alternative arrangements for position of fans

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 4.13(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 4.13(c) shows a relief damper controlled from a room pressure sensor, P_R. For extract systems having low resistance this damper could be replaced with simple, weighted pressure relief flaps, also see section 5.5.

4.6.3.4 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 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 other variables are in phase. 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.
- Gains through poorly insulated roofs are quite similar to gains on south facing surfaces, but if adequately insulated they may be treated as intermediate floors.

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.

4.6.4 Performance assessment

Appendix 4.A1 considers the techniques of dynamic thermal simulation and air movement analysis. For England and Wales, Building Regulations Approved Document L(54) includes specific performance requirements for air conditioning systems, see section 4.4.4. 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 Building Energy Code 2(64)
- ASHRAE BIN method(65)
- Energy Efficiency Best Practice Programme (EEBPP) Energy Consumption Guides(66).

4.7 Night cooling and thermal mass

4.7.1 Description

Night cooling in combination with a thermally heavyweight building can be used as a means of avoiding or minimising the need for mechanical refrigeration 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, thereby reducing or eliminating the need for mechanical refrigeration to maintain thermal comfort.

Interaction between the mass and the air, see Figure 4.14, 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[67]. System pressure drops should be minimised to maximise energy efficiency.

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** $W.m^{-2}$.

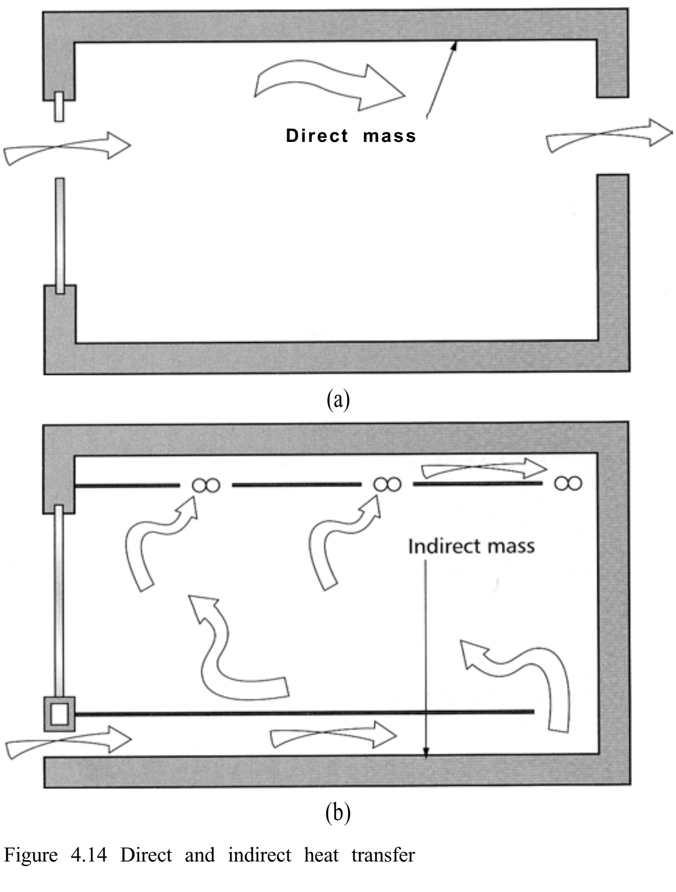


Figure 4.14 Direct and indirect heat transfer

Where mechanical cooling is provided, night cooling of the building mass can either 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:

- reduction in mechanical cooling requirements during the occupied period
- 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.

4.7.2 Design

Figures 4.15 and 4.16 illustrate a number of design solutions 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
- 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[68]. 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.

4.7.2.1 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
- the rate of heat transfer between the element and the air.

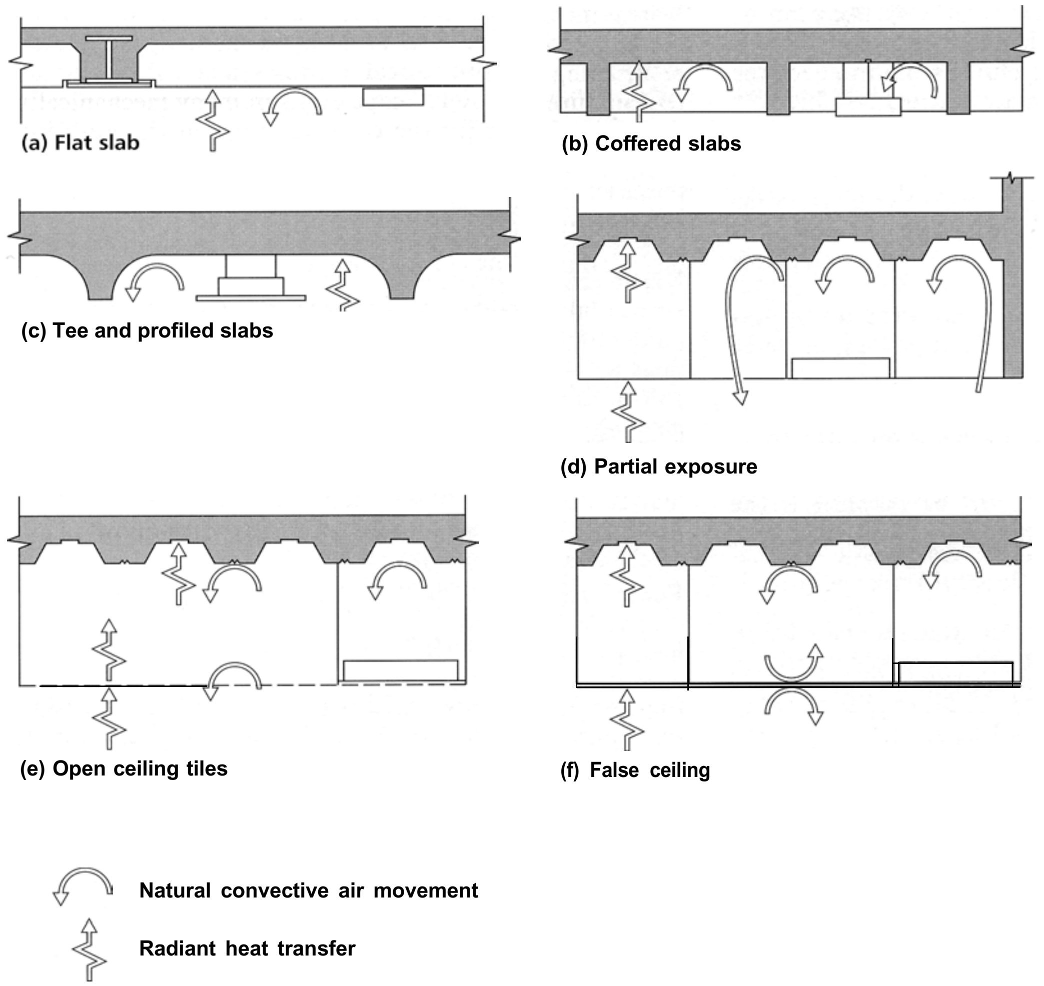


Figure 4.15 Design details for exposing thermal mass

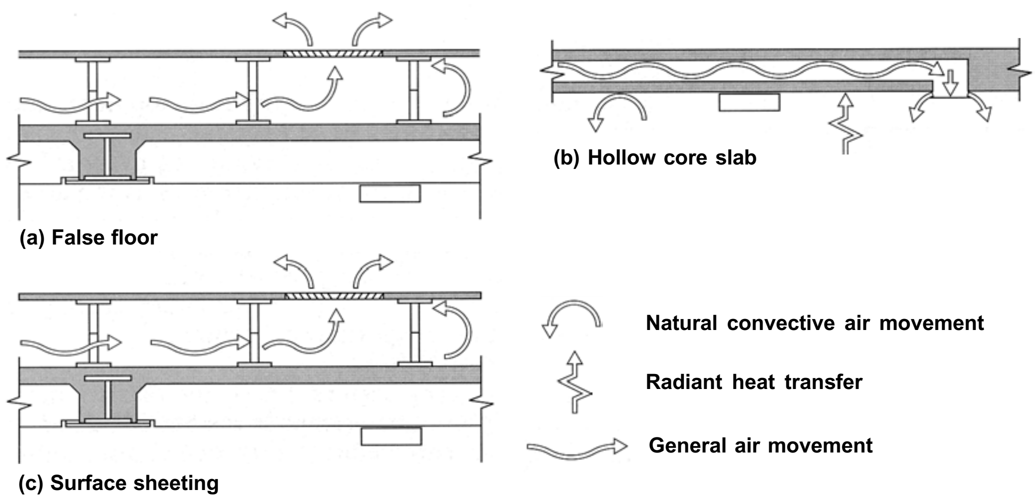


Figure 4.16 Design details for indirect interaction

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 section 3 of CIBSE Guide C[48]. For exposed plane surfaces typical values are 5 $W.m^{-2}.K^{-1}$ for radiation and 2-3 $W.m^{-2}.K^{-1}$ 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[48,69]. 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 e.g. carpets), although thin layers of relatively conductive materials (e.g. plaster) should not 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 $W.m^{-2}.K^{-1}$)[67], 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 $W.m^{-2}.K^{-1}$ 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[70]. 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 section 3 of CIBSE Guide C[48]. (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 section 4 of CIBSE Guide C[48] (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[1]. 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 (see BRE Information Paper IP6/2000[71]).

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.

4.7.2.2 Conflict with air heating/cooling

For mechanical ventilation systems, there is a potential for conflict between heat exchange with the therma1 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 pick-up from lights increasing the cooling effect of the thermal mass.

4.7.2.3 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 visua1 appearance. Where there is a desire to conceal parts 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.

4.7.2.4 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 (usually) mineral fibre ceiling material. This can give rise to increased reverberation time and increased reflected sound across an open plan space. Measures available to overcome these problems 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[29]. 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 B5: ***Sound control***[34] for detailed guidance on acoustics and surface finishes.

4.7.2.5 Integration

The absence of a suspended ceiling (and with it the ease of integration of services including lighting, smoke detectors and sprinklers) 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 identified in the Steel Construction Institute report ***Environmental floor systems***[70] include pendant systems, floor or furniture-mounted uplighters, cornice and slab recessed[29]. With uplighting, the soffit form is highlighted as an important consideration together with a high surface reflectance of at least 70-80% and a gloss factor of no more than 10% (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.

4.7.2.6 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. overcooling). Inappropriate control strategies can result in significant increases in heating demand (+20%) without appreciable reductions in peak temperatures[68].

Strategies are generally 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.

Refer to BSRIA Technical Note TN14/96[72] for detailed guidance on developing suitable strategies. Where the mechanical cooling is provided, refer to BSRIA Technical Note TN16/95[73] for detailed guidance on pre-cooling strategies.

4.7.3 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[74].

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

- 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.

4.8 Chilled ceilings/chilled beams

4.8.1 General[25]

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% (maximum) of the heat is transferred by radiation.

Chilled ceilings use chilled or cooled water as the cooling medium, normally between 13 °C and 18 °C. There are many different types of chilled ceiling devices, but essentially they fall into three main categories, see Figure 4.17. 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 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 ventilation is an integral part of the beam, being induced by the central air handling plant. However with passive chilled beams and panels, ventilation has to be introduced separately, either by mixed flow or more normally by displacement ventilation. 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[75]. However, in this context, it is very important to consider condensation control, see section 4.8.2.4.

4.8.2 Design

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

4.8.2.1 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}. To provide 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 downdraughts from chilled ceilings delivering high cooling outputs. At these loads physical testing or CFD modelling of the design may be required. Further information on these systems is available elsewhere[76].

The ratio of convective to radiative heat output for various systems is shown in Table 4.5.

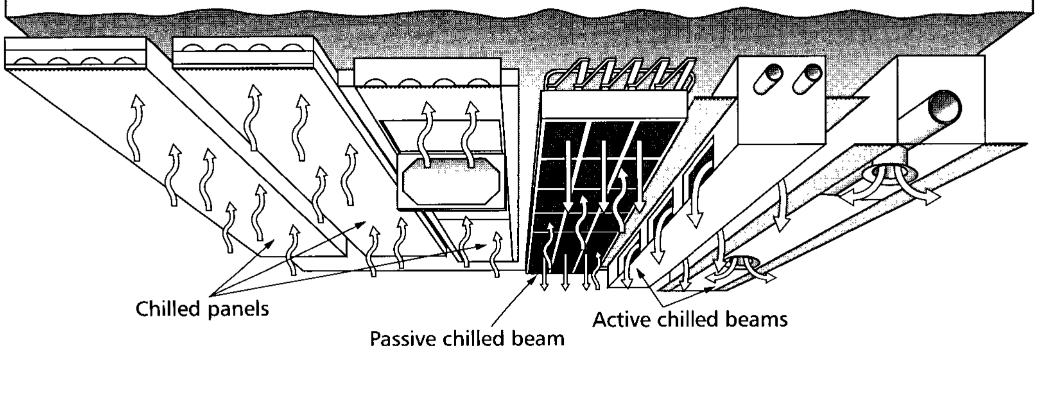


Figure 4.17 Chilled ceiling categories

Table 4.5 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 section 4.15) or cooling towers. This will increase their coefficient of performance(77). As cooling is supplied within the space this limits the requirements for the ventilation system to provide fresh air, thus also saving fan energy.

4.8.2.2 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 field 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(78) 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 field 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.

4.8.2.3 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 2-port, 2-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 by-pass 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. 2-, 3- or 4-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 section 4.8.2.4. For central control a 3-port valve is needed to regulate the inlet water temperature. If 2-port valves are used in rooms, then a header tank between the chiller and pipework will ensure a constant flow rate to the chiller. A by-pass valve at the end of each branch decreases the pressure in the pipework and is particularly important with a 2-port valve system. This also ensures that a constant chilled water supply is available.

4.8.2.4 Condensation risk(79)

The avoidance of condensation on the surface of chilled panels and beams has been a major design issue in the UK, with fears over the relatively wet climate of the UK. 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 either 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 elsewhere 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.

To minimise the condensation risk, it is important to lag the chilled water pipework between the panels or beams 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, e.g. 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 re-calibrated.

Condensation can also be avoided by reducing the room dew point temperature by reducing the supply air temperature. This may increase the risk of draught, particularly when using displacement ventilation.

4.8.2.5 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.

4.8.2.6 Noise

Compared to 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).

4.8.2.7 Performance monitoring

Guidance on modelling the performance of chilled ceilings in conjunction with displacement ventilation is available at both early stage design[78] and detailed design[25].

4.8.3 Construction

4.8.3.1 Ceiling layout

In practice the high heat gains in modern office spaces are served by a combination of 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 4.18. It is also important to consider integration with the light fittings.

4.8.3.2 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.

4.8.4 Further reading

Alamdari F, Butler D J G, Grigg P F, Shaw M R Chilled ceilings and displacement ventilation ***Renmuble Energy*** **15 30C305 (1998)**

Thomas D, Davies G, Dickson D and Bunn R Chilled ceilings - a design primer ***BuildingServicesJ.*** **16** ***(6)*** **27-34 (June 1994)**

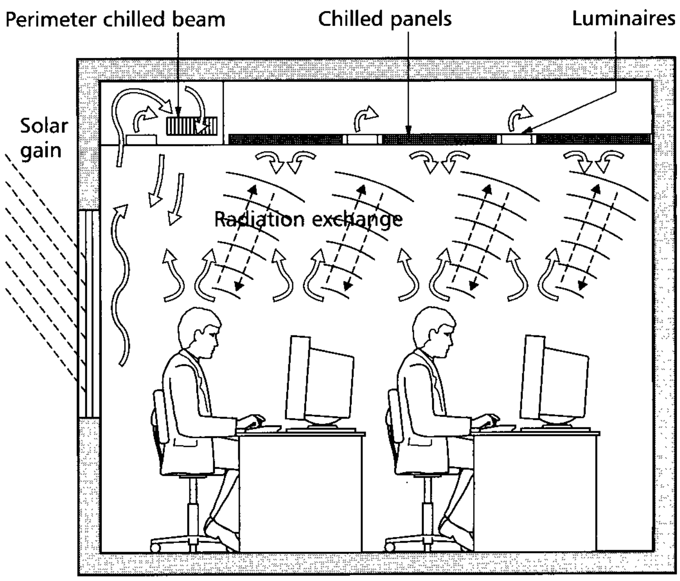


Figure 4.18 Typical ceiling layout incorporating chilled panels and chilled beams

4.9 Cooled surfaces (floors and slabs)

4.9.1 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, air blast coolers (see Figure 4.19) or aquifers. This helps either 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.

4.9.2 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[1], section 5 (equation 5.133). 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[1]. 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.

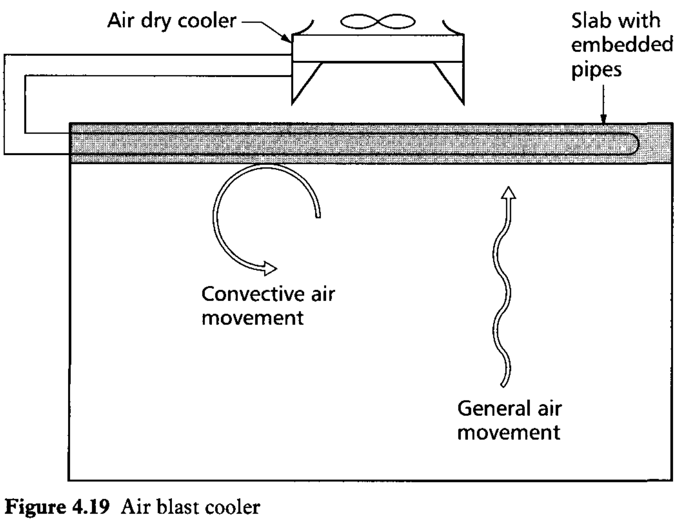


Figure 4.19 Air blast cooler

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[48,69].

Exposure of soffits raises a number of issues that should be considered including aesthetics, acoustics and integration. Refer to section 4.7 for further guidance.

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[80]. 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[80] to achieve effective storage and heat conduction to the surface, see Figure 4.20. Optimum values can be evaluated using conduction models.

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.

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:

- ***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.

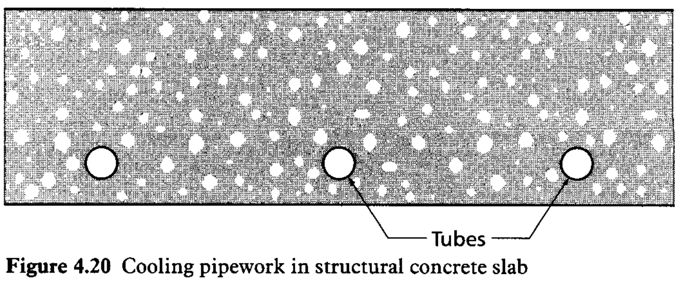


Figure 4.20 Cooling pipework in structural concrete slab

4.9.3 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 reducing 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 Figures 4.22 and 4.23, including:

support via joists or battens

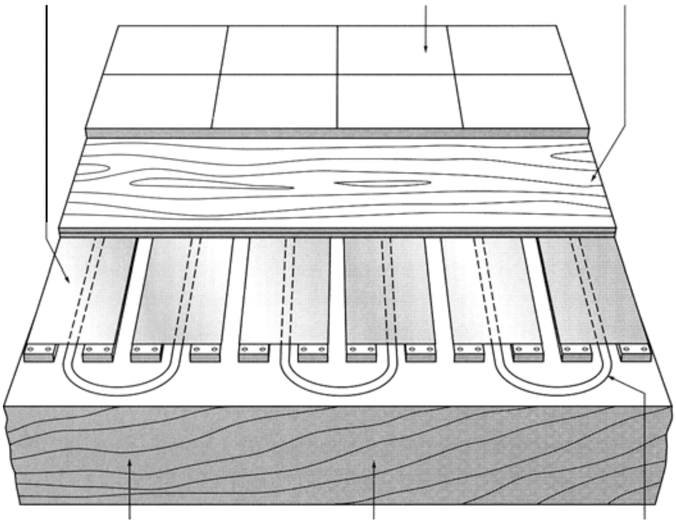


Figure 4.21 Cooling pipework in sub-floor

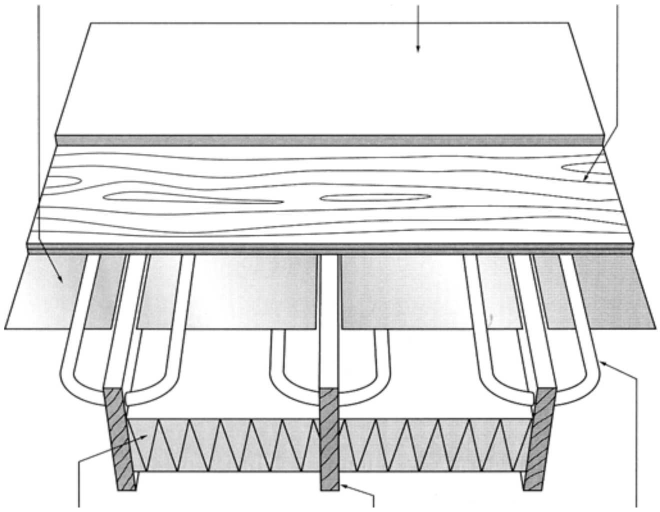


Figure 4.22 Cooling pipework below sub-floor

- 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 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 4.23. 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 4.24. 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).

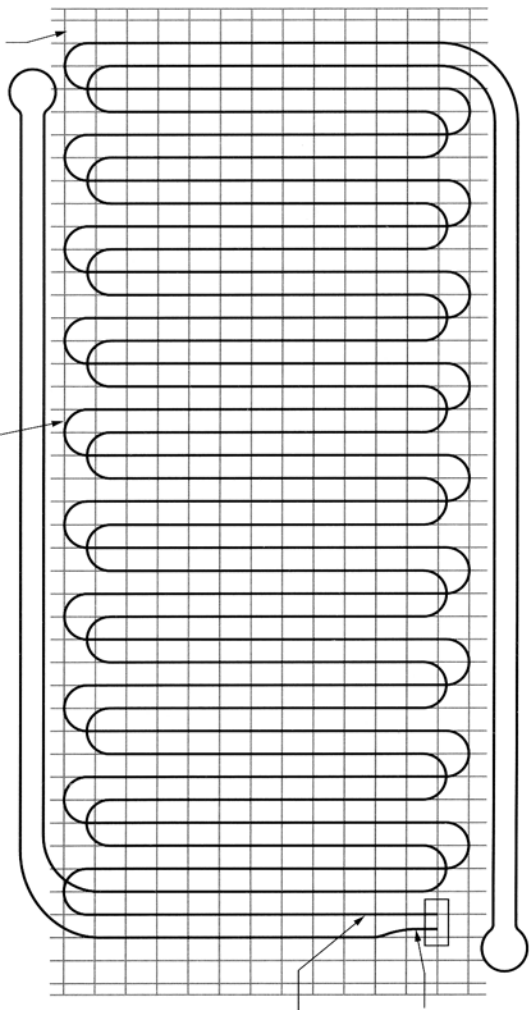


Figure 4.23 Cooling pipework supported by reinforcement cage

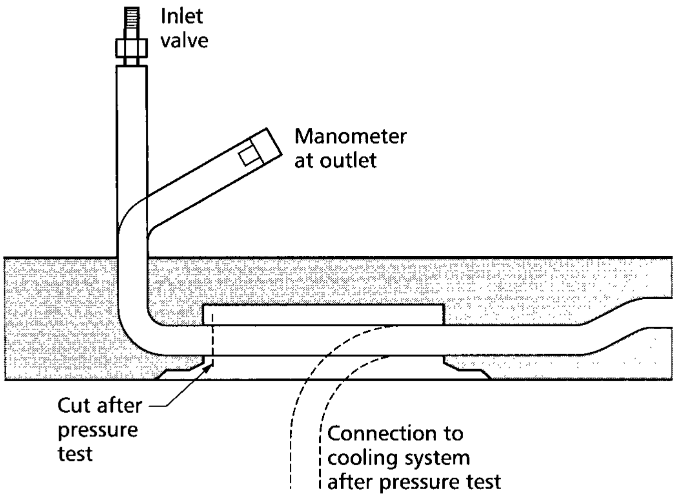


Figure 4.24 Pre-mounted connection box for cooling pipework

4.10 Desiccant cooling systems

4.10.1 Description[81]

Desiccants are hygroscopic materials that readily absorb or give off moisture to the surrounding air. They can be solids or liquids, although application of dessicant 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 air stream then 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'.

Dessicant systems can be applied where:

- high latent loads are present that would otherwise require very low chilled water temperatures, e.g. supermarkets
- contaminant control is required
- a source of low-grade energy such as waste heat or solar energy can be used to regenerate the dessicant.

4.1 0.2 Design

4.10.2.1 Operation

Figure 4.25 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.

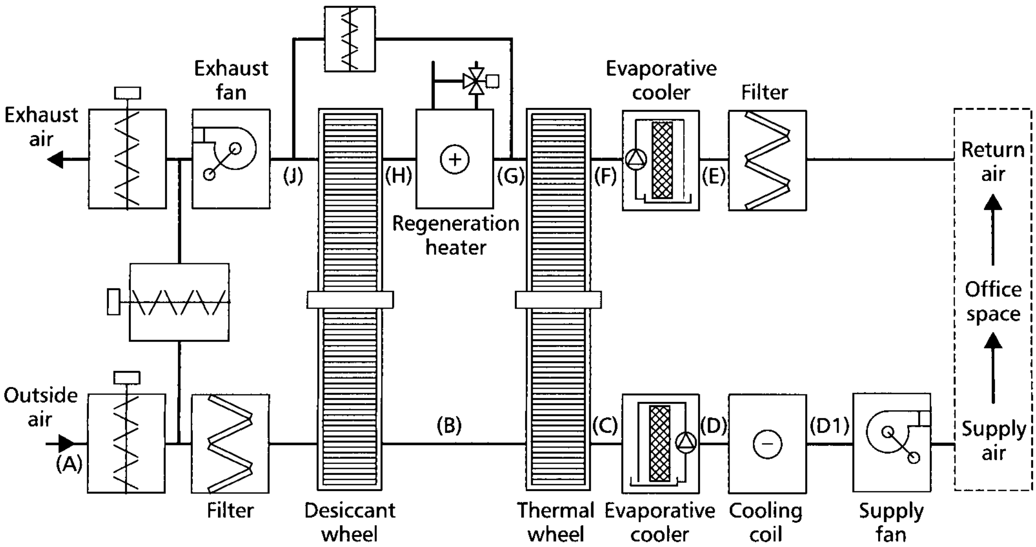


Figure 4.25 Typical air conditioning plant using solid desiccant technology

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 4.26.

4.10.2.2 Performance

Like any other system, performance is dependent on the external and internal conditions. The difference between desiccant systems and those based on HCFC/HFC-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.

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 ($g.kg^{-1}$) can be achieved. The system can be used in all types of air conditioning systems, but is particularly effective with radiant cooling either by chilled ceilings or fabric thermal storage.

The cooling and dehumidification capacity of a dessicant 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% 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% of the internal heat gain without any energy input by using only exhaust air evaporative cooling and the thermal wheel.

At approximately 75% 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.

4.10.2.3 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 thus reducing system resistance and hence fan energy.

4.10.2.4 Maintenance

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

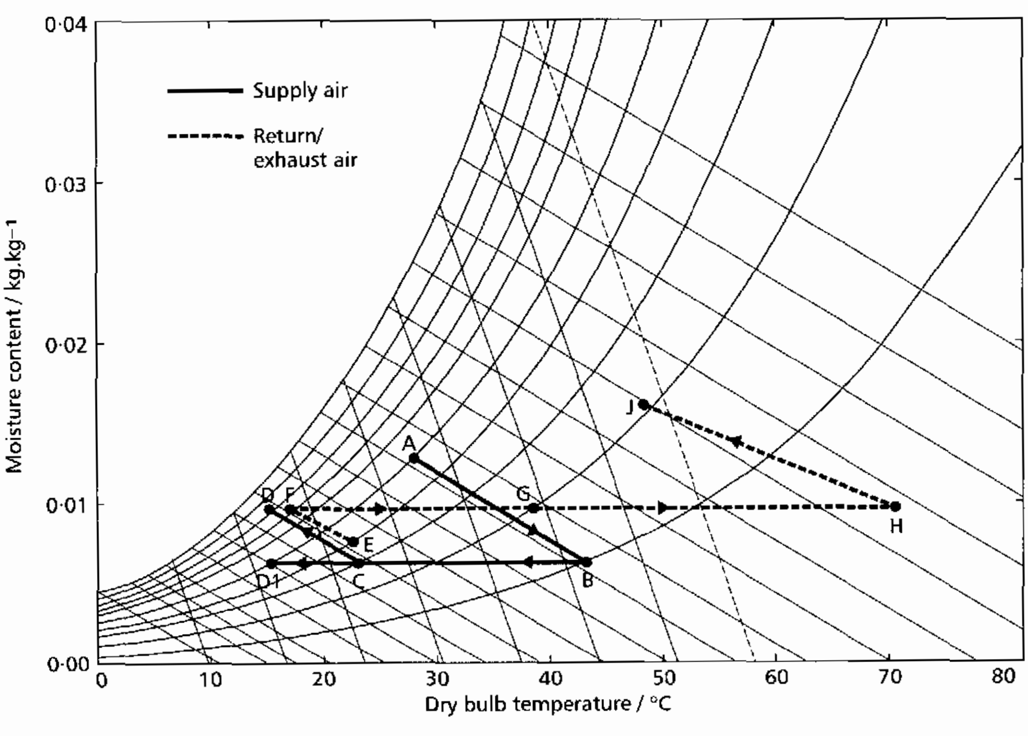


Figure 4.26 Psychrometric process for desiccant cooling

4.10.2.5 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 $m^3.s^{-1}$). However, this should be balanced against running cost and CO, production savings. A cost and environmental benefit analysis will be required for individual projects.

4.10.3 Space requirements

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

4.10.4 Further reading

Barnard N and Jaunzens D (eds.) *Low **energy cooling** - **technologies selection and early design guidance*** IEA Energy Conservation in Buildings and Community Systems Programme EP56 (London: Construction Research Communications) (2001) **ISBN 186 081 4581**

Brister A Cold calling ***Building ServicesJ.*** **18** (8) 27-28 (August 1996)

Desiccant dehumidification and pressure drying equipment Chapter 22 in ASHRAE Handbook: ***HVAC Systems and Equipment*** (Atlanta, CA: American Society of Heating, Refrigerating and Air-conditioning Engineers) (2000)

Zimmermann M and Anderson J (eds.) *Low **energy cooling** - **case study buildings*** IEA Energy Conservation in Buildings and Community Systems Programme (St Albans: Oscar Faber Group)

4.11 Dual duct (constant volume and variable air volume) and hot deck/cold deck systems

4.11.1 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 air streams 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.

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 deckhold 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.

4.11.2 Design

4.11.2.1 Dual duct constant volume

A typical system configuration is shown in Figure 4.27, with the associated psychrometrics in Figures 4.28 and 4.29. Supply temperatures from the air handling unit 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 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 most common solution).

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

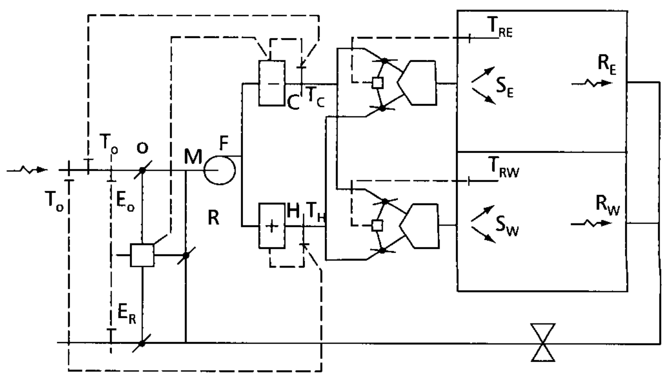


Figure 4.27 Dual duct system

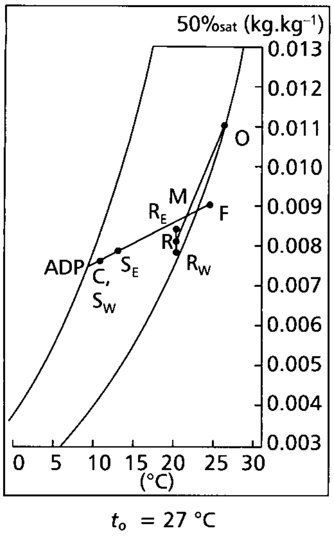


Figure 4.28 Psychrometric chart: dual duct, hot deck/cold deck

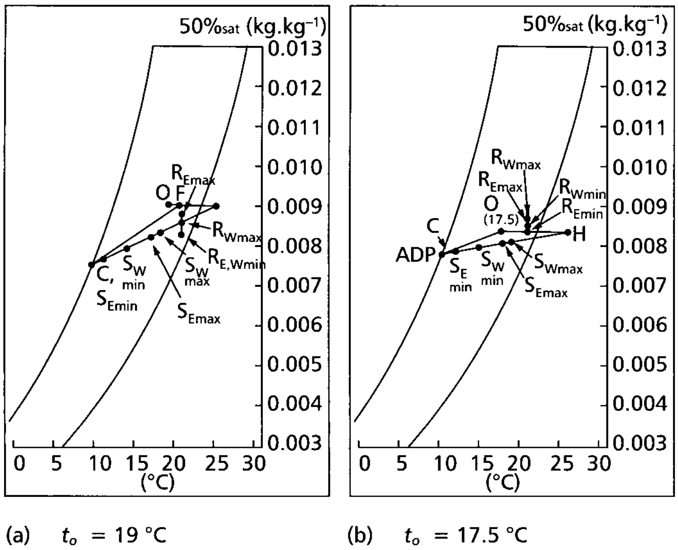


Figure 4.29 Psychrometric chart: dual duct, hot deck/cold deck

Fresh airpreheat

A preheater 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

The provision of 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.

4.11.2.2 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.

4.11.2.3 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 'multizone' air handling units capable of serving a limited number of zones are available (see Figures 4.30 and 4.31) 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% 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.

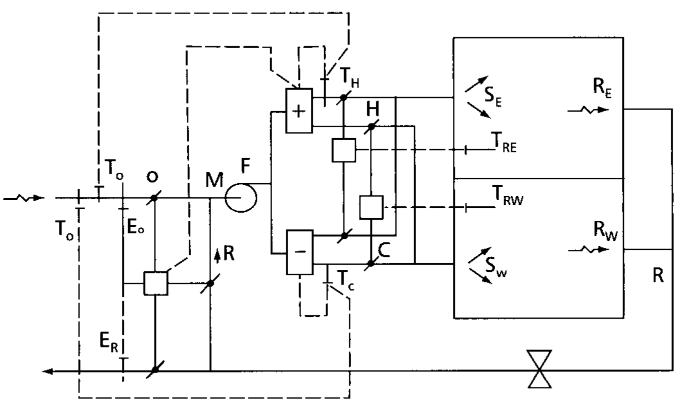


Figure 4.30 Multizone hot deck/cold deck system

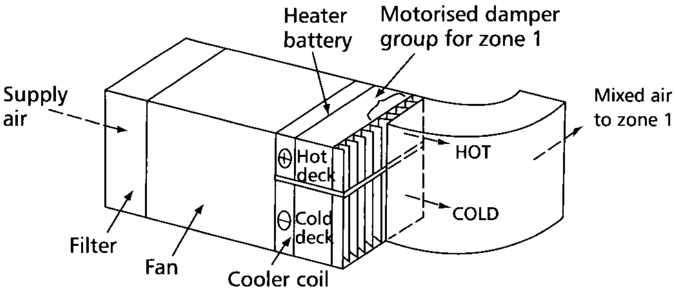


Figure 4.31 Typical packaged multizone arrangement

4.11.3 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 5 for equipment descriptions.

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 4.32 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% of full flow rate for small, well-made devices up to 10–20% 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% 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 air stream 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.

4.12 Evaporative cooling (direct and indirect)

4.12.1 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:

- ***Direct evaporation:*** 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 operate using spray air washers or wetted media.
- ***Indirect evaporation:*** two air streams are used. A secondary air stream is cooled directly using evaporation and then exhausted. This secondary stream may be outdoor or exhaust air. The cooler moist secondary air is used to cool 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 pre-heat

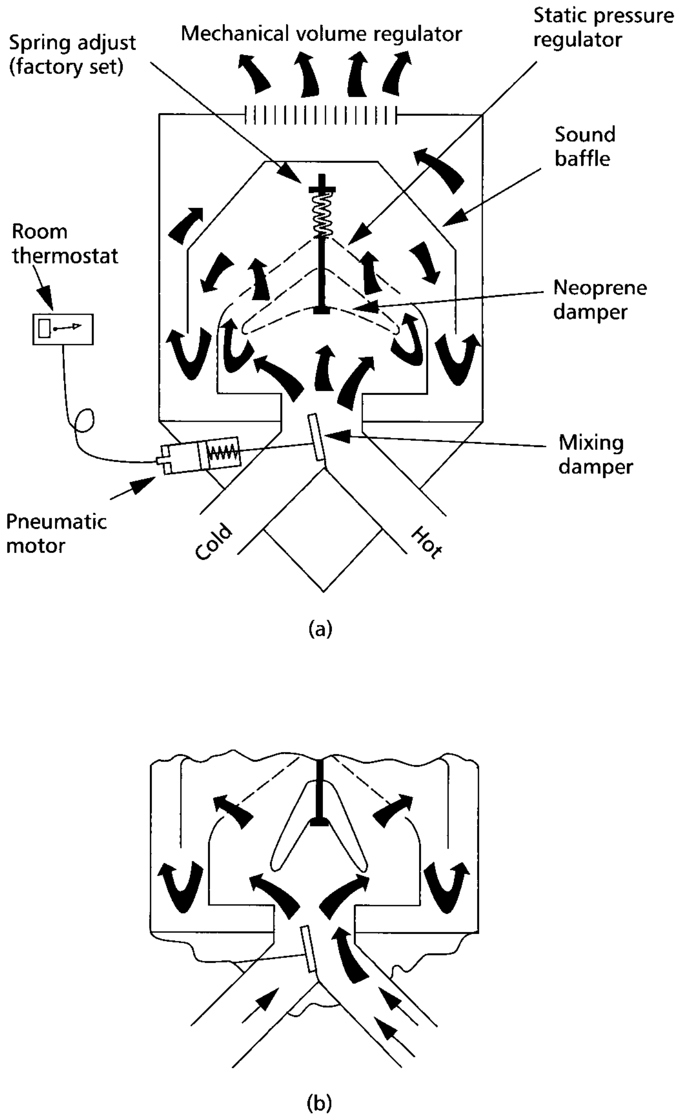


Figure 4.32 Constant volume dual duct unit with integral static pressure regulator and air terminal device; (a) high inlet duct pressure, (b) low inlet pressure

outdoor air in the winter.) Hence indirect evaporative cooling provides sensible cooling without increasing the latent capacity of the supply air.

When designed as a standalone system, an evaporative cooling system requires three to four times the air flow rate of a conventional air conditioning systems. Because of the higher airflow rates, larger ducts are required. However, the higher airflow rates and the absence of recirculated air may improve 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.

4.12.2 Performance details

The lowest temperature that can theoretically be achieved is the dewpoint temperature of the air being treated. The resulting cooling depends on the wet-bulb depression and the cooler effectiveness. In Arizona, where evaporative cooling is used successfully to provide comfort conditions, this is 18 °C whereas in the UK it is 9 °C. 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 air stream velocity, condition and adjustment). For direct spray types, effectiveness is put at 50-90% with the higher values being associated with double spray arrangements. Direct wetted media coolers could have an effectiveness of between 85-95%. Typically indirect pre-cooling stages achieve 60-80% effectiveness. System effectiveness should be considered where more than one stage takes place.

4.12.3 System enhancements

Evaporative cooling may be enhanced by:

- using it in combination 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.

4.12.4 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.

4.12.5 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 standalone system although the temperature depression is less and its humidity exceeds the room air, hence comfort conditions should not be adversely affected.

4.12.6 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 at the end of the summer.

Current guidance on the treatment of water used within direct evaporators[82] would suggest that water treatment should not be undertaken, i.e. the water will need to discharge to waste. Also see CIBSE TM13[83] for guidance on measures to reduce the risk of Legionnaires' disease. No distinct guidance is given on the use of indirect evaporative systems.

4.12.7 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.

4.12.8 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% for a direct system. Hence their use tends to be restricted to providing supplementary cooling or in combination with desiccant cooling in the UK.

4.12.9 Maintenance and health

Within 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.

4.13 Fan coil units

4.13.1 Description

A fan coil is a packaged assembly comprising coils(s), condensate tray collection, 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 air handling unit (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 fitted
- high cooling capacity
- flexibility to accept future changes in load and layout.

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 that of the most efficient AHU fans. Fan coil systems generally have relatively high maintenance costs and short operating lives. 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, shops, 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.

Vertical units require floor and wall space. Vertical units located under windows or on exterior walls are suitable for 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.

4.13.2 Design

The various 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 reheaters 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. 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, see Figure 4.33.

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.

The central AHU is typically a full fresh air system with off-coil control of heating and cooling coils, including humidification if required, see section 5.10. The ventilation air will normally be supplied at a neutral

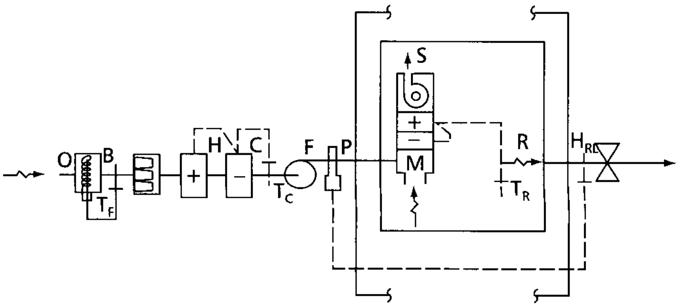


Figure 4.33 Four-pipe fan coil system

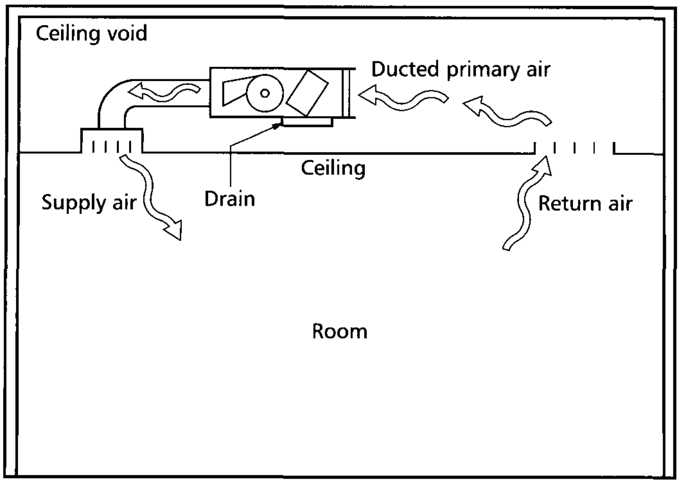


Figure 4.34 Ceiling void fan coil unit with separate primary air

temperature to minimise loads on the fan coils[57]. 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 section 5 of CIBSE Guide H[62] 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 are also likely to 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 can avoid maintenance problems caused by valves blocking, but may require slightly larger units and can suffer from problems such as carryover. It should be understood that 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 either return air or room temperature sensors. Fan coil units can be supplied complete 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 either 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, heat pick-up from light fittings may result in temperatures onto the coils being significantly higher than room temperature[84]. This should be taken into account in unit sizing.

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, also to ensure that there is an adequate dead band between heating and cooling. Care should be taken to avoid conflict between fan coil units with separate control systems but 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.

4.13.3 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 and therefore 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 section 5.7.3) or a cleanable wire mesh type 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 as appropriate, and cleaning and inspection of the condensate drain pan and system.

See section 5 for equipment requirements.

4.14 Ground cooling (air)

4.14.1 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 4.35. 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 and is not recommended in areas with radon gas.

4.14.2 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 4.36. Pipework should be positioned vertically as deep as possible in the ground without incurring prohibitive excavation costs (i.e. 2-4 m)[85]. 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 well insulated, causing the ground to heat up and performance to drop.

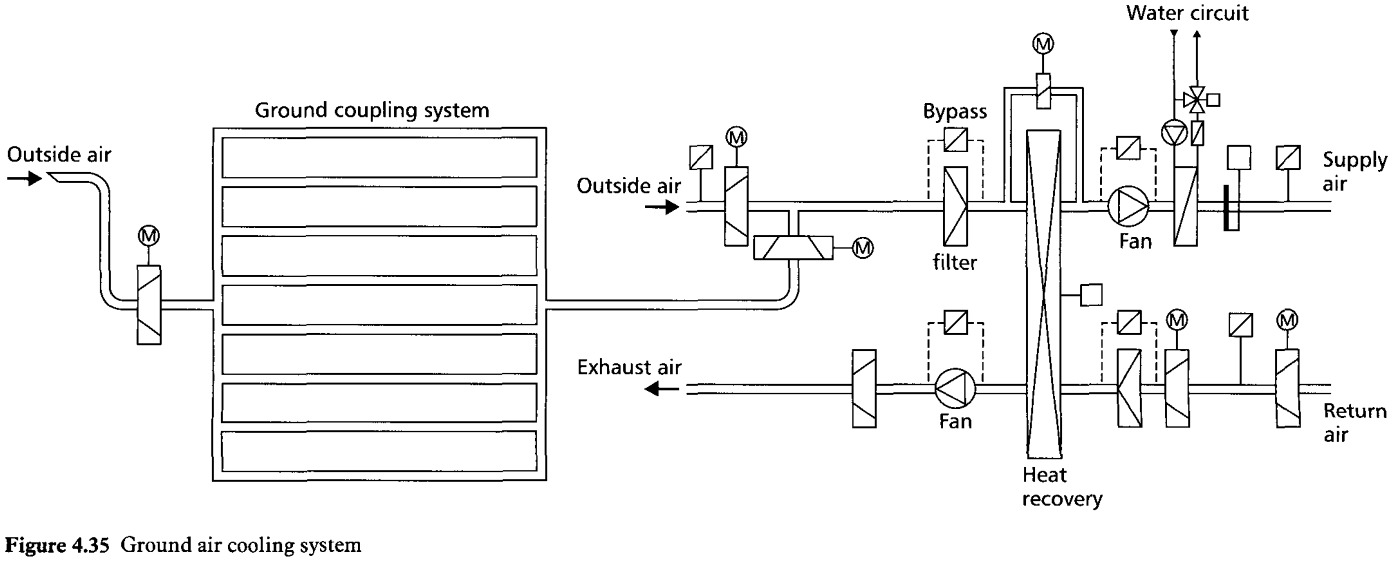


Figure 4.35 Ground air cooling system

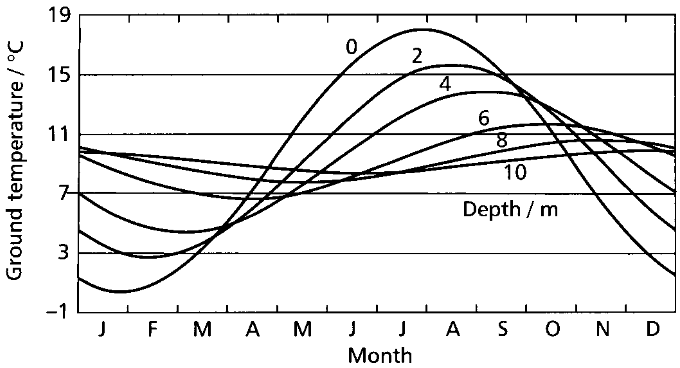


Figure 4.36 Ground temperatures as a function of depth below ground

Parameters that need to be considered when designing the pipework system include the following:

- ***Horizontal spacing of the pipes:*** this should be such that the mutual interference between adjacent pipes is not too great (e.g. 1 m)[85].
- ***Design air velocity:*** this should be selected to achieve good heat transfer performance without incurring high pressure drops (e.g. 2 m.s^{-1})[85].
- ***Pipe diameter and length:*** these should be selected to achieve effective heat exchange[69], typically 80% of the maximum possible (e.g. 200 mm diameter pipes of 20-25 m length, with larger pipes at increased lengths)[85].
- ***Soil type:*** this has a limited influence on thermal performance (e.g. ±10%)[85], 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.

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 pre-set 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[85]. Thermal design simulation packages that have the facility to model three dimensional conduction can also be used for assessment purposes.

4.14.3 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% 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 4.37. 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.

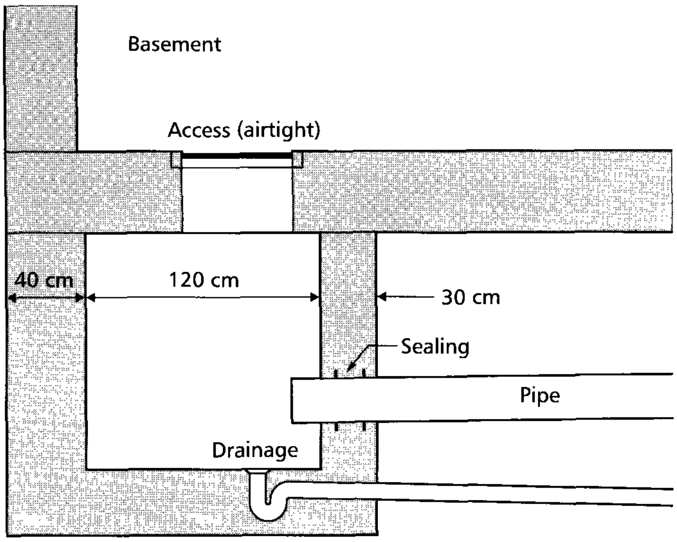


Figure 4.37 Distribution/collection header duct arrangement

4.15 Ground cooling (water)

4.15.1 Description[86,87]

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 two metres 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 level, 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. Section 4.16 deals with the use of ground coupled heating and cooling using heat pumps.

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.

4.15.1.1 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 just 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.

Typical open loop systems require that, following assessment of the geological suitability of the location, 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 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 4.38.

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 litres' 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.

The groundwater cooling system in the BRE Environmental Building (see Figure 4.39) 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.

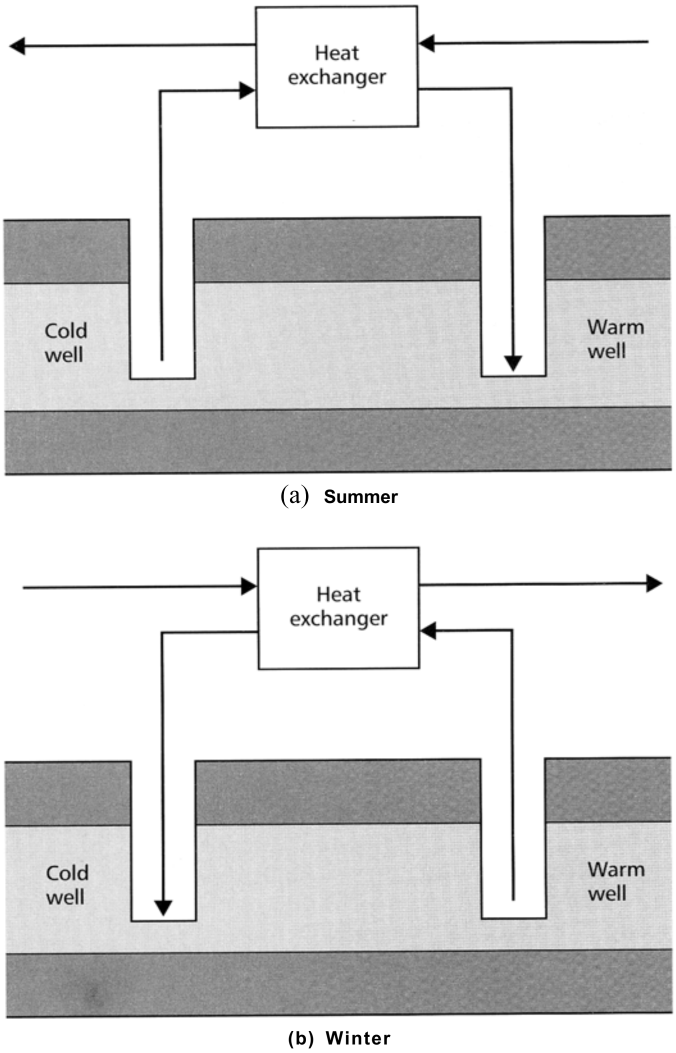


Figure 4.38 Ground water cooling system

4.15.1.2 Closed loop systems

These systems are extremely simple, comprising a continuous loop of high-density polyethylene pipe, through which water is circulated, buried in the ground. The water is recirculated 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 m deep. These are backfilled with high-density grout both 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.

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

- *'Slinkies':* these are a variation of horizontal loops, so-called because they are supplied as a tightly coiled spring similar to but larger than the children's toy of that name. The spring is released and the resulting looped pipework is either spread horizontally at the bottom of a trench one metre in width and depth or installed vertically in a two metre deep narrow (0.25 m) trench. Performance is similar to that of a horizontal loop but may be reduced if the pipe overlaps itself. It is a useful technique for situations where excavation is easy

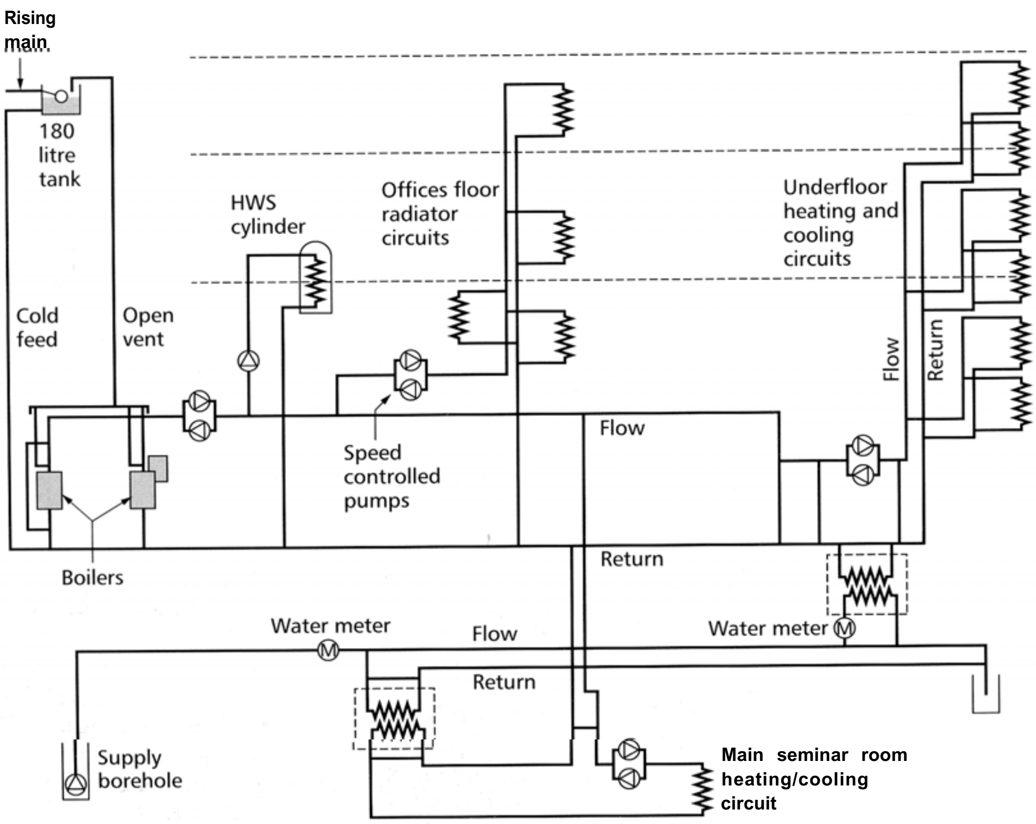


Figure 4.39 Schematic of BRE groundwater cooling system

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 system capacity.

4.15.2 Performance

Heat transfer rates are likely to be low because of the small temperature differences between the loop circulating water and the ground. Extrapolating from closed loop ground source heat pump design suggests that vertical boreholes may deliver 25 $W.m^{-1}$ bore depth, but this has not yet been widely achieved. Horizontal systems are likely to yield less cooling since the ground temperature will be higher in summer when the main demand for cooling occurs.

4.15.3 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% of the UK landmass is suitable for aquifer based open loop technology, and virtually 100% is suitable for closed loop installations.

4.15.4 Space requirements

For both open and closed loop systems the main space

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 pits should be at least 100-150 m.

4.15.5 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.

*British Geological Survey, Keyworth, Nottingham NG12 5GG, UK

4.15.6 Maintenance

With open loop systems it can be difficult to pressurise the ground to return the water, hence there may be problems with boreholes silting-up due to the growth of algae and the settling of suspended solids. No defrost cycle is required for the water-source heat pumps as they operate over a more moderate range of temperatures.

4.15.7 Further reading

Information is available from the International Ground Source Heat Pump Association†.

4.16 Heat pumps

(Note: dehumidifers are considered in section 5.10.)

4.16.1 Description

A heat pump is a machine that transports low-grade energy and converts it into useful heating energy. Heat pumps are available as both heating only or reverse cycle heating/cooling systems and are classified classified according to the type of heat source and the heat distribution medium used, e.g. air-to-water, air-to-air etc. Table **4.6** lists examples of heat source and distribution systems.

Table 4.6 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

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, e.g. fluorocarbon refrigerant.

†International Ground Source Heat Pump Association, 490 Cordell South, Oklahoma State University, Stillwater, OK 74078-8018 (http://www.igshpa.okstate.edu)

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

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

4.16.2 Design

4.16.2.1 Enhancement of operational efficiency of systems

Systems can be enhanced by employing:

- heat recovery from air, steam or water, see Figure 4.40
- renewable primary energy sources, e.g. solar, wind, water, ground, geothermal etc.

4.16.2.2 System Performance evaluation

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

Theoretical COP

For the vapour compression cycle where a cooling output is considered, the COP is the ratio of refrigeration effect to the work done by the compressor and is known as the COP_R

For the vapour compression cycle where a heating output is considered, the COP is the ratio of heat from the condenser to the work done and is know as COP_H, i.e:

$$COP_H = \frac{\text{Enthalpy change due to condensation of vapour}}{\text{Enthalpy change due to compression of vapour}} \quad (4.15)$$

Hence, from Figure 4.41, the theoretical coefficient of performance is given by:

$$COP_H = \frac{H_3 - H_4}{H_3 - H_2} \quad (4.16)$$

The theoretical 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.

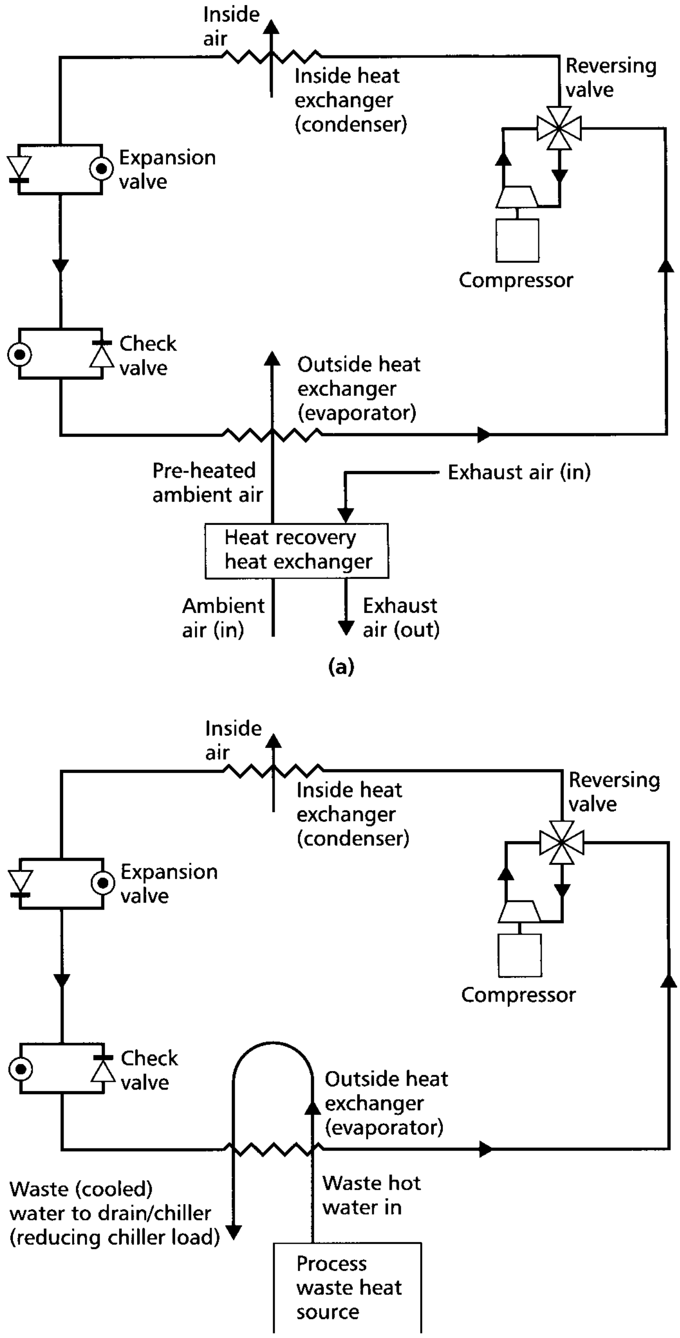


Figure 4.40 Schematic of heat pump systems incorporating heat recovery from (a) exhaust air and (b) process cooling water

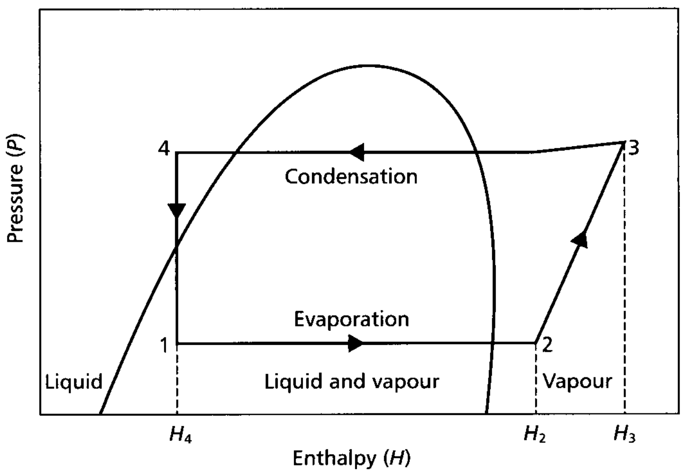


Figure 4.41 Pressure enthalpy diagram for the vapour compression cycle

Practical COP

The coefficient of performance 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 the following, in addition to the energy used by the compressor:

- outdoor fans or pumps required by the low temperature source
- crankcase heater.

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 to supplementary heating and distribution fans or pumps as part of the total energy input. It is not a true COP because items that are not part of the heat pump operation are also considered. It does, however, give an indication of the total energy used by the system compared to the heat output and thus enable an estimation of consumption and running costs to be established.

The seasonal coefficient of performance of a heat pump is defined as the appliance COP averaged over the heating season.

The values of coefficients of performance 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 4.7 shows the variation of cop values for a typical vapour compression air-to-air heat pump using ambient air as a source.

4.16.2.3 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.

4.16.2.4 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 or heating/cooling
- designing to suit energy source
- 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).

Table 4.7 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

4.17 Induction units

4.17.1 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.

4.17.2 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 4.42. Heating is normally provided either by a separate perimeter system or by electric heaters in the induction units. The use of electric reheaters 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

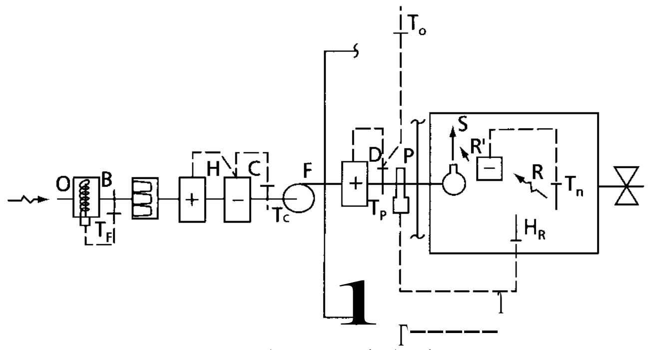


Figure 4.42 Two-pipe non-changeover induction system

cooling of heated ventilation air. Primary air temperatures are usually limited to a maximum of 45 °C.

— ***Four-pipe:*** induction units incorporate separate heating and cooling coils, fed by heating and chilled water circuits respectively[62]. 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 4.43. The elevated temperatures can improve the efficiency of the central cooling plant and provide increased opportunity for 'free cooling', see Figure 4.44.

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 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.

The induction units 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.

Sufficient access should be provided for maintenance, particularly for cleaning and inspection of the condensate drain pan.

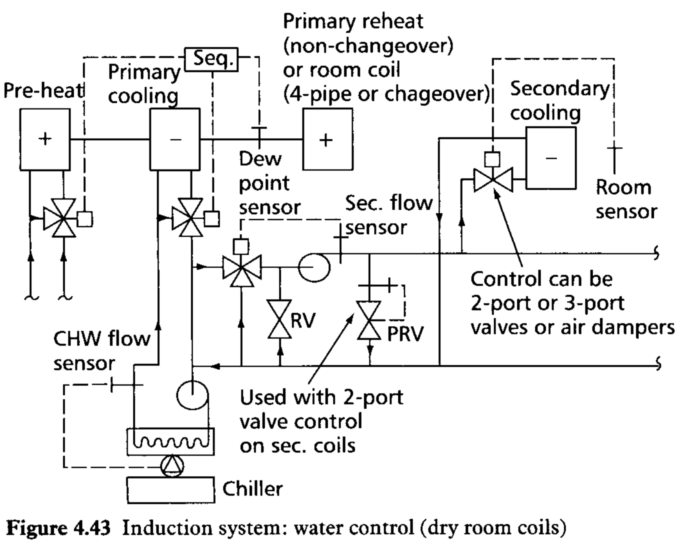


Figure 4.43 Induction system: water control (dry room coils)

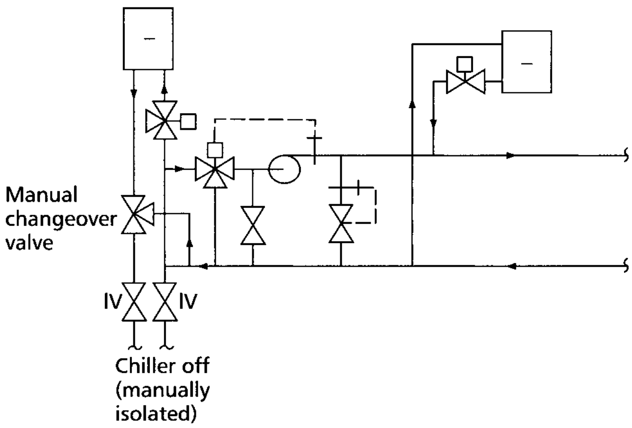


Figure 4.44 Induction system: utilisation of 'free cooling' in primary chilled water

4.17.3 Construction

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

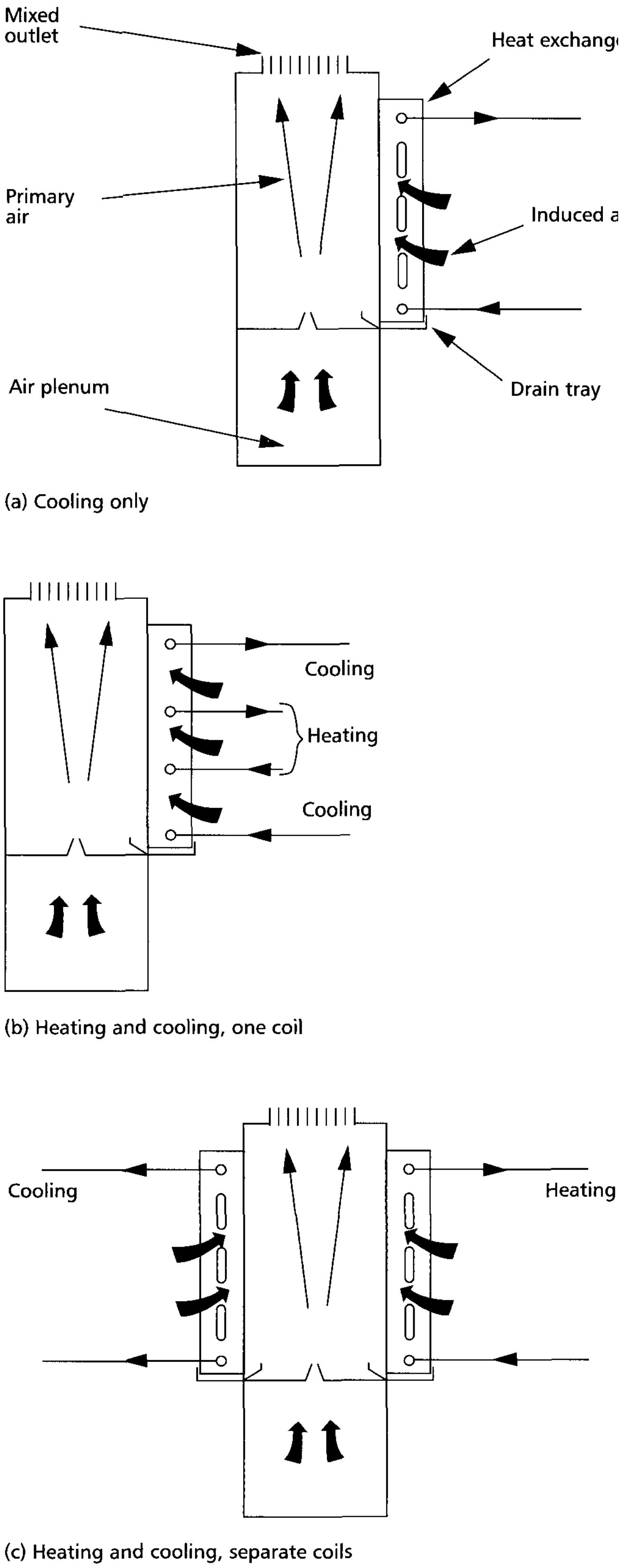


Figure 4.45 Induction units: alternative coil arrangements

4.18 Room air conditioners

4.18.1 General

Also known as window units and through-wall air conditioners, these are packaged units incorporating a room air-side 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 as an option. Where appropriate, moisture penetration may be minimised by the use of high efficiency louvres. Dust penetration may be minimised by the use of sand trap louvres.

The main advantage of room air conditioners 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.

Manufacturers' literature needs careful interpretation and corrections to ratings will normally be required to account for UK conditions.

4.18.2 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.

4.19 Single duct constant air volume systems

4.19.1 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 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 4.1, 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 5.6.

4.19.2 Design

4.19.2.1 Single-zone room control

The typical arrangement for a simple single zone system is shown in Figure 4.46. The temperature sensor T_R controls the cooling coil and reheater in sequence within its proportional band, see Figure 4.47. 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 4.48, 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 4.49, 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 (HRH) to override damper control, the reheater being brought in to deal with resultant overcooling. The cooling coil can be installed without a control device, provided that chilled water temperatures are maintained at an appropriate level.

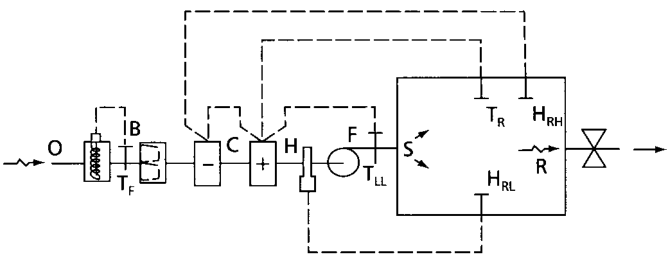


Figure 4.46 Full fresh air system with steam humidification; sequence control with humidity override

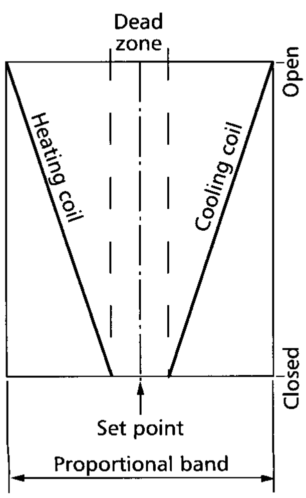


Figure 4.47 Sequential control of heating and cooling coils

Figure 4.50 shows a typical arrangement with recirculation. The temperature sensor T_R controls the cooling coil, mixing dampers and reheater in sequence within its proportional band (see Figure 4.51).

In the scheme shown in Figure 4.52, 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

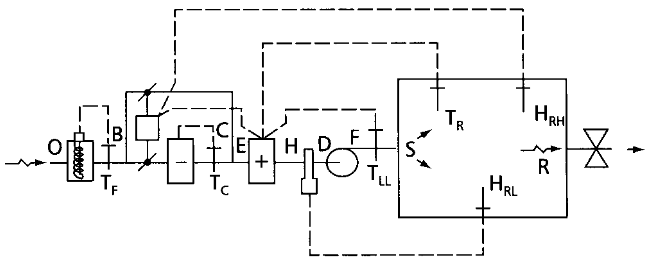


Figure 4.48 Full fresh air system with steam humidification; sequence control with face-and-bypass dampers

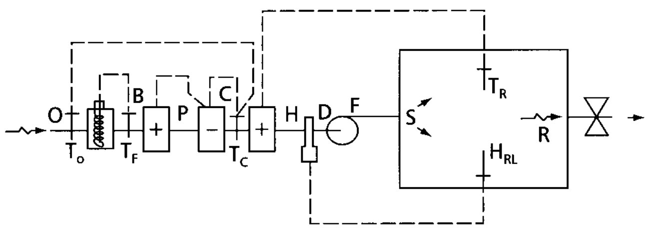


Figure 4.49 Full fresh air with steam humidification; sequence control with face-and-bypass dampers; psychrometric process

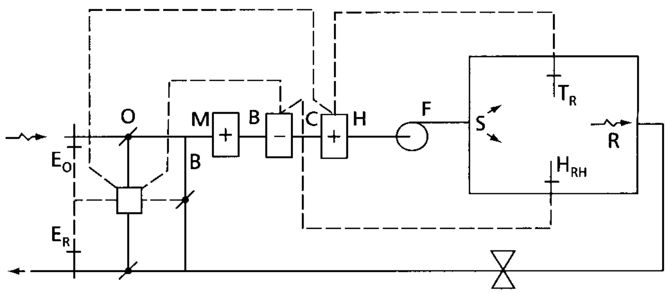
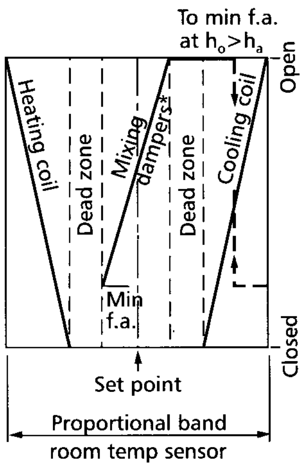


Figure 4.50 Recirculation: sequence control with humidity override



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

Figure 4.51 Sequential control for recirculation systems

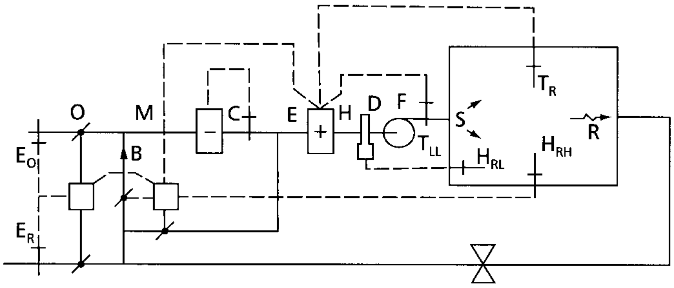
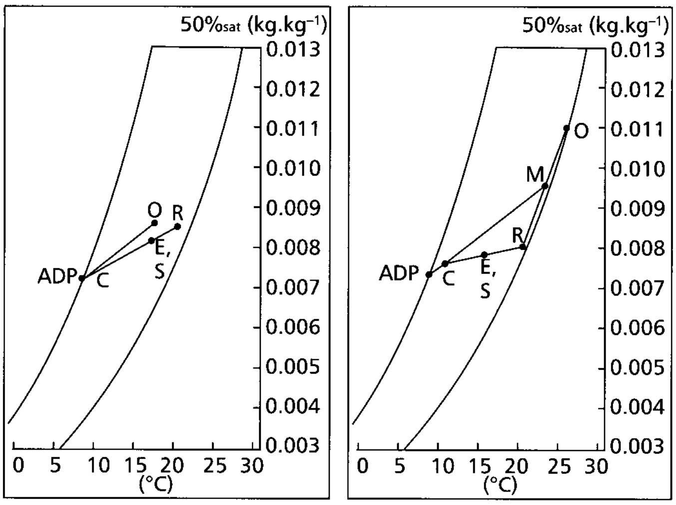


Figure 4.52 Recirculation with steam humidification: sequence control with room air bypass



(a) t_o = 18 °C (b) t_o = 27 °C

Figure 4.53 Recirculation with steam humidification: sequence control with room air bypass; psychrometric chart

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 4.53). Control is otherwise similar to face-and-bypass control.

4.19.2.2 Single-zone off-coil control

Figure 4.54 shows an arrangement in which the off-coil dry bulb temperature sensor T_C controls the cooling coil, pre-heater and reheater in sequence to achieve its set

if appropriate. Alternatively, the room temperature sensor T_R can be used to control the output of the reheater to

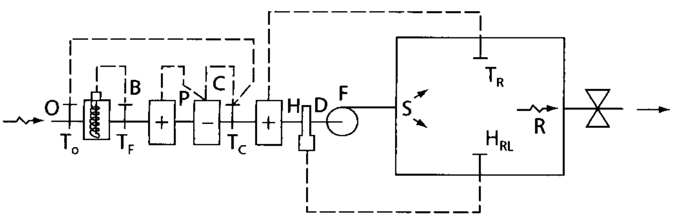


Figure 4.54 Full fresh air with steam humidification; off-coil control

lation. 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 4.51). 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

may be necessary to heat mixed air.

The pre-heater is optional, but if not present a low limit sensor should be provided to bring in the reheater to prevent cold draughts on start-up during wide load variations.

4.19.2.3 Multi-zone reheat system

Figure 4.56 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 which 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 overcooling by the centrally treated air, the room temperature sensor T_R brings in the respective zonal reheater.

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.

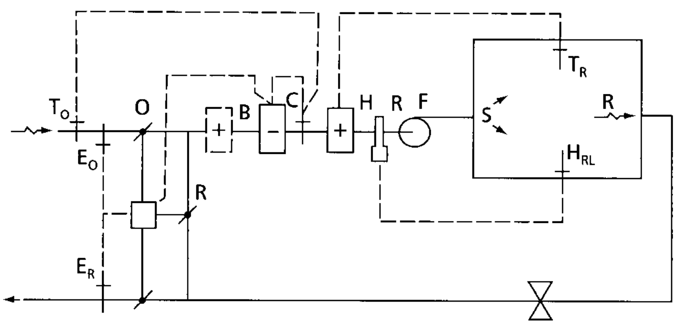


Figure 4.55 Recirculation; off-coil control

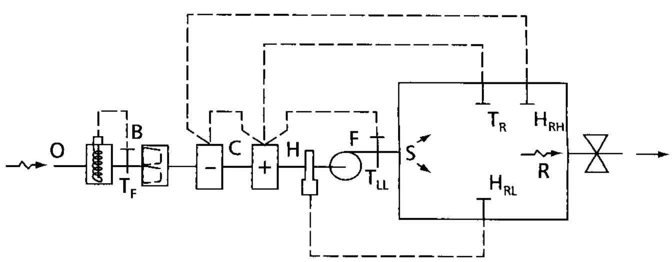


Figure 4.56 Terminal reheat

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'.

In order to reduce unnecessary reheat, control signals from the reheater 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.

4.19.3 Construction

See section 5 for equipment requirements.

4.20 Single duct variable air volume (VAV) systems

4.20.1 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.

4.20.2 Design

The control of VAV systems is considered in detail in CIBSE Guide H[62].

Varying the volume of air supplied to a space has the following consequences:

- its ability to offset sensible heat gains is reduced
- its ability to offset latent heat gains is reduced
- if the mixing ratio remains constant, its ability to dilute odours, carbon dioxide etc. is reduced
- unless special air terminal devices are utilised, its 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 4.57) 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 4.2.1. 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 4.58. 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.

The efficiency with turndown depends on:

- the position selected for sensing flow changes
- the mechanism employed for reducing total flow rate (see section 5.11)
- 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[62,88,89].

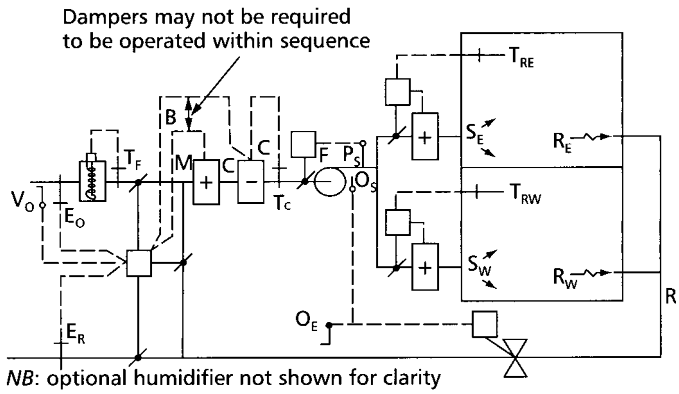


Figure 4.57 VAV with terminal reheat

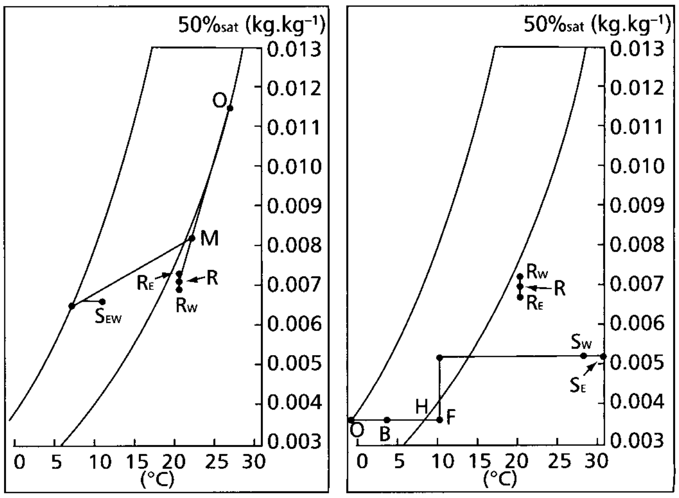


Figure 4.58 VAV with terminal reheat; psychrometric process

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:

(a) *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.

(b) *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.

Air handling unit (AHU) fans normally achieve variable volume by variable speed drive, variable pitch or inlet guide vane control, see section 5.11.

For perimeter zones where minimum loads fall below the potential cooling at full turndown, some means of heating will be necessary to avoid overcooling. ***(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.

4.20.2.1 Terminal reheat

To meet heating requirements reheater 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 reheaters are brought on. Compared to constant volume reheat, this reduces energy consumption as the amount of air being cooled and then reheated is reduced.

4.20.2.2 Perimeter heating

If significant perimeter down-draught is likely, under-window heating may be desirable, see Figure 4.59. 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 4.60 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.

4.20.2.3 Induction VAV— air terminal

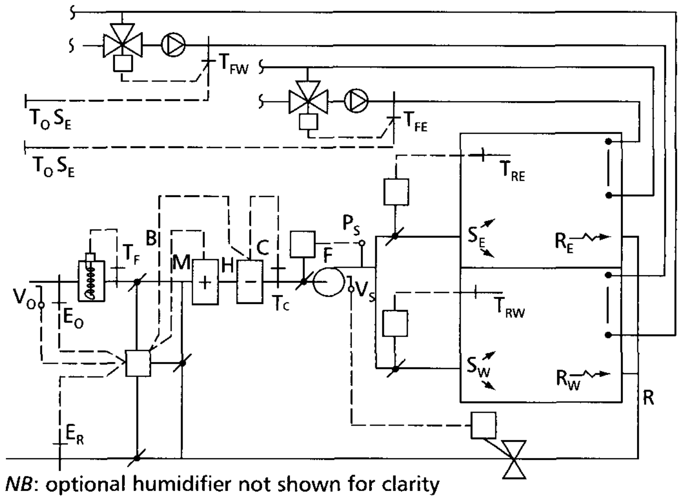
A separate constant volume primary air duct or system is used to encourage constant throw from supply air terminals. A separate source of primary air can be used to provide a constant fresh air supply, scheduled against outside air temperature as appropriate. Primary air is discharged at a constant volume 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.

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

4.20.2.5 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.



NB: optional humidifier not shown for clarity

Figure 4.59 VAV with perimeter heating

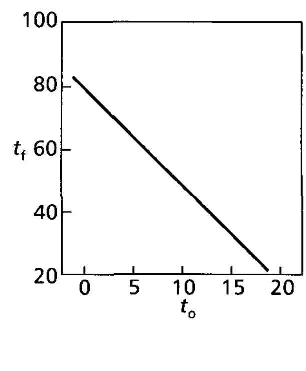


Figure 4.60 VAV with perimeter heating; resetting schedule for flow temperature sensors against outside air temperature

There are two arrangements in common use, whereby the fan and VAV damper are connected either in parallel or in series (see Figure 4.61(e) and 4.61(f) 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 preheat with the central plant held off.

4.20.3 Construction

See section 5 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 4.61 shows examples of these devices.

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.

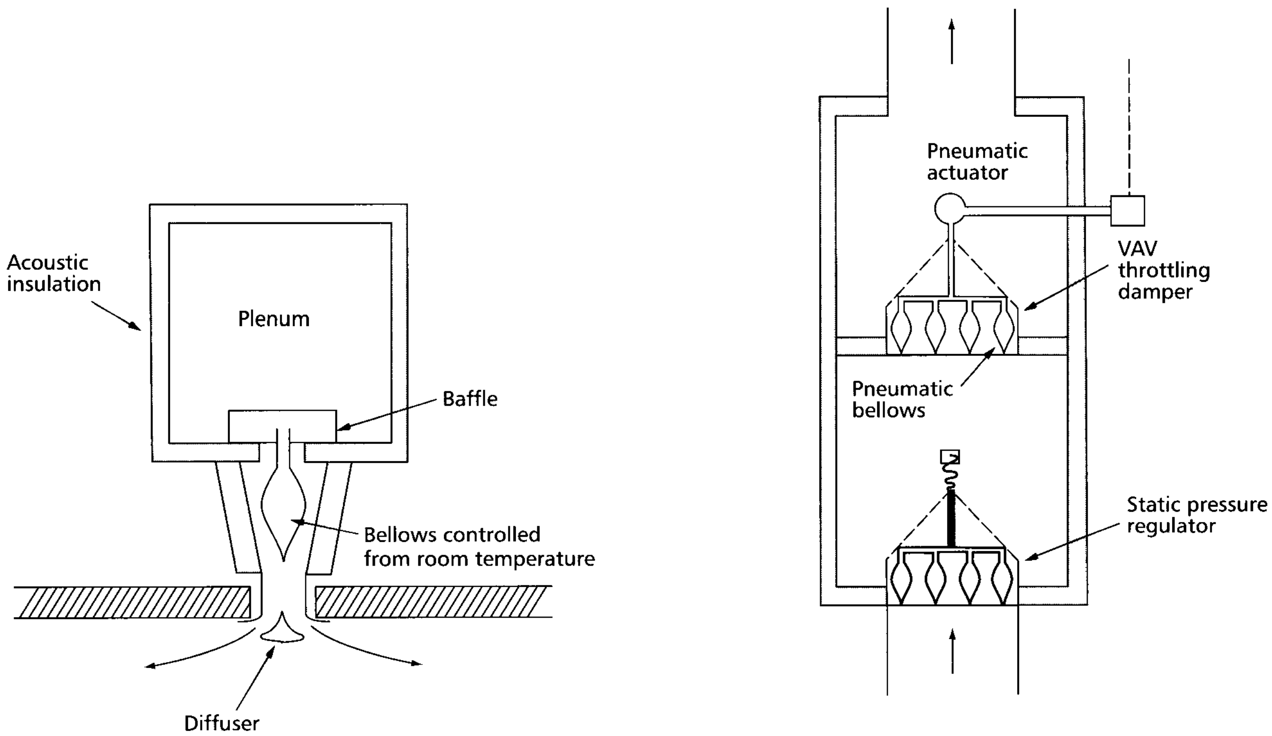
Most modern VAV systems use velocity reset VAV boxes. Primary air volume is set between minimum and maximum settings in relation to space temperature. These are pressure dependent and the system is essentially self-balancing.

VAV devices may be actuated mechanically, by means of a spring loaded regulator which 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.

4.21 Split systems

4.21.1 General notes

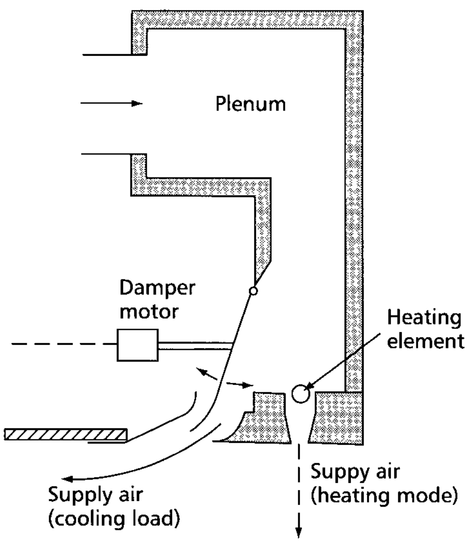
Split systems[88,90] are room air conditioner units, or small air handling units, incorporating a direct-expansion cooling coil, a filter and a fan to recirculate room air. They can be connected to a remote air, or water-cooled, condensing unit via low-pressure vapour and high-pressure liquid refrigerant lines. The external units are normally roof mounted and contain twin compressors, heat exchangers and air circulation fans. In cooling mode the external unit heat exchangers function as a refrigerant condenser producing liquid which 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.



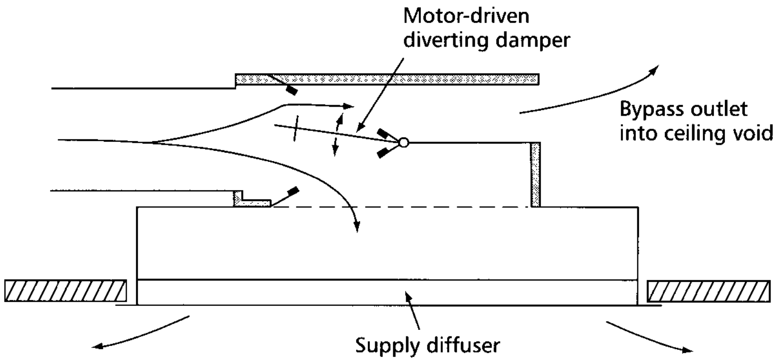
(a) Throtting air terminal device (variable velocity)

(b) Throttling control unit with static pressure regulator (mechanical)

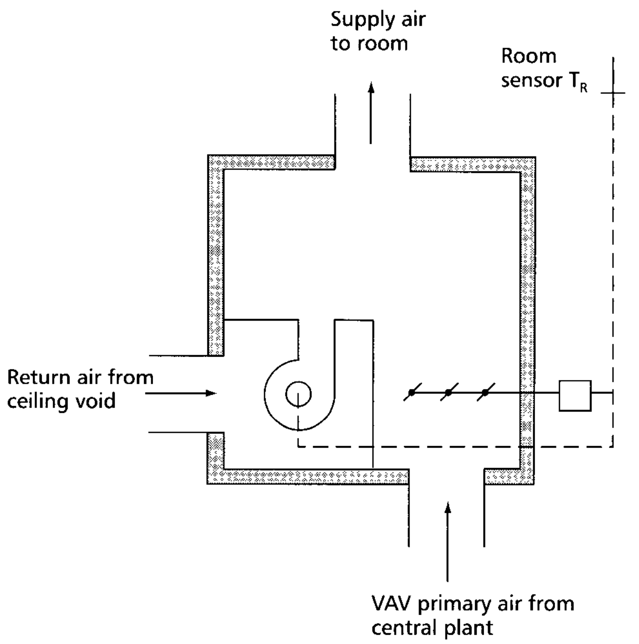
Figure 4.61 VAV devices



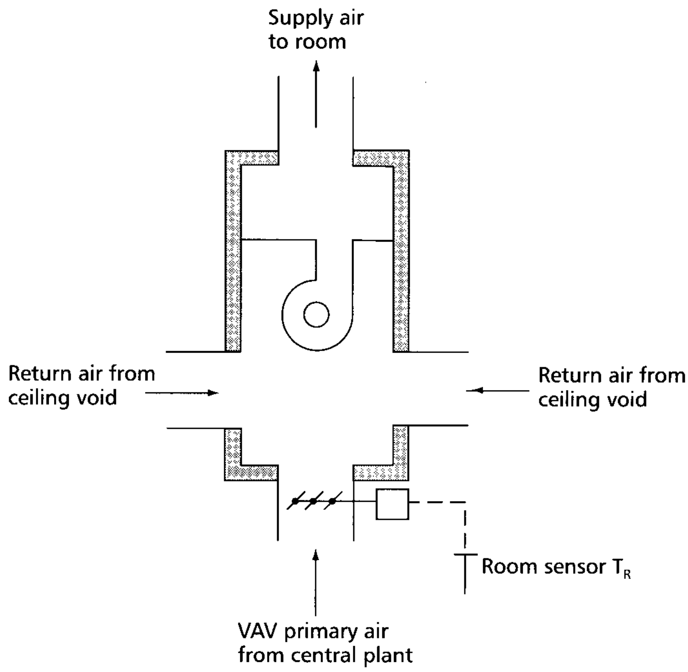
(c) Variable area air terminal device with reheat



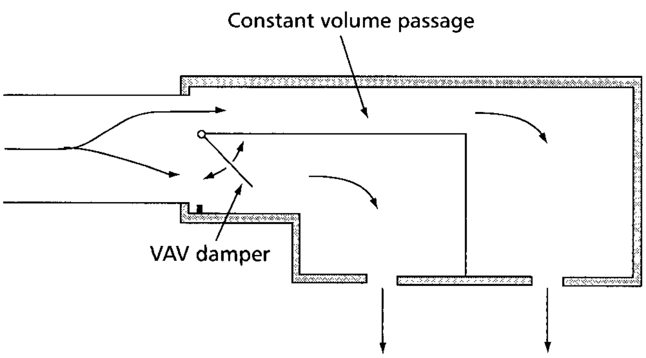
(d) Mechanical bypass air terminal device (slave diffusers possible)



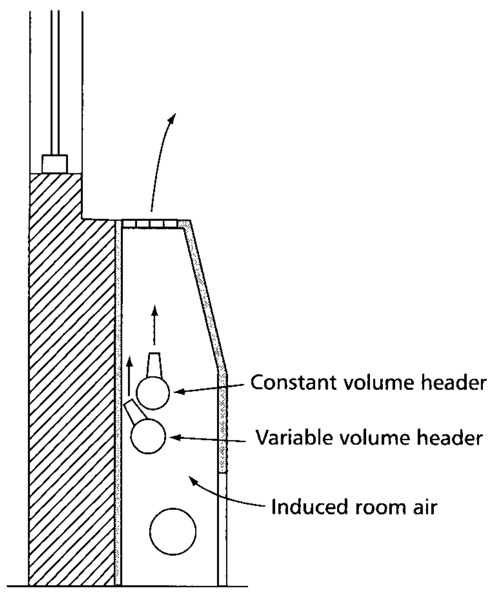
(e) Fan assisted control unit — parallel arrangement



(f) Fan assisted control unit — series arrangement



(g) Induction air terminal device using slots



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

Figure 4.61 VAV devices — *continued*

A three-pipe system can be installed to offer simultaneous heating and cooling within a building.

Applications include small commercial and retail premises.

4.21.2 Performance

The maximum capacity of an external unit is of the order of 30 kW. Up to eight room terminals, having outputs typically in the range 2.5-15 kW, may be served by one external unit. There is normally a 100 m limitation in the length of pipework between the external unit and the most remote room unit, with a maximum height difference of about 50 m.

4.21.3 Control

With smaller units, control can be achieved by switching the compressor. Larger direct expansion coils may incorporate refrigerant flow control or hot-gas bypass, possibly with multi-stage loading and unloading of reciprocating compressors.

4.21.4 Maintenance

Care must be taken in their design to ensure oil entrainment in the refrigerant lines. Appropriate refrigerant leakage detection measures must be put in place.

4.22 Sea/river/lake water cooling

4.22.1 Description

Water is pumped from the depths by an open loop system and cooling extracted via a heat exchanger, see Figure 4.62. 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.

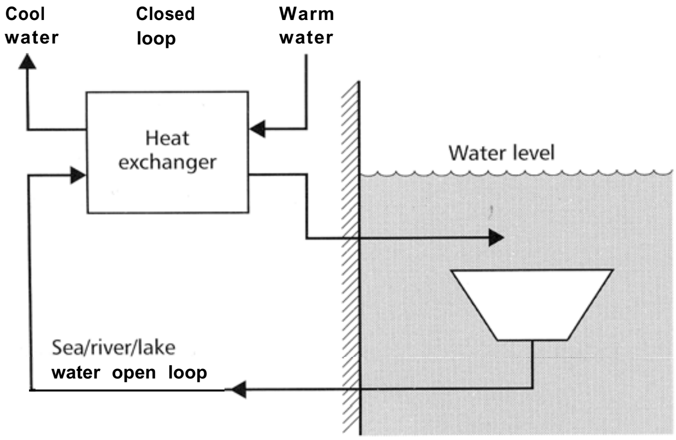


Figure 4.62 Schematic of sea/river/lake water cooling system

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.)

4.22.2 Design

Key design parameters include:

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

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. Equations for surface water heat transfer are provided in CIBSE Guide C, section 3[48].

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.

4.22.3 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.)

References

1 Environmental *design* CIBSE Guide A (London: Chartered Institution of Building Services Engineers) (1999)

2 *Energy efficiency in buildings* CIBSE Guide F (London: Chartered Institution of Building Services Engineers) (1998)

3 *Combustion systems* CIBSE Guide B13 (London: Chartered Institution of Building Services Engineers) (1986)

4 BS EN ISO *7730:* 1995: *Moderate thermal environments - Determination of the PMV and PPD indices* and *specification of the conditions for thermal comfort* (London: British Standards Institution) (1995)

5 Hwang C L et al. Optimal design of an air jet for spot cooling *ASHRAE Trans.* **90 (1B) 476** (Atlanta GA: American Society of Heating, Refrigerating and Air-conditioning Engineers) **(1984)**

6 Skaret E and Mathisen H M Ventilation efficiency - a guide to efficient ventilation *ASHRAE Trans.* **89** (2B) **480** (Atlanta GA: American Society of Heating, Refrigerating and Air-conditioning Engineers) **(1983)**

7 Flateheim G Air conditioning without draft and noise *Indoor Air* **5 1761 (1984)**

8 Skistad H Diffused air *Building ServicesJ.* **6 (9) 61** (September **1984)**

9 *Guide to air distribution technologyfor the internal environment* (Marlow: HEVAC Association) (2000)

10 Straub H E, Gilman S F and Konzo S *Distribution ofair within a roomforyear-round air conditioning* **-** *Parts **I** and* 2 Engineering Station Bulletin Nos. **435** and **442** (Urbana IL: University of Illinois) (July **1956** and March **1957** respectively)

11 Koestel H E and Tuve G L ASHRAE Research Reports **1553** and **1687** (Atlanta GA: American Society of Heating, Refrigerating and Air-conditioning Engineers) (dates unknown)

12 Miller P L and Nevins R G Room air distribution performance of ventilating ceilings and cone type circular ceiling diffusers *ASHRAE Trans.* **76 (1) 186** (Atlanta GA: American Society of Heating, Refrigerating and Air-conditioning Engineers) **(1970)**

13 Holmes M J *Designing variable volume systems for **room** air movement* ApP1ications Guide **AG1/74** (Bracknell: Bui1ding Services Research and Information Association) **(1973)**

14 Holmes M J and Sachariewicz E *The effect* of*ceiling beams and light fittings on ventilating jets* HVRA Laboratory Report **79** (Bracknell: Building Services Research and Information Association)**(1973)**

15 Holmes MJ *Thrcnuofverticallydischargedwarmairjets* Technical Note **3/76** (Bracknell: Building Services Research and Information Association) **(1976)**

16 Holmes M J and Caygill C *Air movement in rooms with low supply airflow rates* HVRA Laboratory Report **83** (Bracknell: Building Services Research and Information Association) **(1983)**

17 Jackman P J *Air movement in rooms with sidewall-mountedgrilles - a design procedure* HVRA Laboratory Report **65** (Bracknell: Building Services Research and Information Association) **(1970)**

18 Jackman P J *Air movement in rooms with sill-mounted diffusers* HVRA Laboratory Report **71** (Bracknell: Building Services Research and Information Association) **(1971)**

19 Jackman **P** J *Air movement in rooms with ceiling mounted diffusers* HVRA Laboratory Report **81** (Bracknell: Building Services Research and Information Association) **(1973)**

20 *Space air diffusion* Chapter **31** in ASHRAE Handbook: *Fundamentals* (Atlanta GA: American Society of Heating, Refrigerating and Air-conditioning Engineers) **(1997)**

21 ISO **3258: 1976:** *Air distribution and air diffusion - vocablary* (Geneva: International Standards Organisation) **(1976)**

22 Sodec F *Air distribution systems* Report **3554E** (Aachen: Krantz GmbH) **(1984)**

23 BS **5295:** *Environmental cleanliness in enclosed spaces:* Part *0:* **1989:** *General introduction, terms anddefinitionsfor clean rooms andcleanairdevices;* Part2: **1989:** *Methodforspecifjingthedesign, constructionandcommissioningofcleanroomsandcleanairdevices;* Part **3: 1989:** *Guide to operational procedures and disciplines applicable **to** clean rooms and clean air devices* (London: British Standards Institution) (dates as indicated)

24 *Clean work and work station requirements - controlled equipment* US Federal Standard **209B** (Washington DC: Superintendent of Documents) **(1966)**

25 Alamdari F and Eagles N *Displacement ventilation and chilled ceilings* BSRIA Technical Note TN2/96 (Bracknell: Building Services Research and Information Association) **(1996)**

26 *REHVA Guide to displacement ventilation in non-industrial premises* (Brussels: Federation of European Heating and Ventilating Associations (REHVA)) (http://www.rehva.com)

27 Sanberg M and Blomqvist C Displacement ventilation systems in office rooms *ASHRAE Trans.* **95 (2) 1041-1049** (Atlanta GA: American Society of Heating, Refrigerating and Air-conditioning Engineers) **(1989)**

28 Jackman P J *Displacement ventilation* BSRIA TM2/90 (Bracknell: Building Services Research and Information Association)**(1990)**

29 *Natural ventilation in non-domestic buildings* CIBSE Applications Manual AM10 (London: Chartered Institution of Building Services Engineers) **(1997)**

30 *Natural ventilation in non-domestic buildings* BRE Digest **399** (Garston: Building Research Establishment) **(1994)**

31 Jackman P J *Air distribution in naturally ventilated ojjices* BSRIA Technical Note TN **4/99** (Bracknell: Building Services Research and Information Association) **(1990)**

32 Walker B Rand White M K Single-sided natural ventilation **-** how deep an office? *Building Serv. Eng. Res. Technol.* **13 (4) 231-236(1992)**

33 Saxon R *Atrium buildings, design and development* (London: Longman) **(1986)**

34 *Testing buildings for air leakage* CIBSE TM23 (London: Chartered Institution of Building Services Engineers) (2000)

35 *Minimising pollution at air intakes* CIBSE TM21 (London: Chartered Institution of Building Services Engineers) **(1999)**

36 BS **8233: 1999:** *Sound insulation and noise reductionfor buildings - Code of practice* (London: British Standards Institution) **(1999)**

37 *Sound control* CIBSE Guide B5 (London: Chartered Institution of Building Services Engineers) (in preparation)

38 Willis S, Fordham M and Bordass W *Avoiding **or** minimising the use ofair conditioning* EEBPP General Information Report GIR **31** (London: Department of Transport, Local Government and the Regions) **(1995)** (available from http://www.energy-efficiency.gov.uk)

39 BS 5588: *Fire precautions in design, construction and use of buildings* **(10** parts) (London: British Standards Institution) (various dates)

40 *Fire engineering* CIBSE Guide E (London: Chartered Institution of Building Services Engineers) **(1997)**

41 Morgan H P, Ghosh BK, Garrad G, Pamlitschka R P, De Smedt J-C and Schoonbaert L R *Design methodologies for smoke and heat exhaust ventilation* BRE Report BR **368** (Garston: Building Research Establishment) **(1999)**

42 *The Building Regulations 1991 Approved Document B: Fire Safety* (London: Department of Transport, Local Government and the Regions/The Stationery Office) (2000)

43 Martin A J *Control ofnatural ventilation* BSRIA Technical Note **TN11/95** (Bracknell: Building Services Research and Information Association) **(1995)**

44 *Automatic controls* CIBSE Commissioning Code C (London: Chartered Institution of Building Services Engineers) **(2001)**

45 Orme M, Liddament M W and Wilson A *An analysis and summary oftheAZVC's numerical database* AIVC Technical Note **44** (Coventry: Air Infiltration and Ventilation Centre) **(1994)**

46 *A guide to energy efficient ventilation* (Coventry: Air Infiltration and Ventilation Centre) (1996)

47 *Space requirements for plant access, operation and maintenance* Defence Estate Organisation (Works) Design and Maintenance Guide **08** (London: Ministry of Defence/The Stationery Office) (1996)

48 *Reference data* CIBSE Guide C (London: Butterworth-Heinemann/Chartered Institution of Building Services Engineers) (2001)

49 *Energy efficient mechanical ventilation systems* EEBPP Good Practice Guide GPG 257 (London: Department of Transport, Local Government and the Regions) (1999) (available from http://www.energy-efficiency.gov.uk)

50 *A practical guide to air leakage testing* HVCA DW143 (London: Heating and Ventilating Contractors' Association) (2000)

51 *Guide to good practice - Cleanliness of ventilation systems* HVCA TR17 (London: Heating and Ventilating Contractors' Association) (2000)

52 *Hygienic maintenance of office ventilation ductwork* CIBSE TM26 (London: Chartered Institution of Building Services Engineers) (2000)

53 *Variable flow control* EEBPP General Information Report GIR 41 (London: Department of Transport, Local Government and the Regions) (1996) (available from http://www.energy-efficiency.gov.uk)

54 *The Building Regulations 2000 Approved Document L: Conservation of fuel and power* (interim draft) (London: Department of Transport, Local Government and the Regions) (March 2001)

55 *Energy use in offices* EEBPP Energy Consumption Guide ECON 19 (London: Department of Transport, Local Government and the Regions) (2000) (available from http://www,energy-efficiency.gov.uk)

56 *Mixed mode ventilation* CIBSE AM13 (London: Chartered Institution of Building Services Engineers) (2000)

57 *Minimising and avoiding the need for air conditioning* EEBPP General Information Report **GIR** 31 (London: Department of Transport, Local Government and the Regions) (1995) (available from http://www.energy-efficiency.gov.uk)

58 Kendrick C, Martin A et al. *Refurbishment of air conditioned buildings for natural ventilation* BSRIA Technical Note TN8/98 (Bracknell: Building Services Research and Information Association) (1995)

59 *Selecting air conditioning systems - a guide for building clients and their advisers* EEBPP Good Practice Guide GPG 71 (Garston: Building Research Energy Conservation Support Unit) (1993) (London: Department of Transport, Local Government and the Regions) (1993) (available from http://www.energy-efficiency.gov.uk)

60 Barnard N and Jauntzens D (eds.) *Low energy cooling - Technologies selection and early design guidance* (London: Construction Research Communications) (2001)

61 Brittain J R J *Oversized air handling plant* BSRIA Guidance Note GN11/97 (Bracknell: Building Services Research and Information Association) (1997)

62 *Building control systems* CIBSE Guide H (London: Chartered Institution of Building Services Engineers) (2000)

63 de Saulles T *Free cooling systems design and application guide* BSRIA RR16/96 (Bracknell: Building Services Research and Information Association) (1996)

64 *Air conditioned buildings* CIBSE Building Energy Code **2** (London: Chartered Institution of Building Services Engineers) (1999)

65 *Energy estimating and modelling methods* Chapter 30 in ASHRAE Handbook: *Fundamentals* (Atlanta GA: American Society of Heating, Refrigerating and Air-conditioning Engineers) (1997)

66 (various titles) Energy Efficiency Best Practice Programme (EEBPP) Energy Consumption Guides (London: Department of Transport, Local Government and the Regions) (various dates) (available from http://www.energy-efficiency.gov.uk)

67 Barnard N *Dynamic energy storage in the building* BSRIA Technical Report TR9/94 (Bracknell: Building Services Research and Information Association) (1994)

68 Braham D, Barnard N and Jauntzens D *Thermal mass in office buildings* BRE Digest 454 Part 1 (Garston: Building Research Establishment) (2001)

69 Holman J P *Heat transfer* (New York NY: McGraw-Hill) (1986) ISBN 0 07 Y66459-5

70 *Environmental floor systems* (Ascot: Steel Construction Institute) (1997)

71 *Modelling the performance of thermal mass* BRE Information Paper IP 6/2000 (Garston: Building Research Establishment) (2000)

72 Fletcher J and Martin A *J Night cooling control strategies* BSRIA Technical Appraisal TN14/96 (Bracknell: Building Services Research and Information Association) (1996)

73 Fletcher J *Pre-cooling in mechanically cooled buildings* BSRIA TN 16/95 (Bracknell: Building Services Research and Information Association) (1995)

74 Peason C C and Barnard N *Guidance and the standard specification for thermal imaging of non-electrical installations* BSRIA Facilities Management Specification FMS6/00 (Bracknell: Building Services Research and Information Association) (2000)

75 Arnold D Chilled beams in naturally ventilated buildings *Proc. CIBSE/ASHRAE Joint Nat. Conf., Harrogate, September 1996* (1) 333-338 (London: Chartered Institution of Building Services Engineers) (1996)

76 Abbas T *Displacement ventilation and static cooling devices* BSRIA COP 17/99 (Bracknell: Building Services Research and Information Association) (1994)

77 Butler D J G Chilled ceilings - free cooling opportunity *Proc. CIBSE Nut. Conf, Bournemouth, October 1998* 273-279 (London: Chartered Institution of Building Services Engineers) (1998)

78 Davies G A model performance *Building Serv. J. 16* (6) 29-30 (June 1994)

79 Martin A and Alamdari A Condensation control for chilled ceilings and beams *Proc. CIBSE Nat. Conf, Alexandra Palace, October* 1997 *1* 45-52 (London: Chartered Institution of Building Services Engineers) (1997)

80 *Panel heating and cooling* Chapter 6 in ASHRAE Handbook: *HVAC Systems and Equipment* (Atlanta GA: American Society of Heating, Refrigerating and Air-conditioning Engineers) (2000)

81 Warwicker B Low humidity air and air conditioning *Building Serv. 3. 17* (1 1) 4749 (November 1995)

82 *The control of legionellosis including Legionnaires disease* Technical Guidance Note HS(G)70 (London: Health and Safety Executive) (1991)

83 *Minimising the risk of Legionnaires' disease* CIBSE TM13 (London: Chartered Institution of Building Services Engineers) (2000)

84 *Non-residential cooling and heating load calculations* Chapter 28 in ASHRAE Handbook: *Fundamentals* (Atlanta GA: American Society of Heating, Refrigerating and Air-conditioning Engineers) (1997)

85 Barnard N and Jaunzens D (eds.) Low ***energy cooling - technologies selection and early design guidance*** IEA Energy Conservation in Buildings and Community Systems Programme **EP56** (London: Construction Research Communications) **(2001)** ISBN **186081 4581**

86 Bunn R Ground coupling explained ***Building Serv.*** 3. 20 **22-24** (December **1998)**

87 Zimmermann M and Andersson J (eds.) ***Low energy cooling - case study buildings*** IEA Energy Conservation in Buildings and Community Systems Programme (St Albans: Oscar Faber Group) **(1999)**

88 Jones W P ***Air Conditioning Engineering*** (4th ed.) (London: Edward Arnold) **(1994)**

89 Hittle D C et al. Theory meets practice in a full-scale HVAC *1aboratoryASHRAE J.* 24 **(1** 1) **36** (November **1982)**

90 Martin P L and Oughton D L ***Faber and Kell's Heating and Air Conditioning of Buildings*** (London: Buttenvorth-Heinemann) **(1995)**

Appendix 4.A1: Techniques for assessment of ventilation

4.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[A1.1] and AM10[A1.2] provide more detail on dynamic thermal simulation and assessment techniques for natural ventilation respectively.

4.A1.2 Ventilation and cooling

Section 3.2.1 includes airflow rate requirements for ventilation purposes. Airflow 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. Airflow 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*[A1.3]), although the user can work only within the range of variables covered by the charts. Section 5 of CIBSE Guide A[A1.4] 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[A1.2]. 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 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[A1.5].

Thermal mass should be modelled with appropriate surface heat transfer values and representation of heat flow within the mass (refer to section 4.7.2.1). 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.

4.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.

4.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.

4.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(A1.6) 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(A1.7).

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(A1.8).

References

A1.1 *Building energy and environmental modelling* CIBSE Applications Manual AM11 (London: Chartered Institution of Building Services Engineers) (1998)

A1.2 *Natural ventilation in non-domestic buildings* CIBSE Applications Manual AM10 (London: Chartered Institution of Building Services Engineers) (1997)

A1.3 Rennie D and Parand F *Environmental design guide* for *naturally ventilated and daylit offices* BRE Report BR 345 (Garston: Building Research Establishment) (1998)

A1.4 *Environmental design* CIBSE Guide A (London: Chartered Institution of Building Services Engineers) (1999)

A1.5 Dubrul C *Inhabitant’s behaviour with regard to ventilation* AIVC Technical Note 23 (Coventry: Air Infiltration and Ventilation Centre) (1988)

A1.6 Liddament M W *Air Infiltration Development Algorithm (AZDA)* Appendix 3 in *A guide to energy efficient ventilation* (Coventry: Air Infiltration and Ventilation Centre) (1996)

A1.7 Feustel H E and Dieris A *Survey of airflow models for multi-zone structures* (Berkeley, CA: Earnest Orlando Lawrence Berkeley National Laboratory) (1991)

A1.8 Kendrick J *An overview of combined modelling of heat transport and air movement* AIVC Technical Note 40 (Coventry: Air Infiltration and Ventilation Centre) (1993)

Appendix 4.A2: Psychrometric processes

Table 4.8 illustrates the basic psychrometric processes and lists the equipment concerned. See section 5 for details of the various items of equipment.

Table 4.8 Basic psychrometric processes

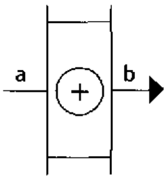
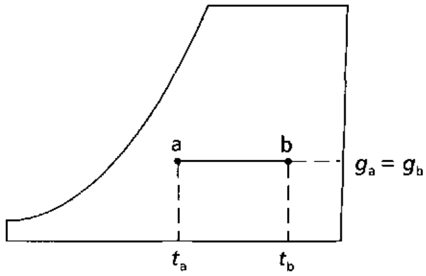
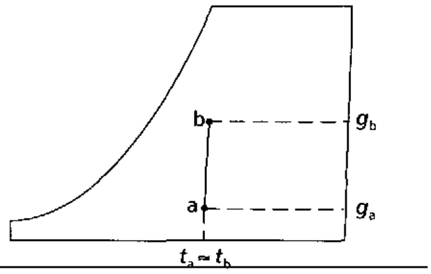
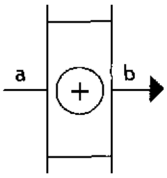
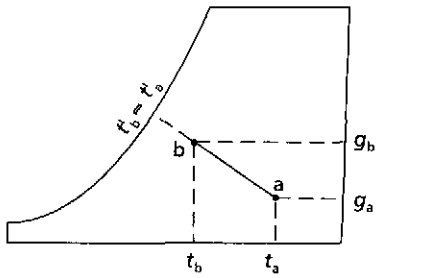
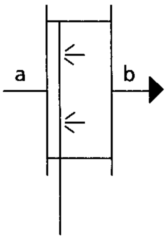
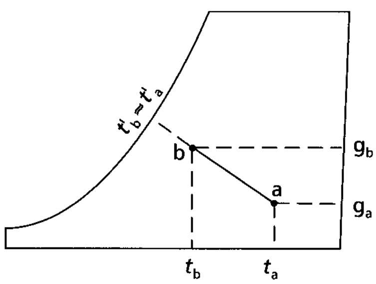
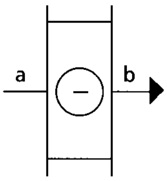
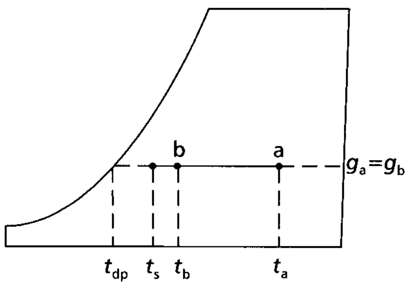
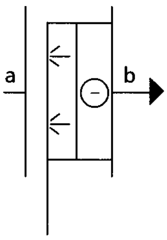
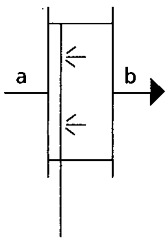
Process	Method	Remarks	Psychrometric process
Heating	Electric	No additional plant required. High energy costs. Wiring and switch gear costs high for large duties. Usually only step control available.	
	Steam	Small heat transfer surface. Plant cost high unless steam required for other services. Condensate 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-reciculatory 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 4.8 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 usedin 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}.	
	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}.	

5 Equipment

5.1 Introduction

This section provides information about a wide range of equipment that is required for ventilation and air conditioning systems. It sets out critical design issues relating to the specific items of equipment and the key points to be considered in the selection of equipment. Where relevant, it provides references to key regulations and guidance relating specifically to the design, installation or use of the equipment.

5.2 Ventilation 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 within the building. Bird screens and insert mesh should be used to prevent entry by birds or other large objects.

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

- traffic
- boiler flues and exhausts from standby generators (or combined heat and power engines)
- cooling towers and other heat rejection plant
- vents from plumbing, oil storage tanks etc.
- ventilation exhausts from fume cupboards, kitchens, toilets, car parks, print rooms
- 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 litter might accumulate
- radon gas.

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

Whilst wet cooling towers give rise to the greatest health concern because of the potential risk of ***Legionnella,*** other heat rejection equipment 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 'short-circuiting'. 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 short-circuiting. Locating the intake and discharge on different facades can also help to reduce the risk. 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 pressures are low. The influence of wind pressures can be reduced by:

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

Measures that should be considered to minimise re-entry from contaminated sources include[1]:

- group exhaust to increase plume rise due to the greater momentum of the combined exhaust
- place inlets on roof where wind pressures will not vary greatly with direction to ensure greater system stability
- avoid locating exhaust outlets within enclosures or architectural screens that may hold contaminants within areas of flow recirculation
- discharge exhausts vertically
- 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)
- avoiding locating inlets and exhausts near edges of walls or roofs due to pressure fluctuations.

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. A European Directive[2] gives mandatory air quality standards for smoke and sulphur dioxide, see also section 3.2.1. A vertical discharge stack, capable of imparting a high efflux velocity to the exhaust, may be required. If so, provision must be made for handling rainwater and avoiding corrosion. Industrial processes resulting in polluting emissions to air, water or land come under the requirements of the Environmental Protection Act[3].

Sections 1.6.4 and 1.6.5 of CIBSE Guide A[4] provide guideline values for pollutants and guidance on filtration strategy, respectively. Reference should also be made to CIBSE TM21[5] for more detailed guidance on pollution sources and assessment methods.

5.3 Natural ventilation devices

5.3.2 Openable window design

5.3.2.1 Window performance testing

BS EN 12207[6], which partially replaces **BS** 6375: Part 1[7], classifies window and door performance according to their permeability. Reference air permeabilities are recorded for each class of window related to the permeability of both the overall area and of the opening joint. These are defined at a test pressure of 100 Pa. **BS EN** 12207 describes how limits can be defined for other test pressures and how windows are subsequently classified according to the relationship between the two permeability assessments.

Figure 5.1 shows the upper limits of each class, which are derived from the reference air permeabilities at 100 Pa related to the overall area and length of opening joint, see Table 5.1.

5.3.2.2 Required window functionality

General information on window design and selection is available from other **CIBSE** publications[8,9]. There are a number of important criteria, which are outlined in the following sections.

Ventilation capacity

The ventilation capacity is the amount of air that will flow through a given window area of different designs. It depends on the ratio of the effective open area to the facade area of the window unit. Ventilation capacity will be maximised by increasing the vertical separation and magnitude of those open areas. This will in turn depend on the way the window opens (i.e. side, topbottom, centre pivot, sliding etc.), and the distribution of the open area over the vertical height of the window. Figure 5.2 shows

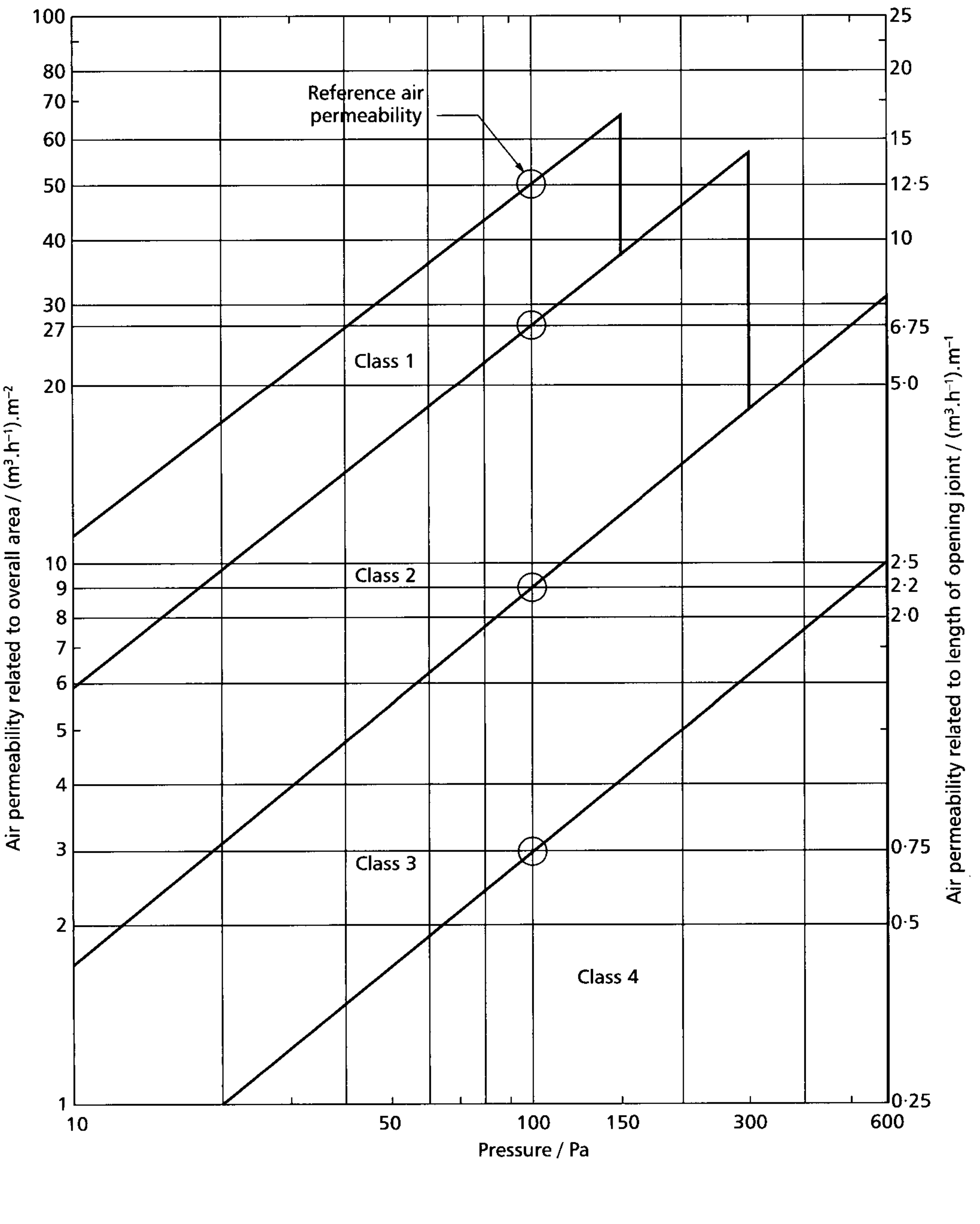


Figure 5.1 Classification of doors and windows by air permeability (reproduced from BS EN 12207 by permission of the British Standards Institution)

Table 5.1 Reference air permeabilities at 100 Pa and maximum test pressures related to overall area and length of opening joint[6]

Class	Reference air permeability at 100 Pa and maximum test pressure			
	Related to overall area		Related to length of opening joint	
	Permeability / $(m^3 \cdot h^{-1}) \cdot m^{-2}$	Max. test pressure / Pa	Permeability / $(m^3 \cdot h^{-1}) \cdot m^{-1}$	Max. test pressure / Pa
0*	—	—	—	—
1	50	150	12.50	150
2	27	300	6.75	300
3	9	600	2.25	600
4	3	600	0.75	600

* Not tested

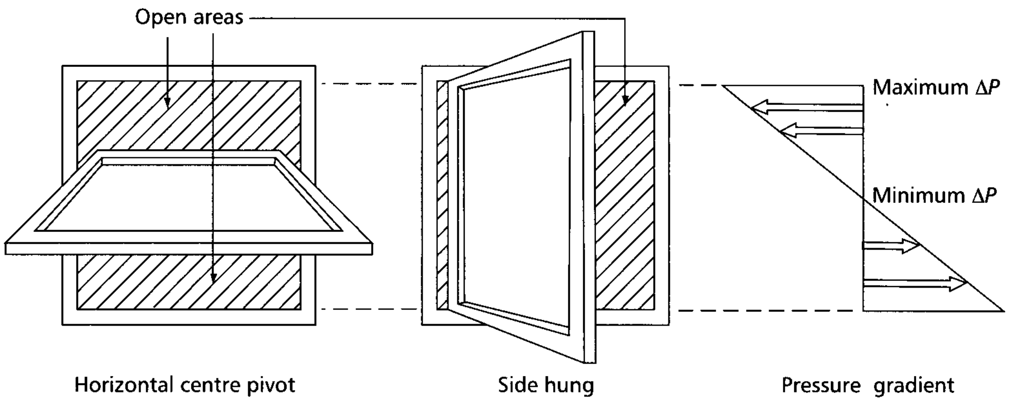


Figure 5.2 Ventilation capacity of different window configurations

the open areas for a horizontal centre pivot window compared to a side hung window. A typical pressure gradient caused by inside-outside temperature differences is also shown. The centre pivot window has a much higher ventilation capacity because the open area is concentrated at regions of high-pressure difference. In contrast, much of the open area of side hung windows is in a region of small pressure difference.

Controllability

Good control at small openings is particularly important for winter comfort. The flow characteristic is influenced by the mode of opening the window and factors such as the shape and thickness of the window frame. Figure 5.3 illustrates that the effective open area of a window may not increase very rapidly until the opening angle is quite large.

Impact on comfort

The position of the room air inlet will have an effect on comfort factors such as draughts. Air entering the space at the occupied level can improve comfort in summer, when the air movement will provide a cooling benefit. In winter when the entering air is much colder, the same opening may result in discomfort from draughts. Consequently, separate winter openings may be preferred (either separate high-level windows or trickle ventilators). To avoid high summer ventilation rates (causing papers to be disturbed), the height of that part of the window where air enters the space should be above desk level, by say 150 mm.

Thermal contact

In strategies utilising night cooling and thermal capacity, the ventilation air needs to be able to make good thermal contact with the fabric in order to effect good heat transfer.

Security

The implications of open windows, particularly in night ventilation mode, need to be considered. Some window designs can be lockable in a part-open position which allows adequate night ventilation rates but which minimises the risk of intruders gaining access to the building.

Integration with solar control strategies (particularly blinds)

There are a number of ways in which the blind and window opening may interact. These include:

- the physical movement of the window may be restricted by an independent internal (or external) blind; this is mainly a problem for pivoting windows
- with pivoting windows and mid-pane blinds, there is the impact on shading performance when the

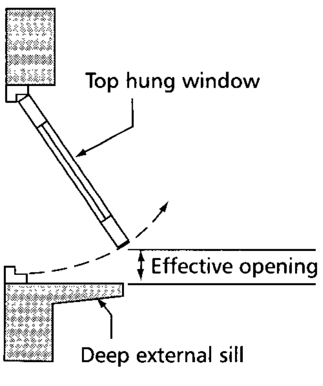


Figure 5.3 Effect of a sill on the effective open area of a window

angle of the blind louvres to the incident radiation changes as the window is opened

- the effect of the blind in providing a resistance to airflow; the blind elements (unless they are mid-pane) will provide an obstruction to the free area of the opening (this is independent of window type, see Pitts and Georgiadis[10]).

Maintenance and cleaning of windows

Maintenance is an important feature; can the window be cleaned from the inside?

5.3.2.3 Window specification

Information on the performance characteristics of various window types, see Figure 5.4, is given below. The effect of these different characteristics should be assessed with reference to the criteria listed above.

Horizontal pivot windows

These windows have a high ventilation capacity because large open areas are created at a separation equivalent to the window height. With single-sided ventilation, air will enter at the lower level and exit via the top of the window. An opening of 22 degrees is usually considered the norm for 'fully open', and for a typical 1200 mm wide by 1600 mm high window this results in an effective open area of 0.66 m^2. They are easily adjustable to provide control of the ventilation rate.

Maximising the height of the top of the window in the room will help exhaust warmer air at ceiling level when operating in single-sided ventilation mode. Glazing at high level will also promote good levels of natural light deep into the space.

When operating in wind-driven cross-ventilation mode, air will enter at the top and bottom of the window. The air entering through the top gap will be directed upward and this can improve thermal contact with exposed ceilings for effective night ventilation. Solar radiation striking the opaque surfaces of the wall or the ground adjacent to the facade can generate rising convection currents. These can be deflected into the room if the outward projection of the window extends beyond the window reveal.

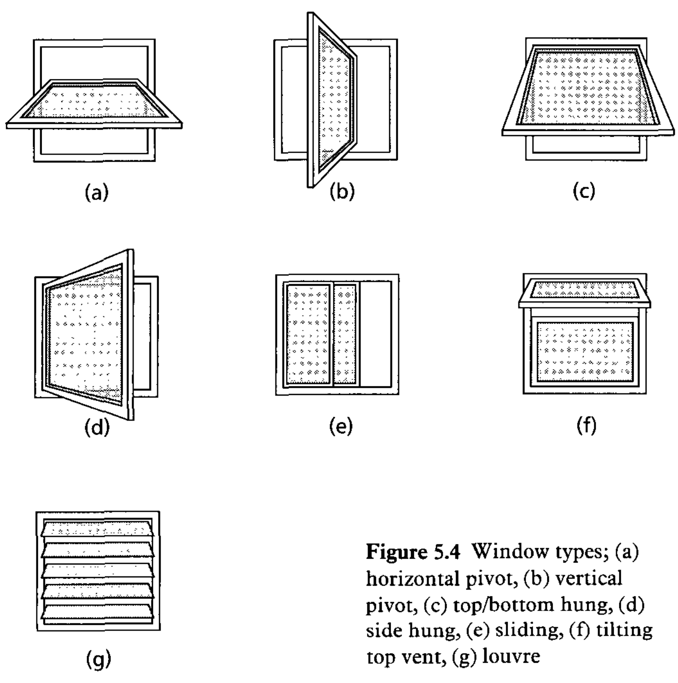


Figure 5.4 Window types; (a) horizontal pivot, (b) vertical pivot, (c) top/bottom hung, (d) side hung, (e) sliding, (f) tilting top vent, (g) louvre

Vertical pivot windows

Because the opening is distributed uniformly over their height, these windows have a lower ventilation capacity. For the same 22 degree opening, the effective open area is reduced by 40% relative to the horizontal pivot.

Vertical pivot windows can act as a form of 'wind scoop' for wind directions parallel to the face of the building. Because they have a large vertical opening, they are more likely to allow rain into the building. Carefully designed weather-stripping is required for both horizontal and vertical centre pivot windows to achieve a good performance in winter.

Top/bottom hung windows

As ventilators, these are less effective still, since all the opening area is concentrated at one end of the window height. The effective open area is about 35% of the horizontal pivot type. Depending on where the opening is, the summer ventilation will either provide cooling to the occupant and poor thermal contact with the ceiling, or vice versa. Top hung windows can act as scoops for warm air rising up the outside of the building (e.g. from convection currents generated by solar gain absorbed on building surfaces).

Side hung windows

These are similar in performance terms to vertical pivot windows. Because of the greater distance from window edge to pivot (and hence greater turning moment), they are more susceptible to being blown by gusts of wind.

Inward opening windows can cause clashes with furniture positions. The combination of top hung winter ventilators and side hung summer windows (with effective weather-stripping) provides good all-round performance. The top hung winter ventilator can also provide a secure opening for summer ventilation that, in combination with the side hung opening, will enhance stack effect.

Tilt and turn windows

Tilt and turn windows are a combination unit offering bottom and side hung options (although the side hung mode is mainly intended for cleaning purposes). A study of several buildings by Willis et al.[11] suggests that the tilt setting provides too much ventilation in winter and insufficient in summer. The turn mode can cause clashes with furniture.

Sliding windows (including sash)

Depending on whether they are vertical sliding (sash) or horizontal sliding windows, these will have similar ventilation characteristics to the horizontal and vertical pivot window respectively. Sliding windows can provide good control over summer ventilation. Sash windows allow the stack effect to be controlled through adjustment of the opening size at both the top and bottom of the window. However, ensuring a good seal in the closed position requires particular attention. This is important in terms of reducing draughts and energy losses in winter. Recent designs have significantly improved the performance of sliding windows in this respect. The design of sash windows needs to be such as to facilitate easy opening of the upper sash.

Tilting top vents

These units provide smaller opening areas than the other systems, because the opening portion occupies only a proportion of the window height. However they can provide good draught-free ventilation, especially in cross-ventilation mode. If the vent is bottom hung, opening inwards, (the 'hopper' window), the natural flow pattern may encourage good thermal contact with the ceiling. However care must be taken to protect the opening from driving rain.

Whereas windows perform many different functions, sections 5.3.3 to 5.3.7 describe openings in the facade whose sole purpose is to provide ventilation. Note that any such devices should offer a very low resistance to airflow as the driving forces for natural ventilation may only be in the region of 10 Pa. Further guidance on product development and natural ventilation design tools is available from BRE[12].

5.3.3 Air bricks and trickle ventilators

Air bricks incorporate no provision for control of infiltration rate. Automatic ventilators, which provide nominally constant infiltration under variable wind velocities, should be considered as an alternative.

The concept of 'build tight, ventilate right' is increasingly recognised as the basis of good design for ventilation. This relies upon an airtight fabric and the provision of a means of controlled background ventilation. In a naturally ventilated building this is often provided by trickle ventilators with higher rates of ventilation provided by other means such as the window.

Trickle ventilators are designed to provide the required minimum fresh air rate, particularly in winter.

For England and Wales, Building Regulations Approved Document F[13] should be consulted for further details of the requirements.

Draughts, especially those occurring at ankle height, can be avoided by directing the incoming air upwards, or positioning the ventilators at high level, e.g. > 1.75 m above the floor. This allows incoming air to mix with the warmer room air before reaching floor level. Alternatively, air can enter through wall ventilators positioned behind heaters. The form of the ventilator should promote rapid mixing with the room air to minimise cold draughts. General guidance on the use of trickle ventilators has been published by BRE(14).

Added to this is the daily 'reservoir' effect of the trickle vents that purge the room overnight and provide a room full of fresh air ready for the following day's occupants. The larger the room volume, as with the higher ceilings in naturally ventilated rooms, the longer this reservoir effect will last during the occupied period. As trickle vents are intended to promote background ventilation only (about 5 litre.s^{-1} per person), their main application is for fresh air supply in the winter months. Twenty-four hour use of trickle ventilation can provide a reservoir of fresh air that may be sufficient to maintain air quality throughout the day. With higher pollutant loads, rapid ventilation by opening windows for short periods or by mechanical ventilation might be required. For this reason, trickle ventilators are usually used in conjunction with other types of ventilation opening.

Trickle vents can be in the window frame, part of the glazed unit or independent of the window (usually above it). Various refinements on the basic trickle ventilator are available. Acoustic trickle ventilators are available, which reduce noise level by about 38 dB, but bring a penalty of increased pressure drop.

Control options available include:

- *Basic (uncontrolled):* consisting of a series of holes or slots covered with a formed plastic cover to give protection from the weather; no control is possible, hence positioning and appropriate selection are very important.
- *Standard controllable (including 'hit and miss')*: closure may be possible through the use of a manually operated slide that covers the openings; occupants need to understand the operation of such devices.
- *Humidity controlled:* mostly used in kitchens and bathrooms within dwellings, as the scope for use in offices is limited with moisture not being the dominant pollutant.
- *Pressure controlled:* generally used in offices; inside/outside pressure difference is one possible control strategy.
- *Pollutant (e.g.* CO_2, *CO, smoke controlled)*: used in schools, theatres, shopping malls etc. and sometimes in dwellings; practical use for offices is limited as, except for CO_2 (where considerable drift has been reported), these are not normally the dominant pollutants.

The ventilation performance of trickle ventilators is traditionally specified in terms of 'free air space' or 'open area'. However, in reality, the airflow performance of two ventilators having the same free area (i.e. the physical size of the smallest aperture totalled over the ventilator) can be different due to the differing complexities of the airflow paths. In order to minimise resistance to airflow, the main air passage (excluding insect screens etc.) should have a minimum dimension of 5 mm for slots, or 8 mm for square or circular holes. Acoustic effectiveness is considered in the light of the 'effective area' or 'equivalent area'. This is considered in detail in CIBSE Guide B5: *Sound control*[15] and BRE guidance[16].

Effective area is also considered to be a more realistic measure of airflow performance, although it is not yet used as the basis of a test method. It is defined as the area of a single sharp-edged hole (in a thin plate) that passes the same volume airflow rate and at the same applied pressure difference as the vent being tested. It requires to be measured on an airflow test rig. Most trickle ventilators with the same equivalent area will have similar airflow performance, even though their free areas might differ. A European standard to improve air flow performance testing methods is in preparation[17], although it is not known whether this will be adopted in the UK.

5.3.4 Louvres

These are usually constructed of either glass or aluminium blades. Security bars can be fitted inside the louvres and this enhances their potential application in the night ventilation mode. Versions incorporating acoustic attenuation are also available. Whilst providing good control over summer ventilation, adjustable louvres usually present the greatest crack length for a given opening. However, conventional hinged louvres are usually difficult to seal when closed, making it difficult to limit infiltration losses.

5.3.5 Roof ventilators

In combination with low-level openings in the fabric, roof ventilators can be used to take advantage of summer stack effect, particularly for tall spaces. However, they must be specified to have low crack leakage or wind-induced draughts will cause discomfort in winter.

Rooftop ventilators generally fall into two categories: ridge and circular. The ridge type is less obtrusive but their efficiency is impaired by variations in wind direction, whereas circular stack outlets, if positioned correctly, are not affected. For maximum effect the outlet should be on the ridge of a pitched roof and the cap should project sufficiently above the ridge to minimise the influence of turbulence arising from wind blowing up the slope of the roof. Natural ventilation openings should never be installed on the slope of a roof nor should they be located in high-pressure areas of the building environment, where down-draughts are likely to occur.

5.3.6 Fixed lights

Fixed lights may give crack leakage rates between zero and 1 $m^3.h^{-1}$ per metre length of visible perimeter of glass, depending on the gasket material. Therefore crack leakage from roof lights should not be relied upon to provide winter ventilation.

5.3.7 Dampers

Dampers are usually used for applications where automatic control is required. In the context of natural ventilation, this is usually for air inlets below false floor level, and at main exhaust points (e.g. roof vents). Again, the key performance criterion is the ability of the damper to provide an airtight seal when closed to minimise energy losses in winter. If effective control is required, then a significant proportion of the available pressure differential must occur across the damper in order to provide control authority. This goes against the design requirement to minimise pressure drops because of the relatively low driving forces available with natural ventilation. This is partly compensated for by the fact that higher pressure differentials are available in the winter, when minimum damper openings are required.

5.3.8 Shafts and ducts

Many ventilation strategies rely on shafts to take air vertically through a building. Similarly, ducts (including floor voids) are used to provide horizontal distribution. The criteria for sizing these airways are very different to those used in sizing conventional mechanical ventilation systems in order to keep pressure drops within the range available from natural driving forces. This means that adequate space must be allowed to incorporate these larger ducts or shafts. A second crucial issue is the requirement to keep the inlet ducts clean to minimise air quality problems. This will require inlet screens and access for cleaning.

By definition, shaft outlets are at high level and therefore are in a region of higher wind speed. This means that the magnitude of the wind pressure acting on the shaft is likely to be large. Wind effects will probably dominate the pressure distribution through the system except at very low wind speeds. It is therefore vital that outlets are designed to create wind pressures that reinforce the intended flow direction. Usually this means creating a negative pressure coefficient at the top of the shaft, the exception being the wind scoop.

Orme et al.[18] provides information on the above roof pressure coefficients. For isolated buildings with no local flow interference, the minimum height of the stack above roof level to avoid back-draughts **is** given by:

$$h = a\,[0.5 + 0.16\,(\theta - 23)] \qquad (5.1)$$

where h is the height above roof level (m), a is the horizontal distance between the outlet and the highest point of the roof (m) and is the pitch of the roof (degree).

For roof pitches of less than 23°, the height of the outlet must be at least 0.5 m above the roof level. These simplified relationships represent a minimum stack height; greater heights may well provide higher suction pressures. This can be beneficial since it is possible to generate a suction greater than that generated on an opening on the leeward vertical face of the building.

More information on pressure coefficients over roofs is given in **BS** 6399-2[19]. For complex roof profiles or where surrounding buildings or other obstructions disturb the wind, model testing would be advisable.

As well as the position of the roof outlet, the geometry of the cowl will also effect the pressure coefficient. The cowl should prevent rain entering the stack and can provide flow acceleration local to the outlet to further reduce static pressures.

5.3.9 Combined openings

When designing the ventilation inlets, it is usual to use combinations of opening types in an overall design. These openings may be combined in a single window unit (e.g. opening window with a trickle vent in the frame), or may be independent. Combinations of window types in a single window unit should be considered. For example, a hopper over a centre pivot window has many advantages. The hopper can provide night ventilation, and also helps provide air to occupants deeper into the room. The centre pivot allows high summer ventilation rates and is especially beneficial to those nearer the perimeter. The different sizes of opening also allow finer control over ventilation rate by progressively opening the hopper, then closing the hopper and opening the main window and then opening both together.

As the design of the window unit is developed, the other functional requirements of the window need to be considered (e.g. lighting, shading, security, transmission losses etc.). Considerable development effort is underway to produce such 'multi-function' windows, see Figure 5.5[9].

5.3.10 Internal obstructions

Transfer grilles may be required as a minimum to allow air movement across a building if cellular accommodation has been provided. The resistance of these transfer grilles must be included in the design calculation when sizing the facade openings.

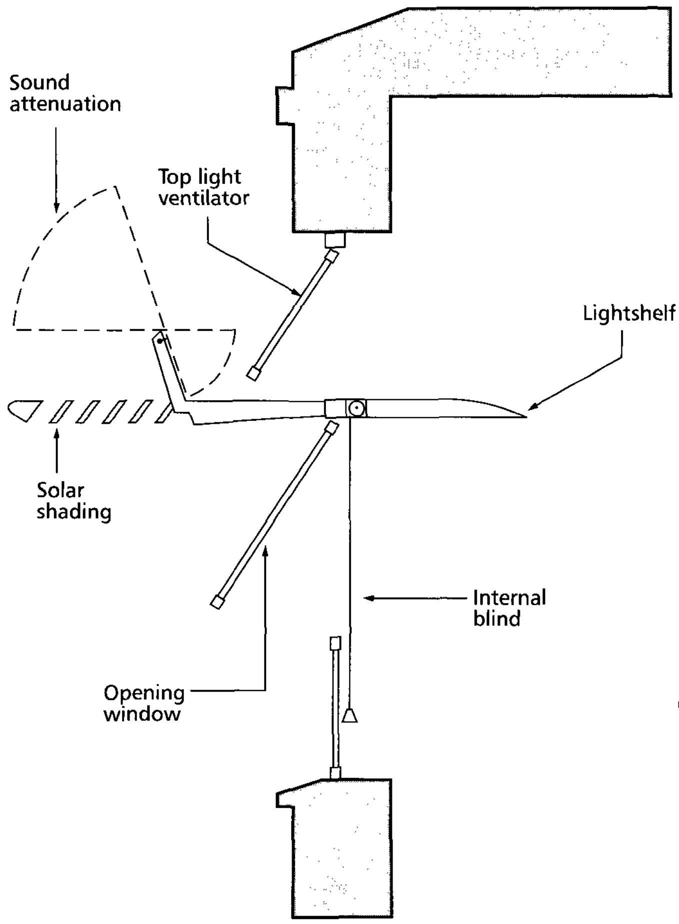


Figure 5.5 'Multi-function' window unit

5.3.11 Control options for natural ventilation openings

Control options for natural ventilation openings should be specified with the needs of the occupants in mind, see section 4.3. Control mechanisms for natural ventilation opening include the following

5.3.11.1 Window/damper actuators

A number of different actuator types are available for window control. These are electrically driven and include chain, helical cable, piston, and rack and pinion type actuators. Because of their linear action, the last two types suffer some disadvantage because they protrude into the space. The actuator will have to cope with the weight of the window and with any wind forces. The use of vertically pivoted windows minimises the effect of the weight of the window but they are less efficient as ventilators.

If dampers are used, then conventional control mechanisms (pneumatic or electric actuators) can be considered.

5.3.11.2 Sensors

Any automatic control system must be regulated in response to signals from appropriate sensors. Equipment to be specified includes the following:

- ***Temperature sensors:*** room temperature sensors may be sufficient to indicate excessive ventilation rates because of the influence of ventilation on room temperature. However this approach will need to be integrated with the heating system controls to avoid the two systems fighting each other. Other control parameters may be required in addition to temperature.
- ***Wind sensors:*** wind speed sensors (anemometers) can be used to reduce window opening as wind speeds increase in order to maintain a nominally constant ventilation rate. They may also be used in conjunction with rain sensors to give an indication of potential ingress of driving rain. Wind direction sensors can be used to shut exhaust vents on the windward side of a building and simultaneously open leeward vents in order to avoid back-draughts.
- ***Solar sensors:*** solar sensors (pyranometers) can be used to indicate periods of high solar gain. The sensor must integrate the gain over a certain time period to avoid hunting during periods of patchy cloud.
- ***Rain sensors:*** windows and vents may need to be closed during periods of rainfall to prevent ingress of water. Typical sensors include the 'tipping bucket', which collects rainfall and tips over at a certain level. Each tipping action generates a pulse, the frequency of which can be used to determine the intensity of the rainfall. An alternative approach is to use a device whereby the capacitance changes as the area of moisture on its surface increases. The sensor is heated to dry off the surface when the rain stops.
- ***Air quality sensors:*** a number of approaches to measuring air quality have been used. These

usually rely on taking a particular pollutant as indicator for the overall air quality. CO_2 and humidity sensors have been most commonly used, the former in commercial buildings, the latter in residential buildings where condensation is a more severe problem.

- ***Occupancy sensors:*** infra-red sensors which detect movement have been used to identify the presence of occupants and adjust ventilation rates (and lighting etc.) accordingly.

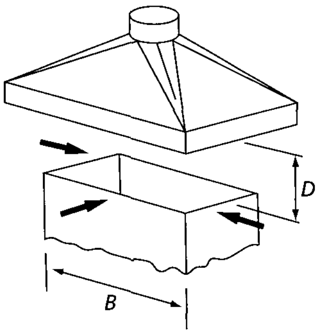
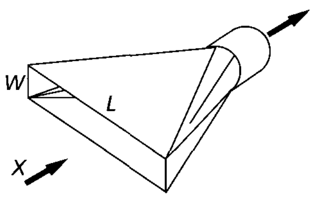
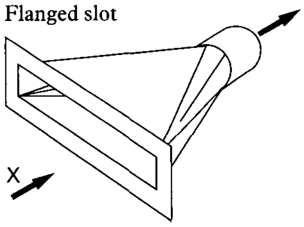
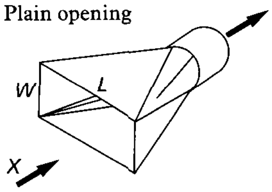
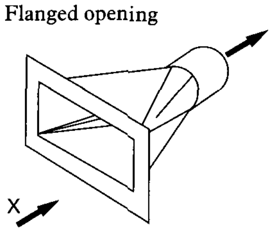
Further details on the application of these sensors can be found elsewhere[20].

5.4 Exhaust systems

5.4.1 General

Table 5.2 gives solutions to the equation for calculating air velocity at a set distance from an exhaust hood for various types of opening and the appropriate equations for air volume flow rates through overhead canopies for both cold and hot processes[21]. Appropriate control velocities and convective heat transfer rates are given in Tables 5.3 and 5.4 respectively.

Table 5.2 Air volume flow equations for hoods and canopies

Type of opening	Equation	Notes
Canopy 	Cold source: $Q = 1.4\,P\,D\,v$	If $D > 0.3\,B$, use equation for hot source Canopy should overhang tank by $0.4\,D$ on each side
	Hot source, exposed horizontal surface: $Q = 0.038\,A_s \sqrt[3]{h\,D} + 0.5\,(A - A_s)$	Q is progressively under-estimated as D increases above 1 m Canopy should overhang tank by $0.4\,D$ on each side
	Hot source, exposed sides and top: $Q = 0.038\,A_s \sqrt[3]{\left(h\,A_t\,D\right)/A_s} + 0.5\,(A - A_s)$	Q is progressively under-estimated as D increases above 1 m Canopy should overhang tank by $0.4\,D$ on each side
Plain slot 	$Q = L\,v\left(4\,X\sqrt{X/W} + W\right)$	Aspect ratio R should be not less than 10
Flanged slot 	$Q = 0.75\,L\,v\left(4\,X\sqrt{X/W} + W\right)$	Aspect ratio R should be not less than 10 If $X > 0.75\,W$, use equation for plain slot
Plain opening 	$Q = v\left(10\sqrt{R}\,X^2 + A\right)$	Aspect ratio R should not exceed 5; equation may be used for $R > 5$ but with loss of accuracy
Flanged opening 	$Q = 0.75\,v\left(10\sqrt{R}\,X^2 + A\right)$	Aspect ratio R should not exceed 5; equation may be used for R > 5 but with loss of accuracy If $X > 0.75\,W$, use equation for plain opening

Symbols:

A = area of hood/opening (m^2)
A_f = horizontal surface area of source (m^2)
A_t = total exposed heated surface area of source (m^2)
B = breadth of source (m)
D = height above source (m)
L = length of hood/opening (m)
P = perimeter of source (m)
Q = volume flow rate ($m^3 \cdot s^{-1}$)
R = aspect ratio (L/W)
W = width of hood/opening (m)
X = distance from source (m)
h = rate of convective heat transfer ($W \cdot m^{-2}$)
v = control velocity ($m \cdot s^{-1}$)

Table 5.3 Control velocities for hoods

Condition	Example	Control velocity / $m \cdot s^{-1}$
Released with practically no velocity into quiet air	Evaporation from tanks, degreasing etc.	0.25–0.5
Released at low velocity into moderately still air	Spray booths, intermittent container filling, low speed conveyor transfers, welding, plating	0.5–1.0
Active generation into zone of rapid air motion	Spray painting in shallow booths, conveyor loading.	1.0–2.5
Released at high initial velocity into zone of very rapid air motion	Grinding, abrasive blasting	2.5–10

Note: the higher values apply if (*a*) small hoods handling low volumes are used, (*b*) hoods are subject to draughts, (*c*) the airborne contaminant is hazardous, or (*d*) hoods are in frequent use.

Table 5.4 Convective heat transfer rates for horizontal surfaces

Surface temperature / °C	Rate of heat transfer / $W \cdot m^{-2}$
100	580
200	1700
300	3060
400	4600
500	6600

Table 5.5 shows the effects of adjacent surfaces on the basic form of hoods and canopies. However, specific processes may require other hood arrangements not shown in either Table 5.1 or 5.5. The American Conference of Government Industrial Hygienists' (ACGIH) publication, *Industrial ventilation*[22], which gives a wide range of empirically based design data sheets for many common industrial processes, should always be consulted before proceeding with the design of a local exhaust system.

The size, aspect ratio, position and number of openings used depends upon:

- the size and nature of the source (opening must overhang source if possible)
- the access requirements for, and position of, the
- the prevailing room air currents (side baffles should be provided if possible).

5.4.2 Overhead canopies

Overhead canopies are only appropriate for hot processes which cannot be kept covered, and must not be used if the operator is likely to lean over the process or if strong cross draughts are likely to occur. Baffle plates can be incorporated into larger hoods to ensure an even velocity across the opening, whilst very large hoods should be sectioned, each section having its own off-take.

5.4.3 Lateral exhaust

For processes in which the emission momentum is small or tends to carry the pollutant horizontally away from the source, horizontal slots or hoods at the edge of a work surface or tank may be used. Slots may be arranged one above the other, see Figure 5.6, or facing each other along opposite long edges, depending upon the vertical distance of the source above the rim of the tank. If the most remote part of the source is less than 0.5 m from the slot, a single exhaust slot along the longer edge is adequate otherwise two slots, on opposite sides of the source, are required.

5.4.4 Jet-assisted hoods

Jet-assisted hoods are non-enclosing hoods combined with compact, linear or radial jets. They are used to separate contaminated zones from relatively clean zones in working spaces. They prevent contaminated air from moving into clean zones by creating positive static pressures, typically in the form of an air curtain.

Table 5.5 Effect of side walls and adjacent surfaces

Type of opening	Baffle	Effect
Canopy, cold source	Side walls	Reduces effective perimeter, hence flow rate Q is reduced
Canopy, hot source	Side walls	Reduces cross draughts but minimal effect on flow rate Q
Plain slot	Long side on flat surface	$Q = L\,v\left(X\sqrt{2X/W} + W\right)$
Plain opening	Long surface on flat surface	For $R \leq 2$: $Q = v\left(5\sqrt{(2/R)}\,X^2 + A\right)$ For $2 < R \leq 5$: $Q = v\left(5\sqrt{(R/2)}\,X^2 + A\right)$
Flanged slot or opening	Long side (not flanged) on flat surface	For $X > 0.75$, calculate flow rate Q for plain arrangement and multiply by 0.75 W

Symbols:

A = area of hood/opening (m2)
L = length of hood/opening (m)
Q = volume flow rate ($m^3 \cdot s^{-1}$)
R = aspect ratio (= L / W)
v = control velocity ($m \cdot s^{-1}$)
W = width of hood/opening (m)
X = distance form source (m)

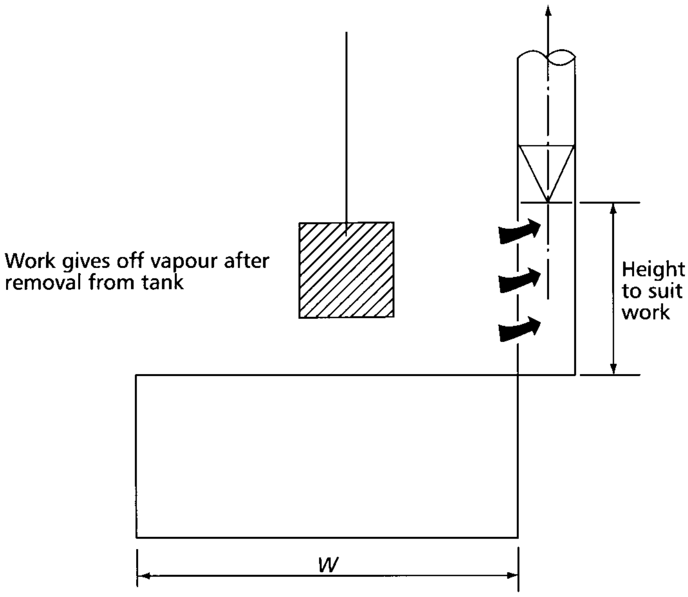


Figure 5.6 Open surface tank with drying facility, single side exhaust

5.4.5 Push-pull hoods

For sources larger than 1 m across, a push-pull hood arrangement should be used, see Figure 5.7, whereby a slot or row of nozzles is used to blow air across the source. Design data for the hood illustrated in Figure 5.7 are given below.

Exhaust air quantity:

$$Q_e = (0.5 \text{ to } 0.75) \times A \tag{5.2}$$

where Q_e is the exhaust air flow rate ($m^3 \cdot s^{-1}$) and A is the area of open surface (m^2).

The value of the numerical factor depends on the temperature of the liquid, presence of cross-draughts, agitation of liquid etc.

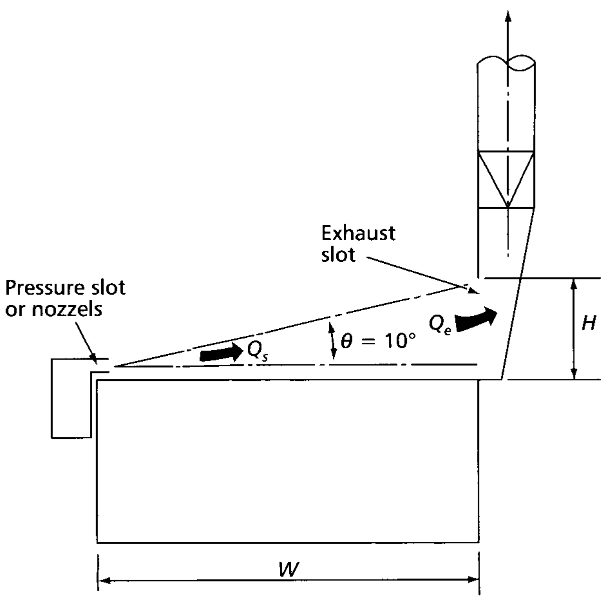


Figure 5.7 Push-pull hood

Supply air quantity:

$$Q_s = \frac{Q_e}{w \times E} \tag{5.3}$$

where Q_s is the supply air flow rate ($m^3 \cdot s^{-1}$), w is the throw length (m) and E is the entrainment factor (see Table 5.6).

Height of exhaust opening:

$$H = 0.18\, w \tag{5.4}$$

where H is the height of the exhaust opening (m).

Width of supply opening:

— size for a supply velocity of 5–10 $m \cdot s^{-1}$.

The input air volume is usually about 10% of the exhaust volume and the input air should be tempered to avoid frost damage. The source must not be placed in the input air path since this could result in deflection of the contaminant into the workspace. If necessary, baffles or screens should be used to deflect cross draughts.

Table 5.6 Entrainment factors for push-pull hoods

Throw length / m	Entrainment factor
0–2.5	6.6
2.5–5.0	4.6
5.0–7.5	3.3
>7.5	2.3

5.4.6 Equipment selection principles and integration

Air cleaning equipment may be selected to:

— conform with emissions standards; industrial processes resulting in polluting emissions to air, water or land come under the requirements of the Environmental Protection Act[3]

— prevent re-entrainment where they may become a health or safety hazard in the workplace

— reclaim usable materials

— permit cleaned air to be recirculated

— prevent physical damage to adjacent properties

— protect neighbours from contaminants.

Circular ductwork is normally preferred as it offers a more uniform air velocity to resist the settling of material and can withstand the higher pressures normally found in exhaust systems. Design velocities can be higher than the minimum transport velocity but should never be significantly lower. Fans (or other air-moving devices) and duct materials and construction should be suitable for the temperatures, abrasion and corrosion likely to be encountered. Fans should normally be located downstream of the air cleaner to reduce possible abrasions and create a negative pressure in the air cleaner so leakage will be inward. However, in some instances the fan may be located upstream from the cleaner to help remove dust.

Exhaust stacks must be designed and located to prevent the re-entrainment of discharged air into supply system inlets, see section 5.2. Toxic and hazardous exhaust must not be discharged in a manner that will result in environmental pollution and the local authority Environmental Health Officer should be consulted to ensure that the proposed discharges will be acceptable.

5.5 Mixing boxes

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

Mixing boxes must be designed to provide sufficient mixing so that freezing outside air does not stratify below warm recirculation air on entering the filters. If in doubt, a frost 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 mixing. Air blenders/baffles can also be used.

To improve the rangeability of a motorised control damper, the face velocity should be increased to 5-6 $m.s^{-1}$ by adjusting the duct size or by blanking-off an appropriate area of the duct at the damper. Damper quality is critical; play in linkages and pivots should be minimal and leakage on shut-off should be less than 0.02.

5.6 Heat recovery devices

5.6.1 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 ventilation rate is fixed by cooling rather than ventilation needs, and is therefore only applicable to all-air systems. The air quality implications of recirculation can also limit its use.

Heat recovery devices used in ventilation systems generally provide heat recovery from exhaust to supply air in winter and, in addition, can recover cooling in peak summer conditions. They are also used in specific system configurations such as indirect evaporative cooling, see section 4.12.

Devices used to recover heat from process applications (e.g. dryers, flues) may transfer the heat to the process or to another application. Selection of equipment should be suitable for process exhaust temperatures. Where the recovered heat is fed to a ventilation system, modulation control is normally required to prevent overheating in warm weather.

Buildings should be airtight as infiltration has a significant impact on the viability of heat recovery[23].

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

- heat recovery efficiency (sensible and total)
- airflow arrangement
- fouling (filters should be placed in both supply and exhaust airstreams)
- corrosion (particularly in process applications)
- cross-leakage
- condensation and freeze-up
- pressure drop
- face velocity
- construction materials (suitability for temperatures, pressures, contaminants)
- maintenance (in particular cleaning of surfaces)
- controls.

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

$$\text{Efficiency} = \frac{\text{Actual heat transfer}}{\text{Maximum possible heat transfer}} \qquad (5.5)$$

The maximum theoretical efficiency is a function of the exchanger flow configuration; counterflow exchangers have a higher theoretical efficiency than parallel flow exchangers. Practical consideration often favour crossflow arrangements that lie between the two[24].

Sensible heat recovery devices do not transfer moisture. 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 air and in cold, dry climates to increase the moisture in the supply air-

Drains should be included to collect and dispose of the 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.

Pressure drops depend on a number of factors including exchanger design, airflow rates, temperatures, and connections. These pressure drops should be minimised as they impose a fan energy penalty that will need to be balanced against 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:

- cost expenditure on device, filters etc. and savings on other plant (e.g. boilers, chillers) due to heat recovery

Table 5.7 Comparison of heat recovery devices[27]

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 air stream 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

- energy, both recovered and required to operate the system (e.g. fan, pump, wheel)
- maintenance requirements
- space requirements of device, filters etc.

Energy analysis may be undertaken using simulation modelling or spreadsheets calculations based on hourly conditions[25], or using graphical approaches such as load duration curves[26]. Table 5.7, which is based on information from ASHRAE[27], compares a number of heat recovery devices. These devices are described below. See BSRIA Technical Note TN11/86[28] and CIBSE Research Report RR2[29] for further information on selection and evaluation of heat recovery devices.

5.6.2 Recuperators

Recuperators usually take the form of simple and robust air-to-air plate heat exchangers, see Figure 5.8. Their efficiencies depend on the number of 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.

Modulation control is normally achieved by means of a 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. In applications with high differential pressures (> 1000 Pa) exchangers should be selected to avoid plate deformation.

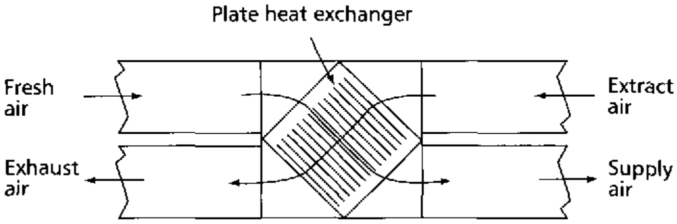


Figure 5.8 Recuperator using a plate heat exchanger

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

5.6.3 Run-around coil

Finned air-to-water heat exchangers are installed in the ducts between which the heat is to be transferred. A water or water/glycol (for freeze protection) circuit is used to transfer heat from the warm extract air to the cooler supply air (or vice versa in summer), see Figure 5.9. An expansion tank is required to allow fluid expansion and contraction. Overall heat transfer efficiencies are relatively low, as it is a two-stage heat transfer process, and pump energy (in addition to the fan energy penalty) and maintenance costs need to be taken into account. However, the system is flexible in application, as it places no constraints on the relative location of the two airstreams and can be extended to include multiple sources and uses. They are suitable for applications where contaminants in the exhaust airstream prohibit recirculation.

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

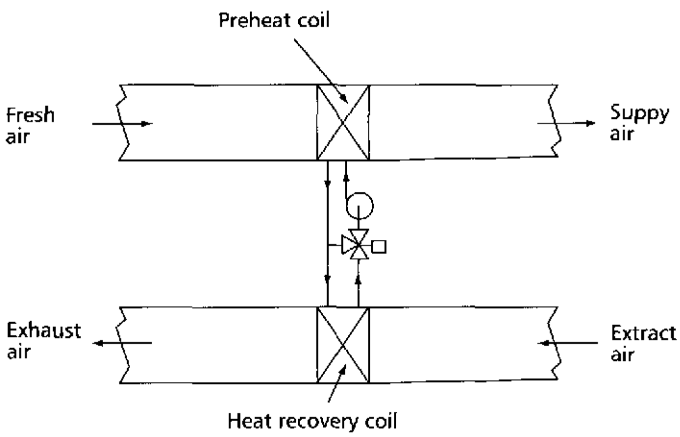


Figure 5.9 Run-around coil arrangement

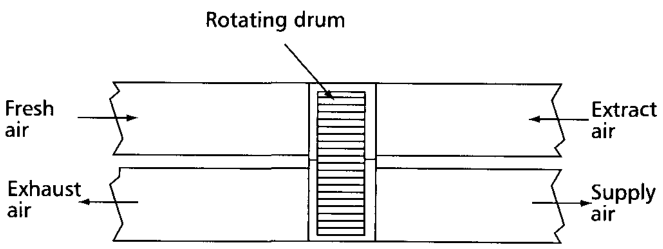


Figure 5.10 Thermal wheel

5.6.4 Thermal wheels

A thermal wheel comprises a cylinder, packed with a suitable heat transfer medium, that rotates slowly within an airtight casing which bridges the ducts between which heat is to be transferred, see Figure 5.10. Thermal wheels are generally quite compact and achieve high efficiencies due to a counterflow configuration. The heat transfer properties are determined by the material contained in the wheel, i.e:

- corrugated, inorganic, fibrous, hygroscopic material which 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 the thermal wheel need to be taken into account, since they can be difficult to clean[28], as do the additional energy penalties due to the drive (although these are usually low).

Cross-contamination 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 recirculation is undesirable. Leakage occurs due to the pressure difference between the two airstreams. This can be minimised by avoiding large pressure differences, providing an effective seal, and placing the fans to promote leakage into the exhaust airstream. Hygroscopic media may transfer toxic gases or vapours from a contaminated exhaust to a clean air supply.

Modulation control is commonly achieved either by 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.

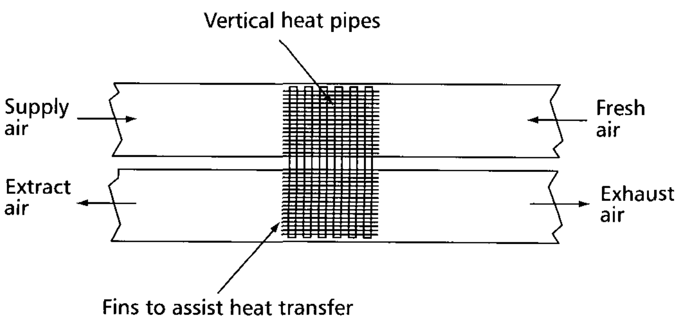


Figure 5.11 Vertical heat-pipe arrangement

5.6.5 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 transfer liquid by capillary action
- ***vertical:*** in which heat from the warmer lower duct is transferred to the cold upper duct by means of a phase change in the refrigerant, see Figure 5.11.

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.s^{-1}) in order to improve efficiency.

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

5.6.6 Regenerator

A regenerator, see Figure 5.12, consists of two accumulators (or a single unit split into two halves) with a damper arrangement to cycle the supply and exhaust air flows between the two. In the first part of the cycle, the exhaust air flows through, and heats, one of the accumulators. The dampers then changeover 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% and above. Latent efficiencies are normally significantly lower and vary with flow velocity and 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

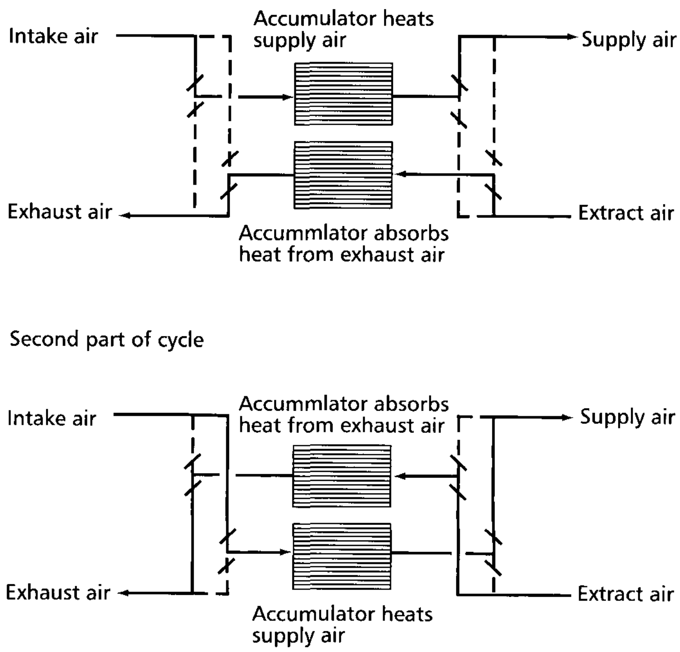


Figure 5.12 Regenerator

time required for damper changeover should be kept to a minimum using high torque dampers. Cross-leakage can range from below 1% on well designed systems up to 5% 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.

5.6.7 Heat pump

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 4.16 for further information on heat pumps.

5.6.8 Further reading

Holman **J P** ***Heat transfer*** (New York NY: McGraw-Hill) (1986) ISBN 0 07Y66459-5

5.7 Air cleaners and filtration

5.7.1 Nature of airborne contaminants

Atmospheric dust is a complex mixture of solid particulate matter, comprising dusts, smokes, and fumes and non-particulate vapours and gases. A sample of atmospheric dust may contain minute quantities of soot and smoke, minerals such as rock, metal or sand, organic material such as grain, flour, wool, hair, lint and plant fibres and, perhaps, mould spores, bacteria and pollen. Particles are not generally called dust unless they are smaller than 80 m.

Smokes are suspensions of fine particles produced by the incomplete combustion of organic substances such as coal or wood, or by the release into the atmosphere of a wide variety of chemical compounds in a finely divided state. Smoke particles vary considerably in size from about 0.3 m downwards. Fumes are solid particles, predominantly smaller than 1.0 m, formed by the condensation of vapours.

Non-particulate contaminants consist of vapours condensable at normal pressures and temperatures, and gases, of which the most damaging to plants and buildings is sulphur dioxide. Carbon monoxide and various oxides of nitrogen are also present in minute quantities.

There is a wide variation in atmospheric solids between rural, suburban and industrial areas, as shown in Table 5.8.

Table 5.9 shows an analysis of a sample of atmospheric dust, in terms of the total numbers of particles for the size range. The figures may be considered typical for average urban and suburban conditions, but wide variations may be encountered in particular cases.

Current emphasis in office and other 'standard' accommodation is on the removal of particles smaller than 10 m. These, along with chemicals outgassed from carpets and furnishings in modern workspaces, have been linked with reports of sick building syndrome and are able to penetrate into the lungs, causing respiratory problems.

Considerable work has been carried out on the performance of filters and air cleaning units in relation to cigarette smoke, see section 5.7.3.

5.7.2 Definitions

The following definitions, drawn from BS EN 779[30] are commonly used in describing the properties of air filters.

Rated airflow rate

The quantity of air the filter is designed to handle as specified by the manufacturer. Expressed in m^3.s^{-1} (for a reference air density of 1.20 kg.m^{-3}).

Face velocity

The airflow rate divided by the face area (m.s^{-1}).

Initial pressure drop

The pressure drop (Pa) of the clean filter operating at its rated airflow rate.

Final pressure drop

The pressure drop (Pa) up to which the filtration performance is measured for classification purposes-

Atmospheric dust spot efficiency (E)

A measure of the ability of the filter to remove atmospheric dust from the test air. This efficiency is measured on a light transmission basis (%).

Average atmospheric dust spot efficiency (E_m)

The average of the dust spot efficiency values (%).

Table 5.8 Typical amounts of solids in the atmosphere for various localities

Locality	Total mass of solids / mg.m^{-3}
Rural and suburban	0.05-0.5
Metropolitan	0.1-1.0
Industrial	0.2-5.0
Factories or work rooms	0.5-10.0

Table 5.9 Analysis of typical atmospheric dust in relation to particle size

Range of particle size (diameter) / mm	Amount of solid as percentage of number of particles and total mass of particles / %	
	Number of particles	Total mass of particles
30 to 10	0.005	28
10 to 5	0.175	52
5 to 3	0.250	11
3 to 1	1.100	6
1 to 0.5	6.970	2
Less than 0.5	91.500	1

Synthetic dust weight arrestance (A)

A measure of the ability of the filter to remove injected synthetic dust from the air (%). This arrestance is calculated on a weight basis.

Initial synthetic dust weight arrestance (A_i).

The first dust weigh arrestance (%) obtained from a dust feed increment of 30 g.

Average synthetic dust weight arrestance (A_m)

The average of the values of synthetic dust weight arrestance (%).

5.7.3 Filter testing

5.7.3.1 Tests for filters for general purposes

The comparative method of testing of air filters for general purposes, BS 6540: Part 1[33], has now been superseded by BS EN 779[30].

These tests are intended for filters for use in air systems handling between 0.24 $m^3.S^{-1}$ and 1.4 $m^3.s^{-1}$ and with initial atmospheric dust spot efficiencies no greater than 98%. For higher efficiencies, the sodium flame test given in BS 3928[32] is appropriate.

Based on their average synthetic dust weight arrestance or average dust spot efficiency, see Table 5.10, filters are classified into two groups, as follows:

- Group G: coarse dust filters, classes Gl-G4
- Group F: fine dust filters, classes F5-F9.

If the initial average dust spot efficiency is less than 20%, the filter is automatically classified as group G and no further tests, other than arrestance, are carried out.

If the filter is classified as a fine dust filter it is subsequently tested for:

- air flow
- initial pressure drop
- atmospheric dust spot efficiency.
- synthetic dust weight arrestance
- dust holding capacity
- average efficiency
- average arrestance
- pressure drop.

Atmospheric dust spot efficiency

Otherwise known as the blackness test, this test involves sampling upstream and downstream air quality by drawing sample air quantities over target filters and comparing changes in opacity with time.

Synthetic dust weight arrestance

This gravimetric test uses a synthetic dust comprised of carbon, sand and lint in controlled proportions similar to those found in a typical atmosphere. **A** known mass of dust is injected into test apparatus upstream of the filter and the dust passing the filter is collected in a more efficient final filter. The increase in mass of the final filter is used to calculate arrestance.

Dust holding capacity

The synthetic dust weight arrestance test can be continued in cycles to achieve a picture of changes in efficiency and arrestance with increasing dust loading until the rated maximum pressure loss or minimum arrestance has been reached. The dust holding capacity can be determined from the total mass of synthetic dust held by the filter.

5.7.3.2 Test for high efficiency filters

The preferred pan-European test method for testing high efficiency HEPA and ULPA (ultra-low particle arrestor) filters is BS EN 1822[33]. This test method is based on scanning by a particle counter at the most penetrating particle size (MPPS) of the filter. MPPS is variable and is determined by testing samples of the filter medium used in the manufacture of the filter being tested. The challenge aerosol is DEHS mineral oil or equivalent, but other oils are permitted. condensation nucleus counters (CNC) are used for monodispersed aerosols and laser particle counters (LPC) for polydispersed aerosols.

Based on their performance in the aerosol challenge test HEPA and ULPA filters are classified into two groups, as follows, see Table 5.11:

- Group H: HEPA filters, classes H10-Hl4
- Group F: ULPA filters, classes U15-Ul7.

Table 5.10 Classification according to filtration performance[30]

Characteristics	Filter group	Filter class	Old EU rating	Class limits / %	
				Average arrestance, A_m	Average efficiency, E_m
Coarse*	G	G1	EU1	$A_m < 65$	—
		G2	EU2	$65 \le A_m \le 80$	—
		G3	EU3	$80 \le A_m \le 90$	—
		G4	EU4	$90 \le A_m$	—
Fine	F	F5	EU5	—	$40 \le E_m < 60$
		F6	EU6	—	$60 \le E_m < 80$
		F7	EU7	—	$80 \le E_m < 90$
		F8	EU8	—	$90 \le E_m < 95$
		F9	EU9	—	$95 \le E_m$

* Initial dust spot efficiency < 20%

BS 3928[32], which is still valid in the UK, describes a test method for high efficiency filters not covered by BS EN 779[30], i.e. filters having a penetration less than 2%. It is not considered to be as vigorous as a DOP or CNC test. Testing involves generation of an aerosol of sodium chloride containing particles ranging in size from 0.02 to 2 µm. The amount of particulate matter passing through the filter is determined by sampling both upstream and downstream of the filter and passing each sample through a flame photometer to determine the concentration of sodium chloride particles captured. BS 3928[32] is based on Eurovent 4/4[34] and results achieved under both standards should be comparable.

5.7.3.3 On-site testing

The efficiency of a filter installation depends not only on the filter efficiency but also on the security of the seal between the filter and the air system. This is particularly vital in high efficiency particulate air filters (HEPA) installations; hence penetration must be established immediately prior to use and at regular intervals throughout the working life of the system.

Tests that have been used to determine on-site penetration include:

- *Di-octyl-phthlate (DOP) test*: DOP is an oily liquid with a high boiling point. Normally, DOP vapour is generated at a concentration of 80 mgm^{-3} and the downstream concentration is determined using a light scattering photometer via a probe which scans the entire downstream face of the filter installation.

- *Sodium flame:* a portable version of BS 3928[32] test apparatus that utilises a salt-stick thermal generator to produce an aerosol and an oxy-propane flame and portable photometer for penetration assessment.

5.7.3.4 Gas and vapour removal

Most manufacturers quote efficiencies for removal of a wide range of gases and vapours based on upstream and downstream concentrations. Adsorption filters are also rated in terms of the mass of gas/vapour that can be adsorbed before saturation of the adsorbent.

Specification and testing methods have been developed for gas and vapour removal by filters and re-circulating air cleaning units[35]. This work has looked at the performance of a wide range of systems including active bonded carbon units and electrostatic filters.

Specialist advice should be sought on any requirements.

5.7.3.5 Dry testing

In applications such as cleanrooms used for the production of semi-conductors, testing for local leaks with an aerosol such as DOP would result in filter contamination and subsequent production problems. In these circumstances filters are tested using atmospheric air or polystyrene latex spheres (PSL).

5.7.4 Filter application and selection

Table 5.12, Page 5-17, presents a broad classification of air cleaners and Figure 5.13, page 5-18, illustrates the various characteristics of dusts, mists etc., together with other relevant data. Table 5.13 provides recommended filter specification data drawn from the National Engineering Specification[36] and promoted within BSRIA guidance[37,38].

CIBSE Technical Memoranda TM26[39] considers other means of reducing the admittance of micro-organisms other than just the installation of a HEPA filter. Under

Table 5.13 Recommended filter specification data

Filter data to be specified	Essential	Desirable
Air flow rate ($m^3.s^{-1}$)	●	
Air velocity ($m.s^{-1}$)		●
Initial filter pressure drop (Pa)	●	
Final filter pressure drop (Pa)	●	
Average arrestance (%)	●	
Initial dust spot efficiency (%)	●	
Average dust spot efficiency (%)	●	
Minimum dust holding capacity (g)	●	
Class of filter (EU number)		●
Size of filter (height, width, limiting depth (mm))	●	
Casing		●
Test standards	●	
Access		●
Filter media		●

Table 5.11 Classification of HEPA and ULPA filters[33]

Type	Filter group	Filter class	Old EU rating	Overall value / %		Local value* / %	
				Efficiency	Penetration	Efficiency	Penetration
HEPA	H	H10	EU10	85	15	-	-
		H11	EU11	95	5	-	-
		H12	EU12	99.5	0.5	-	-
		H13	EU13	99.95	0.05	99.75	0.25
		H14	EU14	99.995	0.005	99.975	0.025
Fine	U	U15	EU15	99.999 5	0.000 5	99.997 5	0.002 5
		U16	-	99.999 95	0.000 05	99.999 75	0.000 25
		U17	-	99.999 995	0.000 005	99.999 9	0.000 1

* Local values lower than those given in the table may be agreed between supplier and purchaser

Table 5.12 Classification of air cleaners

Type	Remarks	Method of cleaning	Face velocity m.s-1	Resistance at face velocity / Pa		Dust holding capacity	Relative efficiency / %		Relative cost
				Initial	Final		Sodium flame	Synthetic dust	
Viscous impingment:									
panel or unit	Thickness ranges 12-100 mm; small or intermediate air volumes; good for particles >10 um diameter; efficiency decreases with dust loadings; used as after-cleaners	Permanent (washable) or disposable	1.5-2.5	20-60 (depending on thickness)	100-150	High; can be critical	10	>85	Low
moving curtain	Will handle heavy dust loas; intermediate or large air loads; used as precleaners etc.	Continuous or intermittent; can be automatic	2-2.5	30-60	100-125 (operating)	Self-cleaning by immersion	10	>85	Medium
Dry:									
panel, bag, cartridge or unit with fabric or fibrous medium	Small or intermediate air volumes	Usually	1.25-2.5	25-185 (depending on efficiency)	125-250 (depending on efficiency)	Generally not as high as viscous impingement; can be critical	30-80 (depending on filter type, medium and face velocity)	96-100 (depending on filter type, medium and face velocity)	Low to high
moving curatin	Intermediate or large air volumes	Continuous or intermittent; can be automatic or disposable	2.50	30-60	100-175 (operating)	Self-cleaning	Can be selected over a wide range		Medium to high depending on efficiency
absolute or diffusion (HEPA)	Pre-filter necessary; small air volumes; particles down to 0.01um diam.	Disposable	Up to 2.5	Up to 250	Up to 625	Low	>99.9	100.00	High
Electrostatic:									
charged plate	Pre-filter desirable; after-filter used to collect agglomerates; power-pack and safety precautions necessary (up to 12 kV); particles down to 0.01 um diam; intermediate to large air volumes	Washable or wipable; can be automatic	1.5-2.5	Negligible; resistance added (40-60 Pa) to improve uniformity of air distribution		Can be critcal	-	Not suitable over 5 um diam.	High; low maintenance costs
charged medium	As for charged plate	Disposable	1.25	25	125	High	55-65	Not suitable over 0.5 um diam.	High; low maintenance costs
Adsorption units	Should be protected from dust, oil and grease; used for odour removal*	Can be reactivated	Low	Low; can be selected; constant		Medium absorbs up to half its own weight of many organic substances	95 (dependent on gas to be removed)		High
Mechanical collectors	Not suitable for particles less than 0.01 um diam.	To be emptied	Varies with design	50-100 (some act as air movers)	Constant (some act as air movers)	High	-	-	High; low maintenance costs

Odours can also be removed by combustion, masking or liquid absorption devices

Note: air washers used for humidification or dehumidification purposes sometimes also act as air cleaning devices. These include capillary air washers, wet filters, adsorption spray chambers etc., for which manufacturers' data should be consulted.

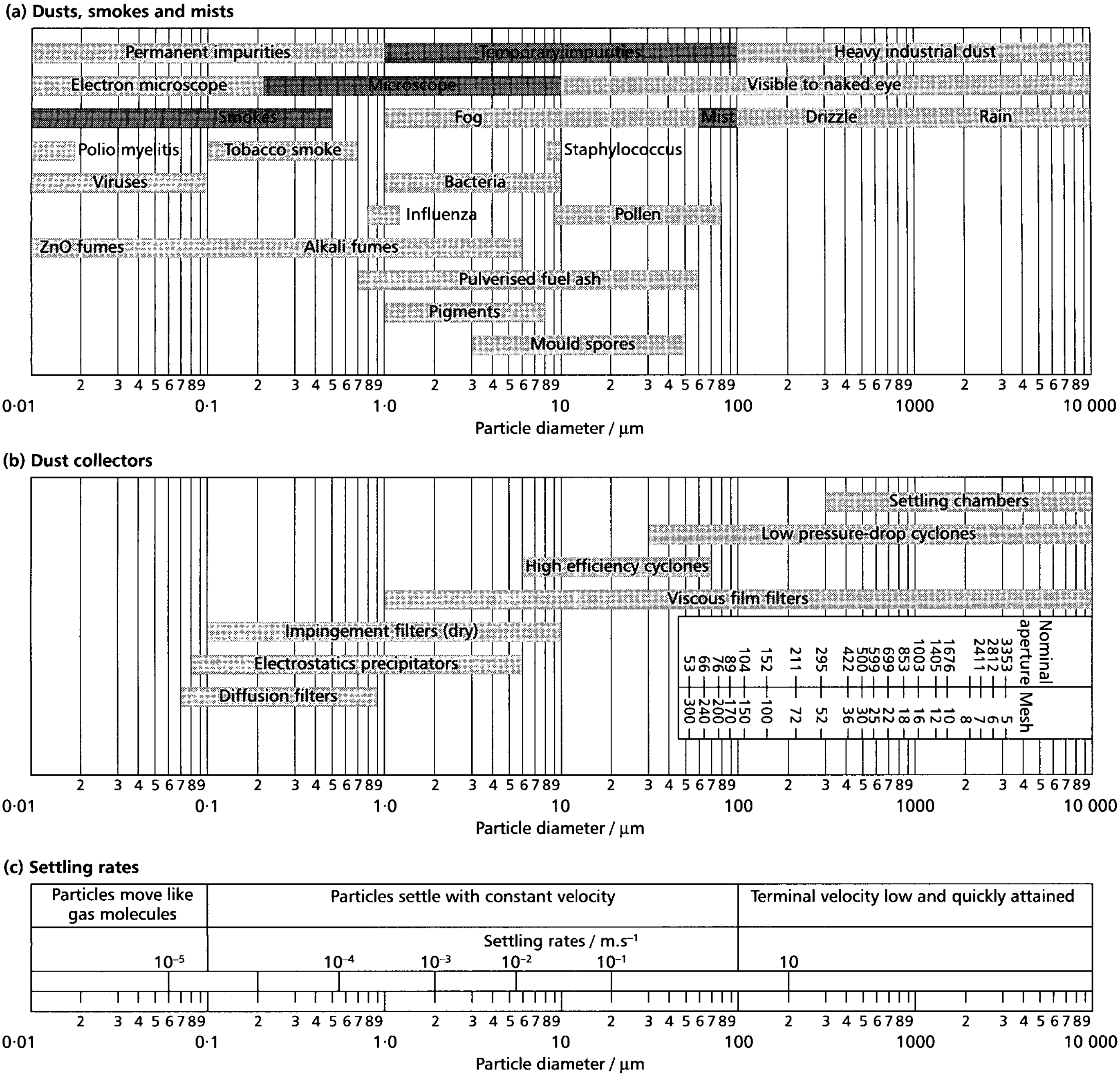


Figure 5.13 Characteristics of dusts and mists

certain conditions, air filters can support the growth of micro-organisms and act as a source of contaminants. Standard air filters can be obtained with an anti-microbial coating that is reported to kill or inhibit the growth of micro-organisms on the filter material and any trapped dust and debris. However, due to the potential for the active biocide to outgas from the surface, the user of such systems should take steps to ensure that they are safe for building occupants. Anti-microbial ductwork coatings are also available. However they also have a potential for the active biocide to outgas from the surface.

Ultraviolet germicidal irradiation (UVGI) is provided by ultraviolet lamps mounted in the supply ductwork. The UV light causes inactivation of micro-organisms by disrupting their DNA. This system is claimed to be effective against all types of bacteria and fungi, as well as spores and viruses, which are normally found in the air. The user of such systems should ensure that staff are protected from exposure to the UV radiation.

Photo-catalytic oxidation technology involves the action of low energy ultraviolet on a catalyst in the presence of water vapour that generates hydroxyl radicals that destroy micro-organisms. As this is an oxidation process the microbial hydrocarbons are reduced to carbon dioxide and water. This technique can be used against bacteria, fungi (including spores), viruses and allergens.

5.7.5 Filter maintenance

The life of a filter depends upon the:

- concentration and nature of the entering contaminants
- filter efficiency
- dust holding capacity corresponding to rise in pressure loss between clean and dirty conditions
- face velocity at the filter.

A maintenance regime can be derived based on time intervals or on condition. Details of local external conditions that may affect filter life, such as the entering pollution concentration, may be determined in consultation with the local Environmental Health Officer. Alternatively, a local survey may be undertaken. Some filter manufacturers provide prediction data for hours of use for different localities. Tables 5.8 and 5.9 give typical data on the amount and nature of solids in the atmosphere. Issues of external air quality, including sulphur dioxide and particulate matter (PM_{10}), are discussed in the *Air quality strategy for England, Scotland, Wales and Northern Ireland*[40], which is subject to periodic review.

If condition-based maintenance is being used the filter pressure differential is monitored. Replacement filters are installed when a specific differential is attained. If the filter represents a significant proportion of the total pressure loss of the system, and there is no provision for automatic fan duty adjustment (e.g. a VAV system), then the rise in pressure loss due to filter soiling should not exceed 20% of the total system loss with a clean filter. This differential can be reported via a building energy management system (BEMS). Note that a method of alerting maintenance staff of filter failure or blockage is also required for the time-based replacement method.

Further details on filter maintenance can be found in guidance produced by BSRIA[37] and HVCA[41]. Designers are also referred to CIBSE TM26: *Hygienic maintenance of office ventilation ductwork*[39]

5.7.6 Filter installation

BSRIA has analysed the whole life performance of filter systems[37,38] (i.e. the balance between space and capital costs and the operating costs such as inspection, change, energy and costs of associated equipment e.g. duct cleaning and redecoration). The conclusion is that filter performance depends not only on the filter specification but also on the design and installation of the filter system.

Poor filter installation will neutralise the benefits of specifying good filters[42]. The overall efficiency for the filter installation must be not less than that specified for the filter. It is suggested that:

- air intakes are located at high level away from the direction of the prevailing wind to prolong filter life and improve the quality of the intake air
- air filters should be protected from direct rain by using weather louvres to prevent water logging
- filters should be installed upstream of mechanical equipment to provide protection for that equipment; a final filter should be located downstream of the fan under positive pressure to reduce the risk of dust entering the system downstream of the filter
- adequate access for cleaning should be provided
- filter frames should be of good quality to prevent leakage and distortion; side withdrawal will make this difficult to achieve.

5.8 Air heater batteries

5.8.1 General

A heater battery 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.

5.8.2 Materials

Tubes should be of solid drawn copper, expanded into collars formed on the copper or aluminium fins. Tube wall thickness should not be less than 0.7 mm for LTHW or 0.9 mm for 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. Fins should not be spaced more closely than 330 fins per metre.

Provision should be made in the tube arrangement, by bowing or otherwise, to take up movement due to thermal expansion. Casings and flanges should be of adequate gauge in mild steel, painted with a rust resisting primer. Alternatively, the casings may be in galvanised mild steel with flanges painted in rust resistant primer. Occasionally both casings and flanges may be galvanised after manufacture.

5.8.3 Test pressure

Batteries should be tested with water at 2.1 MPa or 1.5 times the working pressure, whichever is the greater-

5.8.4 Heating medium

This is usually LTHW, MTHW, HTHW or dry steam. Where steam is used for preheat coils handling 100% outdoor air, so-called ‘non-freeze’ heater batteries should be selected. These coils have co-axial steam and condensate tubes that prevent build-up of condensate, and consequent risk of freezing, in the lower part of the battery.

5.9 Air cooler batteries

5.9.1 General

A cooler coil 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 row, thus approximating contra-flow heat exchange. Condensate drain trays through the depth of the coil are essential. These must be fitted at vertical intervals of not more than 1 m to facilitate proper drainage from the fins. Each such condensate collection tray must be drained using not less than a 22 mm connection. Eliminator plates are necessary if face velocities exceed 2.25 $m.s^{-1}$. Cooler coils should normally be located on the low pressure side of the supply fan to avoid condensate leakage through the casing.

5.9.2 Materials

Tubes should be of solid drawn copper, electro-tinned and expanded into collars formed in aluminium. 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 be to suit the test pressure, but not 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 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 sump.

5.9.3 Sprayed cooler coils

These are generally similar to unsprayed coils, except that eliminator plates must always be fitted; also, the main sump tank is deeper and provides a reservoir of water for the spray pump. An array of standpipes and spray nozzles is fitted on the upstream side of the coil, the main sump is made of 3.2 mm black mild steel, galvanised after manufacture and coated internally with bitumenised paint. Aluminium fins must not be used.

5.9.4 Test pressure

Cooler coils should be tested with water at 2.1 MPa or 1.5 times the working pressure, whichever is the greater.

5.9.5 Cooling medium

The cooling medium is usually chilled water or, occasionally, chilled brine. Where the latter is used the reaction of the brine with the piping and pumping materials must be considered and suitable steps taken to prevent corrosion.

5.9.6 Refrigerant cooling coils

When the coil is a refrigerant evaporator, additional care must be taken with its design, material selection, and control because of interaction with the refrigeration system. The normal vapour-compression refrigeration system using an oil-miscible refrigerant and thermostatic expansion valve has limited rangeability.

For a wide control range it is usually necessary to divide the coil into two or more 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 rangeability of the whole is increased as far as the limit of the operation of the sections remaining. For sectional control, the psychrometric effect of coil section arrangement must be appreciated, as shown in Table 5.14.

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 total 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 duty. This avoids a tendency towards frosting due to unequal evaporating temperatures.

5.9.7 Further reading

Air-cooling and dehumidifying coils **Chapter 21 in ASHRAE Handbook: *HVAC Applications* (Atlanta, GA: American Society of Heating, Refrigerating and Air-conditioning Engineers) (2000)**

Refrigerant-control devices **Chapter 45 in ASHRAE Handbook: *Refrigeration* (Atlanta, GA: American Society of Heating, Refrigerating and Air-conditioning Engineers) (1997)**

5.10 Humidifiers

5.10.1 Requirements for humidity control

The need to provide humidity control is considered in CIBSE Guides A[4] and H[43] and in section 4.4.

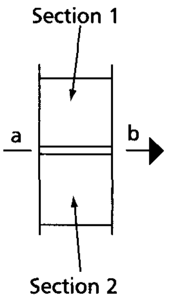
5.10.2 System classifications

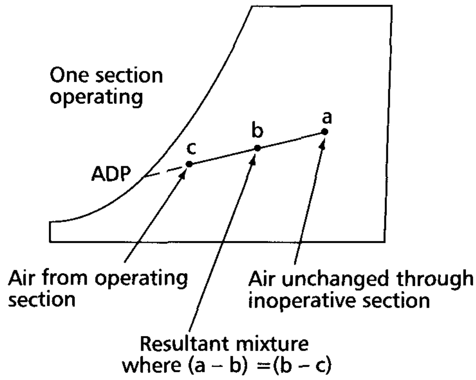
The variety of equipment types likely to be encountered is shown in Tables 5.15 and 5.16. However, the trend is now towards steam and ultrasonic systems due to the fear of risks to health. Some older system types are included in the tables because they may be found in existing buildings.

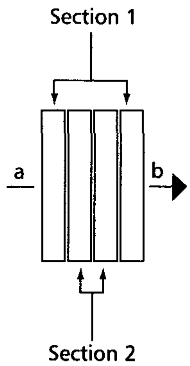
The tables distinguish between direct and indirect humidifiers:

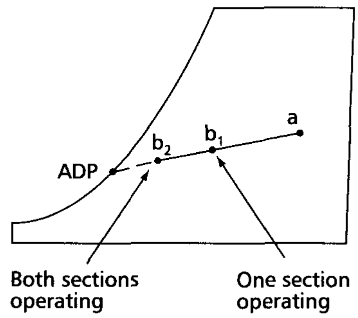
- ***Direct humidifiers:*** have a particular application in industrial fields. They discharge water particles or vapour directly into the space to be treated. The air in the space absorbs the moisture to a degree consistent with the air movement or turbulence

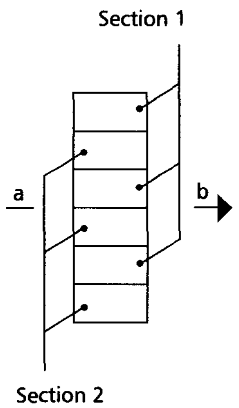
Table 5.14 Psychrometric effect of coil selection arrangement

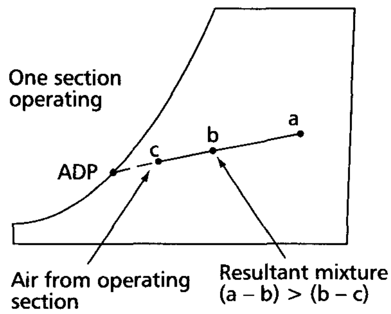
<table>
<tr><th>Schematic</th><th>Remarks</th><th>Psychrometric effect</th></tr>
<tr><td>


</td><td>Effect is that of complete bypassing of half the coil</td><td>


</td></tr>
<tr><td>


</td><td>Effect is to reduce contact factor of coil as though it consisted of fewer rows of pipes</td><td>


</td></tr>
<tr><td>


</td><td>Depending on coil construction and number of sections, part of finned surface in an inoperative section is partially effective in transferring heat to an operative section</td><td>


</td></tr>
</table>

and the fineness of the 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 filmy 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 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[44]:

- *Air washers* (and *evaporative coolers,* see below): found usually in large central air conditioning systems.
- *Wetted media:* used in residential and small commercial buildings (not in UK).
- *Water atomising:* having a wide range of applications as a result of their large capacity range.

Through an efficient humidifier the air can be cooled almost to its entering wet-bulb temperature and 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. These evaporative cooling systems are discussed further in section 4-12.

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[44]:

Table 5.15 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 disk	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

Table 5.16 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 requirement

- steam humidifiers: including those where the steam is produced remotely
- vapour generators: where heat energy is converted to water vapour within the apparatus itself.

See BSRIA AG10/94[44] and manufacturers' information for a more detailed evaluation of the advantages and disadvantages of the various approaches.

5.10.3 Direct humidifiers

5.10.3.1 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 in spray form.

5.10.3.2 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.

5.10.3.3 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.

5.10.3.4 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.

5.10.4 Indirect humidifiers

5.10.4.1 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 overall 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 $m.s^{-1}$. Efficiencies to be expected are 70% for a single-bank spray washer, 85% for a double-bank spray washer, and 95% for a triple-bank spray washer, with a distance between each bank of about 1 m.

Suggested rates of water flow are approximately 5 $litre.s^{-1}$ of water per 3 m^2 of face area of the spray chamber per bank of sprays, which is equivalent to approximately 7 $litre.s^{-1}$ 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.

5.10.4.2 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 air flow 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 air streams providing maximum contact between water and air, resulting in high efficiency of saturation. Most dust particles down to **3** pm in size are also eliminated from the air stream, 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 $m.s^{-1}$ with a maximum of 2 $m.s^{-1}$ through the cells.

Saturation efficiency of 97% 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.

5.10.4.3 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 $litre.s^{-1}$ 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.

5.10.4.4 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.

5.10.4.5 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.

5.10.4.6 Steam humidifiers

Steam provides a relatively simple and hygienic method of humidification providing that the heat in the system can be absorbed. Generally the use of main boiler steam is limited in application to industry due to the characteristic odour and traces of oil which may be present.

For application to ventilation systems, secondary steam can be generated at low or atmospheric pressure from mains steam, an electrode boiler or electrical resistance boiler.

5.10.4.7 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.

5.10.4.8 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.

5.10.4.9 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.

5.10.5 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 $m.s^{-1}$.

5.10.6 Humidifier positioning

Research[44] 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.

5.10.7 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[44].

5.10.8 Water supply

5.10.8.1 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.

5.10.8.2 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[45], Health and Safety Executive[46] and CIBSE[47,48]. The HSE's Approved Code of Practice (ACOP) L8[46] 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* is promoted within water temperatures in the range of 20-45 °C. Additional guidelines are available for humidifiers used for medical purposes[49].

To avoid risks, it is suggested that designers specify equipment that does not create a spray, i.e. steam or evaporative humidifiers[50]. 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 Regulations[51].

5.10.9 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 5.14[52]. 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% 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.

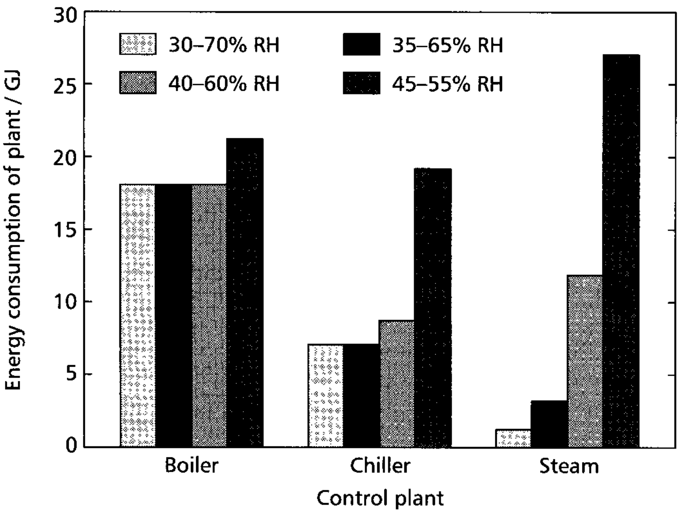


Figure 5.14 Energy use and effect of humidity control

5.11 Fans

5.11.1 General

Fans consume a large proportion of the total energy in mechanically ventilated buildings. A high priority should therefore be given to achieving energy efficient fan operation. Fan volumes and pressure drops should be minimised by good design. Benchmarks for fan volumes and pressure drops are provided in CIBSE Guide F[52]. The specific fan power should be 2 watts per litre.s^{-1} or less to achieve good practice in offices; very energy efficient systems can sometimes achieve around 1 watt per litre.s^{-1}. Consideration could be given to over-sizing parts of the system to reduce pressure drops, e.g. the air handling unit, as this is normally responsible for the majority of the losses.

Selection should favour the more efficient fan types and try to ensure that the fans will be operating at peak efficiency.

Volume control should be incorporated to meet varying levels of demand. This may be at the dictates of temperature, pressure or air quality sensors. Flow control may be achieved by a number of means including:

- on/off, multi-speed or variable speed operation
- varying the blade pitch for axial fans
- inlet guide vanes.

Dampers can also be used but are less energy efficient. See EEBPP General Information Report GIR 41[53] for further information on volume control.

5.11.2 Fan types and components

Table 5.17 provides a summary of fan types. The following definitions should be used in relation to fan types and components:

- *Casing:* those stationary parts of the fan which guide air to and from the impeller.
- *Guide vanes:* a set of stationary vanes, usually radial, on the inlet or discharge side of the impeller, covering the swept annulus of the impeller blades (or wings); their purpose is to correct the helical whirl of the airstream and thus raise the performance and efficiency of the fan.
- *Impeller:* that part of a fan which, by its rotation, imparts movement to the air.
- *Axial-flow fan:* a fan having a cylindrical casing in which the air enters and leaves the impeller in a direction substantially parallel to its axis.
- *Centrifugal or radial-flow fan:* a fan in which the air leaves the impeller in a direction substantially at right angles to its axis.
- *Cross-flow or tangential fan:* a fan in which the air is caused to flow through the impeller in a direction substantially at right angles to its axis both entering and leaving the impeller through the blade passages.
- *Mixed-flow fan:* a fan having a cylindrical casing and a rotor followed by a stator in which the air flowing through the rotor has both axial and radial velocity components.
- *Propeller fan:* a fan having an impeller other than of the centrifugal type rotating in an orifice; the air flow into and out of the impeller not being confined by any casing.

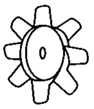
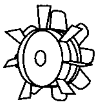
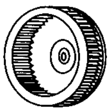
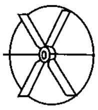
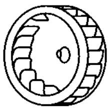
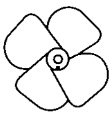
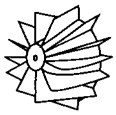
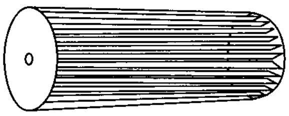
5.11.3 Fan performance

5.11.3.1 Definition of terms

Fan performance is expressed in terms of fan size, air delivery, pressure, speed and power input at a given air density. Efficiency will be implied or specifically expressed. The size of a fan depends on the individual manufacturer's coding but is directly expressed as, or is a function of, either the inlet diameter or the impeller diameter. Other terms are defined in BS 4856[54] and BS 848: Part 1[55] as follows:

- *Reference air:* for the purposes of rating fan performance, reference air is taken as having a density of 1.200 kg.m^{-3}; this value corresponds to atmospheric air at a temperature of 20 °C, a pressure of 101.325 kPa and a relative humidity of 43%.
- *Fan total pressure:* the algebraic difference between the mean total pressure at the fan outlet and the mean total pressure at the fan inlet.
- *Fan velocity pressure:* the velocity pressure corresponding to the average velocity of the fan outlet based on the total outlet area without any deduction for motors, fairings or other bodies.
- *Fan static pressure:* the difference between the fan total pressure and the fan velocity pressure.
- *Fan duty (total):* the inlet volume dealt with by a fan at a stated fan total pressure.
- *Fan duty (static):* the inlet volume dealt with by a fan at a stated fan static pressure.

Table 5.17 Summary of fan types

Fan type	Efficiency / %		Advantages	Disadvantages	Applications
	Static	Total			
1 Axial-flow (without guide vanes 	50–65	50–75	Very compact, straight-through flow. Suitable for installing in any position in run of ducting.	High tip speed. Relatively high noise level comparable with type 5. Low pressure development.	All low pressure atmospheric air applications.
2 Axial-flow (with guide vanes) 	65–75	65–85	Straight-through flow. Suitable for vertical axis.	Same as type 1 but to lesser extent.	As for type 1, and large ventilation schemes such as tunnel ventilation.
3 Forward-curved or multivane centrifugal 	45–60	45–70	Operates with low peripheral speed. Quiet and compact.	Severely rising power characteristic requires large motor margin.	All low and medium pressure atmospheric air and ventilation plants.
4 Straight or paddle-bladed centrifugal 	45–55	45–70 60 (non-shrouded)	Strong, simple impeller. Least likely to clog. Easily cleaned and repaired.	Low efficiency. Rising power characteristic.	Material transport systems and any application where dust burden is high.
5 Backwards-curved or backwards-inclined blade centrifugal 	65–75	65–85	Good efficiency. Non-overloading power characteristic.	High tip speed. Relatively high noise level compared with type 3.	Medium and high pressure applications such as high velocity ventilation schemes.
6 Aerofoil-bladed centrifugal 	80–85	80–90	Highest efficiency of all fan types. Non-overloading fan characteristic	Same as type 5.	Same as type 5 but higher efficiency justifies use for higher power applications.
7 Propeller 	< 40	< 40	Low first cost and ease of installation.	Low efficiency and very low pressure development.	Mainly non-ducted low pressure atmospheric air applications. Pressure development can be increased by diaphragm mounting.
8 Mixed-flow 	45–70	45–70	Straight-through flow. Suitable for installing in any position in run of ducting. Can be used for higher pressure duties than type 2. Lower blade speeds than types 1 or 2, hence lower noise.	Stator vanes are generally highly loaded due to higher pressure ratios. Maximum casing diameter is greater than either inlet or outlet diameters.	Large ventilation schemes where the higher pressures developed and lower noise levels give an advantage over type 2.
9 Cross-flow or tangential-flow 	—	40–50	Straight across flow. Long, narrow discharge.	Low efficiency. Very low pressure development.	Fan coil units. Room conditioners. Domestic heaters.

- ***Air power (total):*** the product of the fan total pressure and the fan duty (total).
- ***Air power (static):*** the product of the fan static pressure and the fan duty (static).
- ***Shaft power:*** the energy input, per unit time, to the fan shaft including the power absorbed by such parts of the transmission system as constitute an integral part of the fan, e.g. fan shaft bearings.
- ***Fan total efficiency:*** the ratio of the air power (total) to the shaft power
- ***Fan static efficiency:*** the ratio of the air power (static) to the shaft power.

5.11.3.2 The fan laws

For a given system in which the total pressure loss is proportional to the square of the volume flow, the performance of a given fan at any changed speed is obtained by applying the first three rules (the air density is considered unchanged throughout):

- ***Rule 1:*** The inlet volume varies directly as the fan speed.
- ***Rule 2:*** The fan total pressure and the fan static pressure vary as the square of the fan speed.
- ***Rule 3:*** The air power (total or static) and impeller power vary as the cube of the fan speed.

For changes in density:

- ***Rule 4:*** The fan total pressure, the fan static pressure and the fan power all vary directly as the mass per unit volume of the air which in turn varies directly as the barometric pressure and inversely as the absolute temperature.

For geometrically similar airways and fans operating at constant speed and efficiency the performance is obtained by applying the following three rules (the air density is considered unchanged throughout):

- ***Rule 5:*** The inlet flowrate varies as the cube of the fan size.
- ***Rule 6:*** The fan total pressure and the fan static pressure vary as the square of the fan size.
- ***Rule 7:*** The air power (total or static) and impeller power vary as the fifth power of the fan size.

5.11.4 Types of fan

5.11.4.1 Axial-flow fans

Axial-flow fans comprise an impeller with a number of blades, usually of aerofoil cross section, operating in a cylindrical casing. The fineness of the tip clearance between impeller blades and casing has a marked effect on the pressure development of the fan and, in turn, its output and efficiency. The blades may also have 'twist', i.e. the pitch angle increases from tip to root.

The pitch cannot be increased beyond the stall point of the aerofoil and the centre of the impeller has to be blanked-off by a hub to avoid recirculation. The hub acts as a fairing for the motor. Large hubs and short blades characterise a high pressure to volume ratio, and vice versa. Refinements include guide vanes to correct whirl at inlet or discharge and fairings and expanders to recover a greater proportion of the velocity head in the blade swept annulus.

Axial-flow fans are of high efficiency and have limiting power characteristics, but as the highest pressure single-stage axial-flow fans develop only about one-fifth of the pressure produced by a forward curved (multi-vane) fan, they are best suited for high volume/pressure ratios. However, axial-flow fans may be staged or placed in series and when fitted with guide vanes the aggregate pressure developed is proportional to the number of stages for a given volume. A two-stage fan can be contra-rotating, and without the use of guide vanes the pressure developed may be up to 2.75 times greater than that of a single stage.

5.11.4.2 Centrifugal fans

Centrifugal fans comprise an impeller that rotates usually in an involute casing. The air flows into the impeller axially, turns through a right angle within it and is discharged radially by centrifugal force. The scroll acts as a collector that permits vortex flow to the casing outlet and converts some of the high velocity pressure at the blade tips into static pressure. There are several variations of the basic form, see below.

Forward-curved or multi-vane

The impeller has a relatively large number of short forward-curved blades. The air is impelled forward in the direction of rotation at a speed greater than the impeller tip speed. For a given duty this type of fan is the smallest of the centrifugal types. It operates with the lowest tip speed and is often referred to as a low-speed fan. As the velocity of the air does not decrease within the blade passages, the efficiency is not high and the motor can easily be overloaded if the system resistance is overestimated.

Straight-radial or paddle-blade

The impeller has a few (typically six) straight blades which may be fixed by the roots to a spider, or may have a back-plate and shroud-plate. This is the simplest, and least efficient, of all fan types but is well suited to applications where airborne material is present as the blades are unlikely to clog. The impeller is of high mechanical strength and is cheap to refurbish. Renewable blades or wear plates are often fitted.

Backwards curved blade

The air leaves the impeller at a speed less than the impeller tip speed and the rotational speed for a given duty is relatively high. The impeller has from ten to sixteen blades of curved or straight form, inclined away from the direction of rotation. Because the blades are deep, good expansion within the blade passages takes place and this, coupled with a relatively low air speed leaving the impeller, ensures high efficiency and a non-limiting power characteristic.

Aerofoil blade

This is a refinement of the backwards-curved fan in which the impeller blades are of aerofoil contour with a venturi throat inlet and fine running clearance between inlet and impeller. The casing is compact and the volumetric output is high. The static efficiency is the highest of all fans, but it is a relatively high-speed fan due to the low pressure development.

5.11.4.3 Propeller fans

Propeller fans comprise an impeller of two or more blades of constant thickness, usually of sheet steel, fixed to a centre boss and are designed for orifice or diaphragm mounting. They have high volumetric capacity at free delivery, but very low pressure development. However, this may be increased by fitting the fan in a diaphragm, which in turn may be installed in a circular or rectangular duct of area greater than the blade-swept area. The efficiency of propellor fans is low.

5.11.4.4 Cross-flow or tangential fans

These comprise a fomard-curved centrifugal type impeller but with greatly increased blade length and the conventional inlets blocked off. The impeller runs in a half casing with conventional discharge but no inlet. Air is scooped inwards through the blade passages on the free side, but at the opposite side of the impeller, due to the influence of the casing, the air obeys the normal centrifugal force and flows out of the impeller and through the fan discharge.

The principle of operation relies on the setting up of a long cylindrical vortex stabilised within the impeller which, being much smaller in diameter than the impeller, rotates at high angular velocity. This in turn drives the main airstream past the blades of the fan with higher velocity than the peripheral speed of the blades themselves. In effect the air flows 'across' the impeller, almost at right angles to the axis.

Because this fan is so different from other types direct comparisons are not valid. A serious disadvantage of this type is that it cannot be operated at shaft speeds widely different from that for which it has been designed. Consequently it obeys the fan laws only within narrow limits of speed change. It operates with a high discharge velocity and an expander is desirable when connected to ductwork, especially as the efficiency (which is rather less than that for the multi-vane fan) peaks at near-free delivery conditions. The discharge opening is characteristically narrow ***so*** the fan is not easily applicable to ducting but is well suited to fan coil units and electric space heaters.

5.11.4.5 Mixed-flow in-line fans

Mixed-flow fans comprise an impeller with a number of blades, often of aerofoil section, similar to the axial flow fan. The hub is of conical shape such that the passage of air through the impeller has both axial and radial components, hence the term mixed-flow. Mixed-flow fans are of high efficiency and can be designed for higher pressure duties than axial flow fans.

To remove the swirl generated by the passage of air through the impeller, stator guide vanes are fitted downstream. These vanes are generally highly loaded due to the high pressure ratios. If the inlet and outlet flanges are to be of the same diameter a change in casing profile is necessary in the region of the guide vanes. Separation of airflow can occur if the conditions for which the fan was designed are not maintained in practice.

5.11.4.6 Bifurcated fans

Bifurcated fans handle atmospheres normally detrimental to the life of the fan motor, including saturated and dust-laden atmospheres, heated air, hot gases and corrosive fumes. They are normally direct drive with the motor isolated from the system air stream.

5.12 Air control units

5.12.1 General

When various areas to be air conditioned have differing heat gain patterns with respect to time, these can be met 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 requirements of the area. Such temperature or volume contro1 may be carried out in ductwork serving a number of rooms or zones or may be carried out in the terminal units feeding individual rooms.

5.12.2 Control of volume

Volume control may be achieved by:

- *Damper:* normally of the butterfly or multileaf type and capable of controlling the volume, providing 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.
- *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 which is self-actuating and capable of automatically maintaining a constant pre-set volume through it, provided that the pressure drop across it is 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 airway, 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. It is factory pre-set to pass a specific volume and, when installed, will automatically give a pre-balanced air distribution system up to and including the terminal unit. It can be adjusted on-site, if desired.

5.12.3 Control of temperature

This may be achieved by:

- ***Blending:*** two separate airstreams, one warm, the other cool are supplied to a zone and mixed in a terminal unit to produce a supply air temperature which 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 give a resultant supply air temperature which will satisfy the zone requirement.

5.13 Air terminal devices

Air can be supplied to the space in a number of ways[56], the principal division being between diffusers and perpendicular jets. Airflow patterns for both types are strongly

Table 5.18 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, transfer	Side wall	7	10	Low free area. Designed to prevent through vision.
6	'Egg crate'	Extract	Ceiling, side wall	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 1-, 2- or 3-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, desk top, 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 luminaire.
11	Ventilated ceiling nozzel	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	Nozzels, 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 nozzels are used for high-induction applications. Velocities depend on throw, noise and induction requirements.

dependent upon the presence or absence of the Coanda effect, see section 4.2.3.5.

Table 5.18 summarises the types of air terminal devices, and provides information on typical face velocities (based on any local contro1 devices being fully open) and noise levels.

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

A perpendicular jet may be formed by discharging air through grilles, louvres, nozzles or any opening which allows perpendicular air flow; direction and spread adjustment can be provided using blades and or swivel adjustment.

Supply air terminals 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.

References

1 *Fundamentals* ASHRAE Handbook (Atlanta, GA: American Society of Heating, Refrigerating and Air-conditioning Engineers) (1997)

2 *Air quality limits and guide values for sulphur dioxide and suspended particulates* EEC Directive 80/779/EEC Official Journal L229 30 (August 1980)

3 Environmental Protection Act 1990 (London: Her Majesty's Stationery Office) (1990)

4 *Environmental design* CIBSE Guide A (London: Chartered Institution of Building Services Engineers) (1999)

5 *Minimising pollution at air intakes* CIBSE Technical Memoranda TM21 (London: Chartered Institution of Building Services Engineers) (1999)

6 BS EN 12207: *2000: Windows and doors. Air permeability. Classification.* (London: British Standards Institution) (2000)

7 BS 6375: *Performance ofwindows:* Part 1: 1989: *Classificationfor weathertightness* (including guidance on selection and specification) (London: British Standards Institution) (1989)

8 *Daylighting and window design* CIBSE Lighting Guide LGlO (London: Chartered Institution of Building Services Engineers) (1999)

9 *Natural ventilation in non-domestic buildings* CIBSE Applications Manual AM10 (London: Chartered Institution of Building Services Engineers) (1997)

10 Pitts A C and Georgiadis S Ventilation air flow through window openings in combination with shading devices *Proc. 15th. AIVC Conference, Buxton* (Coventry: Air Infiltration and Ventilation Centre) (1994)

11 Willis S, Fordham M and Bordass W *Avoiding or minimising the use of air conditioning* EEBPP General Information Report GIR 31 (London: Department of Transport, Local Government and the Regions) (1995) (available from http://www.energy-efficiency.gov.uk)

12 *Natural ventilation for offices* (Garston: Building Research Establishment) (1999)

13 *The Building Regulations 1991 Approved Document F: Ventilation* (London: Department of Transport, Local Government and the Regions/The Stationery Office) (1995)

14 *White* M *Trickle ventilators in offices* BRE Information Paper IP12/98 (Garston: Building Research Establishment) (1998)

15 *Sound control* CIBSE Guide B5 (London: Chartered Institution of Building Services Engineers) (in preparation)

16 White M K and Stephen R *Ventilators: ventilation and acoustic effectiveness* BRE Information Paper IP4/99 (Garston: Building Research Establishment) (1999)

17 prEN 13141-1: *Ventilation for buildings -Performance testing of components/products ofresidential ventilation: Part I: externally and internally mounted air transfer devices* (Brussels: Comité Européen de Normalisation) (1998) (available in UK through BSI)

18 Orme M, Liddament M W and Wilson A *An analysis and data summary of the AIVC's numerical database* AIVC Technical Note 44 (Coventry: Air Infiltration and Ventilation Centre) (1994)

19 BS 6399-2: 1997: *Loadingfor buildings. Code ofpractice for wind loads* (London: British Standards Institution) (1977)

20 Martin A J *Control ofnatural ventilation* BSRIA Technical Note TN11/95 (Bracknell: Building Services Research and Information Association) (1995)

21 Stewart L J *Design guidelines for exhaust hoods* BSRIA Technical Note TN3/85 (Bracknell: Building Services Research and Information Association) (1985)

22 *Industrial ventilation* - *a manual of recommended practice* (Lansing, MI: American Conference of Governmental Industrial Hygienists) (2001)

23 *Air-to-air recovery in ventilation* AIVC Technical Note 45 (Coventry: Air Infiltration and Ventilation Centre) (1994)

24 *Air-to-air energy recovery* Chapter 42 in ASHRAE Handbook: *HVAC Systems and Equipment* (Atlanta, GA: American Society of Heating, Refrigerating and Air-conditioning Engineers) (2000)

25 *Building energy and environmental modelling* CIBSE Applications Manual AM1 1 (London: Chartered Institution of Building Services Engineers) (1999)

26 *New technologies for heating and cooling supply in offices* CADDET Analyses Series No.3 (Harwell: Centre for the Analysis and Dissemination of Demonstrated Energy Technologies) (1990) (http://www.caddet.co.uk)

27 *Applied heat pump and heat recovey systems* Chapter 8 in ASHRAE Handbook: *HVAC Systems and Equipment* (Atlanta, GA: American Society of Heating, Refrigerating and Air-conditioning Engineers) (2000)

28 Hamilton G *Selection of air-to-air heat recovery systems* BSRIA Technical Note TN11/86 (Bracknell: Building Services Research and Information Association) (1986)

29 *Air-to-air heat recovey* CIBSE Research Report RR2 (London: Chartered Institution of Building Services Engineers) (1995)

30 BS EN 779: 1993: *Particulate air filters for general ventilation* - *Requirements, testing, marking* (London: British Standards Institution) (1993)

31 BS 6540: *Airfilters used in air Conditioning and general ventilation:* Part 1: 1985: *Methods oftestfor atmospheric dust spot efficiency and synthetic dust weight arrestance* (London: British Standards Institution) (1985) (withdrawn; replaced by BS EN 779(30)

32 BS 3928: 1969: *Methodfor sodium flame test for air filters (other than for air supply to I.C. engines and compressors)* (London: British Standards Institution) (1969)

33 BS EN 1822-1:1998: *High efficiency airfilters **(HEPA** and ULPA). Classification, performance testing, marking;* BS EN 1822-2:1998: *High efficiency air filters **(HEPA** and ULPA). Aerosol production, measuring equipment, particle counting statistics;* BS EN 1822-3:1998: *High efficiency air filters **(HEPA** and ULPA). Testing flat sheetfilter media;* BS EN 1822-4:2000: *High efficiency airfilters **(HEPA** and ULPA). Determining leakage of filter element (scan method);* BS EN 1822-5:2000: *High efficiency airfilters **(HEPA** and ULPA). Determining the efficiency of filter element* (London: British Standards Institution) (dates as indicated)

34 *Sodium chloride aerosol tests for filters using flame photometric technique* Eurovent 4/4 (Paris: Eurovent/Cecomaf) **(1984)**

35 Gilbert A *Laboratory testing of air cleaners* BSRIA Specification **SS 22/99** (Bracknell: Building Services Research and Information Association) **(1999)**

36 ***Y 42:*** *Air filtration* in National Engineering Specification (Windsor: National Engineering Specification) **(1996)**

37 Pike P G *Air filters* BSRIA Application Guide **AG8/97** (Bracknell: Building Services Research and Information Association)**(1996)**

38 Bennett K M *Air filters* BSRIA Selection Guide **SG7/91** (Bracknell: Building Services Research and Information Association)**(1991)**

39 *Hygienic maintenance of office ventilation ductwork* CIBSE **TM26** (London: Chartered Institution of Building Services Engineers)**(2000)**

40 *Air quality strategy for England, Scotland, Wales and Northern Ireland* **-** *Working together for clean air* CM **4548** (London: The Stationery Office) **(2000)**

41 *Standard maintenance specification for mechanical services in buildings: Vol. II: Ventilating and air conditioning* (London: Heating and Ventilating Contractor's Association) **(1991)**

42 *A guide to energy efficient ventilation* (Coventry: Air Infiltration and Ventilation Centre) **(1996)**

43 *Building control systems* CIBSE Guide H (London: Chartered Institution of Building Services Engineers) **(2000)**

44 Bennett K M *Humidification in buildings* BSRIA Applications Guide AG10/94 (Bracknell: Building Services Research and Information Association) **(1991)**

45 *Water fittings and materials directory* (Slough: Water Research Centre) (updated half-yearly)

46 *Prevention and control of legionellosis (including Legionnaires' disease)* Approved Code of Practice L8 (London: Health and Safety Executive) (1995)

47 *Public health engineering* CIBSE Guide G (London: Chartered Institution of Building Services Engineers) **(1999)**

48 *Minimising the risk of legionnaires disease* CIBSE **TM13** (London: Chartered Institution of Building Services Engineers) **(2000)**

49 BS EN ISO **8185: 1998:** *Humidifiers for medical use. General requirements for humidification systems* (London: British Standards Institution) **(1998)**

50 *The control of legionellosis including Legionnaires' disease* HS(G)70 (London: Health and Safety Executive) **(1991)**

51 The Water Supply (Water Fittings) Regulations **1999** Statutory Instrument **1999** No. **1148** (London: The Stationery Office) **(1999)**

52 *Energy efficiency in buildings* CIBSE Guide **F** (London: Chartered Institution of Building Services Engineers) **(1998)**

53 *Variable flow* EEBPP General Information Report GIR **41** (London: Department of Transport, Local Government and the Regions) (various dates) (available from http://www.energy-efficiency.gov.uk)**(1996)**

54 BS 4856: *Methods for testing and rating fan coil units, unit heaters and unit coolers:* Part 1: **1972:** *Thermal and volumetric performance for heating duties; without additional ducting;* Part **2: 1975:** *Thermal and volumetric performance for cooling duties; without additional ducting;* Part **3: 1975:** *Thermal and volumetric performance for heating and cooling duties; with additional ducting;* Part **4: 1997:** *Determination of sound power levels of fan coil units, unit heaters and unit coolers using reverberating rooms* (London: British Standards Institution) (dates as indicated)

55 BS 848: *Fans for general purposes:* Part 1: 1997: *Performance testing using standardized airways* (London: British Standards Institution) **(1997)**

56 *Guide to air distribution technology for the internal environment* (Marlow: HEVAC Association) **(2000)**

Index