Heating

CIBSE Guide B1

CIBSE

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CORRIGENDUM (CIBSE Guide B1)
The following table replaces Table 6.2 on page 6-2

Table 6.2 Key properties of typical petroleum fuels

Property	Class C2	Class D	Class E	Class F	Class G
Density at 15 °C ($kg \cdot m^{-3}$)	803	850	940	970	980
Minimum closed flash point (°C)	38	60	66	66	66
Kinematic viscosity ($mm^2 \cdot s^{-1}$) at 40 °C	1.0 to 2.0	1.5 to 5.5	—	—	—
Kinematic viscosity ($mm^2 \cdot s^{-1}$) at 100 °C	—	—	≤8.2	≤20.0	≤40.0
Maximum pour point (°C)	—	—	−6	24	30
Gross calorific value ($MJ \cdot kg^{-1}$)	46.4	45.5	42.5	41.8	42.7
Net calorific value ($MJ \cdot kg^{-1}$)	43.6	42.7	40.1	39.5	40.3
Maximum sulphur content by mass (%)	0.2	0.2	3.2	3.5	3.5
Mean specific heat 0–100 °C ($MJ \cdot kg^{-1}$)	2.1	2.06	1.93	1.89	1.89

CIBSE: correction issued June 2004

CORRIGENDUM (Guide B1)
The following table replaces Table 6.3 on page 6-4

Table 6.3 Carbon emission factors for UK in 2000–2005[6]

Fuel	Carbon emission per unit of delivered energy/ $kgC{\cdot}(kW{\cdot}h)^{-1}$
Natural gas	0.053
LPG	0.068
Gas oil/burning oil	0.074
Coal	0.086
Electricity (average of public supply)	0.113

CIBSE: correction issued June 2004

Foreword

This 2002 edition of CIBSE Guide B1 is a completely new guide, thoroughly revised in both format and content from the 1986 Guide B. The format now follows a logical progression through from initial strategic decision making to product selection and echoes the format of the recently published Guide B2. The technical content has been significantly updated, with little of the earlier Guide retained, primarily due to the changes in products and practice in the intervening years.

The production of this Guide has benefited from the use of a professional author, guided by a volunteer task group. I would liketo express my thanks to the author for his dedication and hard work, to the task group for their time, effort, invaluable advice and guidance and to the Guide B steering group, contributors, reviewers and CIBSE staff for their valuable contributions.

Finally, we hope that all users will find this Guide a useful and authoritative source of information and guidance.

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Note from the publisher

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

Contents

1 Introduction

This Guide starts by considering the strategic choices facing the heating system designer, including the requirements imposed by the intended use of the building, energy and environmental targets, legal requirements and possible interaction with other building services. The succeeding sections follow the various stages of design, as follows:

- detailed definition of requirements and the calculation of system loads
- characteristics and selection of systems
- characteristics and selection of system components and equipment
- characteristics of fuels and their requirements for storage
- commissioning and hand-over.

Section 2, which deals with strategic choices, is relatively broad ranging and discursive and is intended to be read from time to time as a reminder of the key decisions to be taken at the start of the design process. The latter sections are sub-divided by topic and are likely to be used for reference, as particular issues arise; they contain a range of useful details but also direct the reader to more specialised sources where appropriate, including other CIBSE publications and BS, EN, and IS0 standards.

When using this Guide, the designer should firstly fully map the design process that is being undertaken. The process for each application will be unique, but will follow the general format:

- problem definition;
- ideas generation;
- analysis; and
- selection of the final solution.

This procedure is illustrated in Figure 1.1 in the form of a outline flowchart.

Reference

1 *Whole Life Costing: A client's guide* Construction Clients *Forum* 2002 (London: Confederation of Construction Clients) (2000)

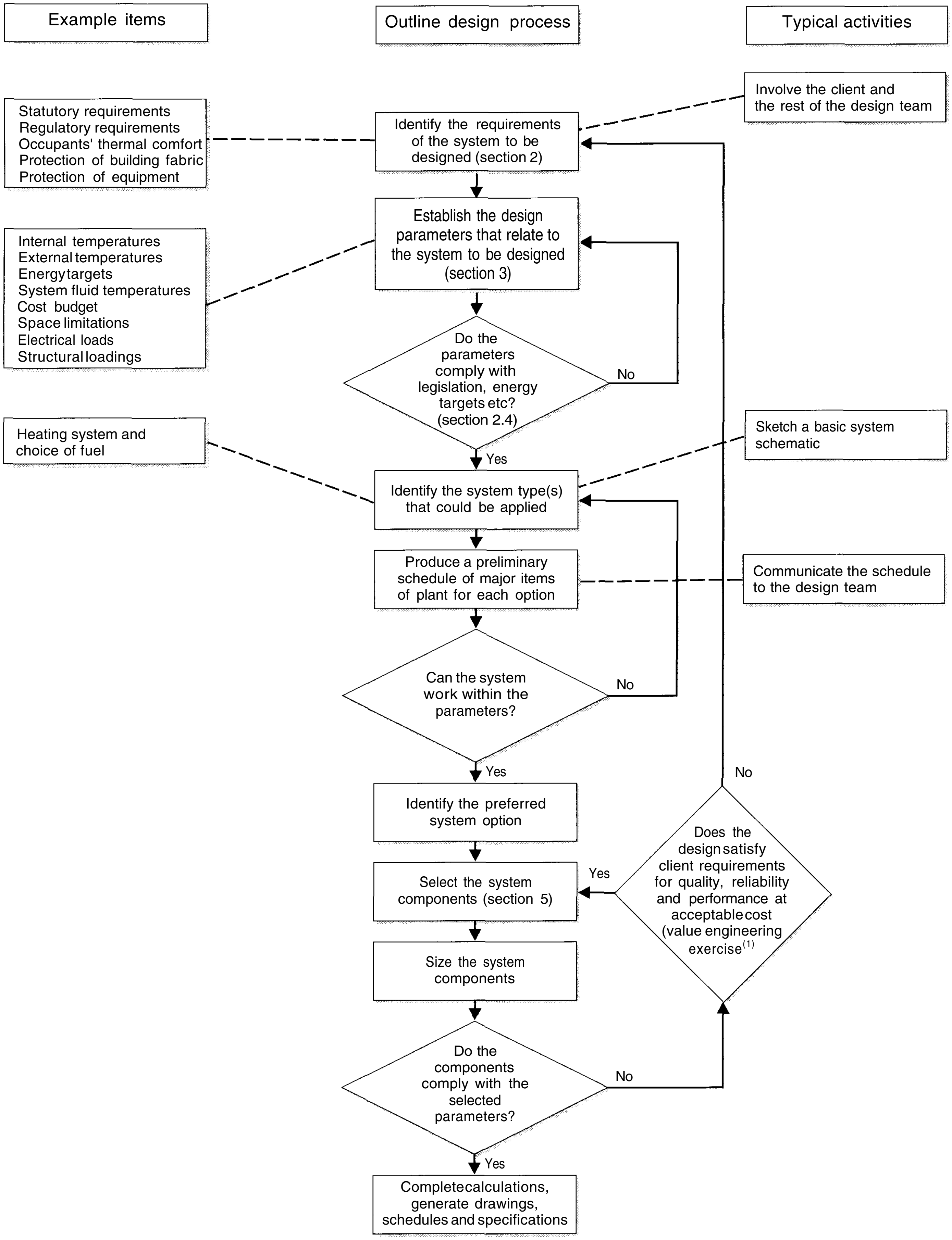

Figure 1.1 Outline design process

2 Strategic design decisions

2.1 General

In common with some other aspects of building services, the requirements placed upon the heating system depend crucially on the form and fabric of the building. It follows that the role of the building services engineer in heating system design is at its greatest when it begins at an early stage, when decisions about the fabric of the building can still be influenced. This allows options for heating to be assessed on an integrated basis that takes account of how the demand for heating is affected by building design as well as by the provision of heating. In other cases, especially in designing replacement heating systems for existing buildings, the scope for integrated design may be much more limited. In all cases, however, the designer should seek to optimise the overall design as far as is possible within the brief.

A successful heating system design will result in a system that can be installed and commissioned to deliver the indoor temperatures required by the client. When in operation, it should operate with high efficiency to minimise fuel costs and environmental emissions while meeting those requirements. It should also sustain its performance over its planned life with limited need for maintenance and replacement of components. Beyond operational and economic requirements, the designer must comply with legal requirements, including those relating to environmental impact and to health and safety.

2.2 Purposes of space heating systems

Heating systems in most buildings are principally required to maintain comfortable conditions for people working or living in the building. As the human body exchanges heat with its surroundings both by convection and by radiation, comfort depends on the temperature of both the air and the exposed surfaces surrounding it and on air movement. Dry resultant temperature, which combines air temperature and mean radiant temperature, has generally been used for assessing comfort. The predicted mean vote (PMV) index, as set out in the European Standard BS EN 7730(1), incorporates a range of factors contributing to thermal comfort. Methods for establishing comfort conditions are described in more detail in section 3.2 below.

In buildings (or parts of buildings) that are not normally occupied by people, heating may not be required to maintain comfort. However, it may be necessary to control temperature or humidity in order to protect the fabric of the building or its contents, e.g. from frost or condensation, or for processes carried out within the building. In either case, the specific requirements for each room or zone need to be established.

2.3 Site related issues

The particular characteristics of the site need to be taken into account, including exposure, site access and connection to gas or heating mains. Exposure is taken into account in the calculation of heat loss (see section 3.3 below). The availability of mains gas or heat supplies is a key factor affecting the choice of fuel.

The form and orientation of buildings can have a significant effect on demand for heating and cooling. If the building services designer is involved early enough in the design process, it will be possible to influence strategic decisions, e.g. to optimise the 'passive solar' contribution to energy requirements

2.4 Legal, economic and general considerations

various strands of legislation affect the design of heating systems. Aspects of the design and performance of heating systems are covered by building regulations aimed at the conservation of fuel and power(2-4) and ventilation(3-5); and regulations implementing the EU Boiler Directive(6) set minimum efficiency levels for boilers. Heat producing appliances are also subject to regulations governing supply of combustion air, flues and chimneys, and emissions of gases and particles to the atmosphere[7], see section 5.5.1. Designers should also be aware of their obligations to comply with the Construction (Design and Management) regulations[8,9] and the Health and Safety at Work etc. Act[10].

Beyond strictly legal requirements, the client may wish to meet energy and environmental targets, which can depend strongly on heating system performance. These include:

- CIBSE *Building Energy Codes*[11] define a method for setting energy targets.
- Carbon performance rating/carbon intensity: although primarily intended as a means of showing compliance with Part L of the Building Regulations(*), 'carbon performance rating' (CPR) and 'carbon intensity' may be used more widely to define performance. CPR applies to the overall energy performance of office buildings with air conditioning and mechanical ventilation. Carbon intensity applies to heating systems generally.
- Broader ranging environmental assessments also take energy use into account, e.g. Building Research Environmental Assessment Method(12) (BREEAM) sets a series of best practice criteria against which aspects of the environmental performance of a building can be assessed. A good BREEAM rating

also depends strongly on the performance of the heating system.

— Clients who own and manage social housing may also have 'affordable warmth' targets, which aim to ensure that low income households will not find their homes too expensive to heat. The UK government's *Standard Assessment Procedure for the Energy Rating of Dwellings*[13] (SAP) and the *National Home Energy Rating*[14] (NHER) are both methods for assessing the energy performance of dwellings.

Economic appraisal of different levels of insulation, heating systems, fuels, controls should be undertaken to show optimum levels of investment according to the client's own criteria, which may be based on a simple payback period, or a specified discount rate over a given lifetime. Public sector procurement policies may specifically require life cycle costing

2.5 Interaction with building design, building fabric, services and facilities

As noted above, the earlier the heating system designer can be involved in the overall design process, the greater the scope for optimisation. The layout of the building, the size and orientation of windows, the extent and location of thermal mass within the building, and the levels of insulation of the building fabric can all have a significant effect on demand for heat. The airtightness of the building shell and the way in which the building is ventilated are also important. Buildings that are very well insulated and airtight may have no net heating demand when occupied, which requires heating systems to be designed principally for pre-heating prior to occupancy[15].

However, the designer is often faced with a situation in which there is little or no opportunity to influence important characteristics of the building that have a strong bearing on the heating system, particularly in the replacement of an existing heating system. For example, there may be constraints on the area and location of plant rooms, the space for and the routing of distribution networks. There may also be a requirement to interface with parts of an existing system, either for heating or ventilation. Where domestic hot water is required, a decision is required on whether it should be heated by the same system as the space heating or heated at the point of use.

2.6 Occupancy

When the building is to be occupied and what activities are to be carried out within it are key determinants of the heating system specification. Are the occupants sedentary or physically active? What heat gains are expected to arise from processes and occupancy, including associated equipment such as computers and office machinery? Do all areas of the building have similar requirements or are there areas with special requirements? These factors may determine or at least constrain the options available. The anticipated occupancy patterns may also influence the heating design at a later stage. Consideration should also be given to flexibility and adaptability of systems, taking account of possible re-allocation of floor space in the future.

2.7 Energy efficiency

The term 'energy efficiency' gained currency during the 1980s and is now widely used.

In general, the energy efficiency of a building can only be assessed in relative terms, either based on the previous performance of the same building or by comparison with other buildings. Thus the energy use of a building might be expressed in terms of annual energy use per square metre of floor area, and compared with benchmark levels for similar buildings. The result so obtained would depend on many physical factors including insulation, boiler efficiency, temperature, control systems, and the luminous efficacy of the lighting installations, but it would also depend on the way the occupants interacted

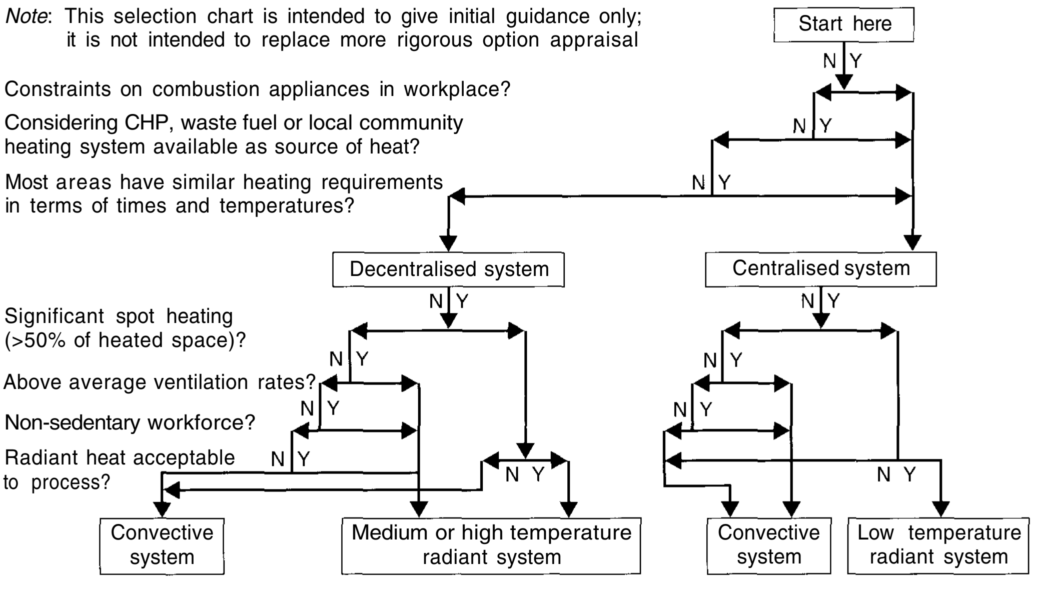

Figure 2.1 Selection chart: heating systems[17] (reproduced from EEBPP Good Practice Guide GPG303 by permission of the Energy Efficiency Best Practice Programme)

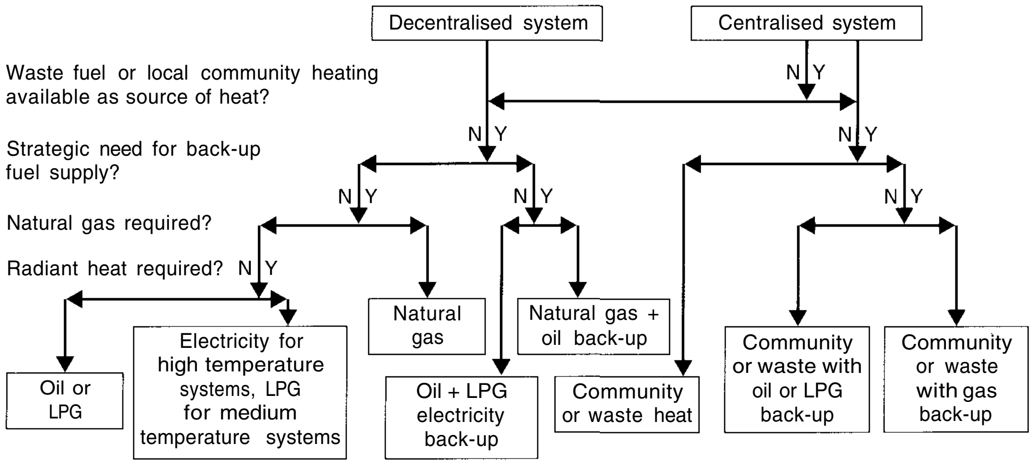

Figure 2.2 Selection chart: fuel[17] (reproduced from EEBPP Good Practice Guide GPG303 by permission of the Energy Efficiency Best Practice Programme)

with the building, particularly if it were naturally ventilated with openable windows.

The energy consumption of buildings is most readily measured in terms of 'delivered' energy, which may be read directly from meters or from records of fuels bought in bulk. Delivered energy fails to distinguish between electricity and fuel which has yet to be converted to heat. 'Primary' energy includes the overheads associated with production of fuels and with the generation and distribution of electricity. Comparisons of energy efficiency are therefore sometimes made on the basis of primary energy or on the emissions of 'greenhouse' gases, which also takes account of energy overheads. Fuel cost may also be used and has the advantage of being both more transparent and more relevant to non-technical building owners and occupants. In any event, it is meaningless to quote energy use in delivered energy obtained by adding electricity use to fuel use. Consequently, if comparisons are to be made in terms of delivered energy, electricity and fuel use must be quoted separately.

Clearly, the performance of the heating system has a major influence on energy efficiency, particularly in an existing building with relatively poor insulation. The designer has the opportunity to influence it through adopting an appropriate design strategy and choice of fuel, by specifying components with good energy performance, and by devising a control system that can accurately match output with occupant needs. Particular aspects of energy efficiency are dealt with in other sections of this Guide as they arise.

The energy efficiency of heating and hot water systems is dealt with in detail in section 9 of CIBSE Guide F: *Energy efficiency in buildings*[16].

2.8 Making the strategic decisions

Each case must be considered on its own merits and rigorous option appraisal based on economic and environmental considerations should be undertaken. However, the flow charts shown in Figures 2.1 and 2.2 are offered as general guidance. They first appeared in Good Practice Guide GPG303[17], which was published under the government's Energy Efficiency Best Practice programme and was aimed specifically at industrial buildings, but they are considered to be generally applicable. Figure 2.1 refers to heating systems in general and Figure 2.2 to choice of fuel.

References

1 BS EN ISO 7730: *Moderate thermal environments. Determination of PMV and PPD indices and specification of the conditions for thermal comfort* (London: British Standards Institution) (1995)

2 The Building Regulations 2000 Approved Documents L1 and L2: *Conservation of fuel and power* (London: The Stationery Office) (2001)

3 *Technical standards for compliance with the Building Standards (Scotland) Regulations* 1990 *(as amended)* (Edinburgh: Scottish Executive) (2001)

4 The Building Regulations (Northern Ireland) 2000 Amendments booklet AMDZ *Amendments to technical booklets* (Belfast: The Stationery Office) (2000) (Republic of Ireland Building Regulation *Technical Guidance Document L* covers energy requirements in buildings in Ireland.)

5 The Building Regulations 2000 Approved Document F: *Ventilation* (London: The Stationery Office) (2001) (Republic of Ireland Building Regulations *Technical Guidance Document F* covers ventilation requirements for buildings in Ireland.)

6 *The Boiler (Efficiency) Regulations* 1993 Statutory Instrument 1993 No. 3083 and *The Boiler (Efficiency)(Amendment) Regulations* 1994 Statutory Instrument 1994 No. 3083 (London: Her Majesty's Stationery Office) (1993 and 1994) (In the Republic of Ireland the EU Boiler Directive is implemented by the *European Communities (Efficiency requirements for hot water boilers fired with liquid and gaseous fuels) Regulations 1994.)*

7 The Building Regulations 2000 Approved Document J: *Combustion appliances and fuel storage systems* (London: The Stationery Office) (2001) (Republic of Ireland Building Regulations *Technical Guidance Document J* covers combustion appliances in Ireland.)

8 Construction (Design and Management) Regulations 1994 Statutory Instrument 1994 No. 3140 (London: Her Majesty's Stationery Office) (1994)

9 *CDM Regulations – Work sector guidance for designers* CIRIA Report 166 (London: Construction Industry Research and Information Association) (1997)

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

11 *Energy demands and targets for heated and ventilated buildings CIBSE Building Energy Code 1; Energy demands for air conditioned buildings CIBSE* Building Energy Code 2 (London: Chartered Institute of Building Services Engineers) (1999)

12 Baldwin R, Yates A, Howard N and Rao S *BREEAM 98 for offices* BRE Report BR350 (Garston: Building Research Establishment) *(1998)* (versions also available for housing, industrial units and retail premises)

13 *The Government's Standard Assessment Procedure for Energy Rating* of *Dwellings* (Garston: Building Research Establishment) (2001) (http://projects.bre.co.uk/sap200l/)

14 *National Home Energy Rating* (Milton Keynes: National Energy *Foundation)*

15 *HVAC strategies for well-insulated airtight buildings* CIBSE TM29 (London: Chartered Institution of Building Services Engineers) (2002)

16 *Energy efficiency in buildings* CIBSE Guide F (London: Chartered Institution of Building Services Engineers) (1998)

17 *The designer's guide to energy-efficient buildings for industry* Energy Efficiency Best Practice Programme Good Practice Guide GPG303 (Garston: Energy Efficiency Best Practice Programme) (2000)

3 Design criteria

3.1 General

After taking the principal strategic decisions on which type of system to install, it is necessary to establish design criteria for the system in detail. Typically this starts by defining the indoor and outdoor climate requirements and the air change rates required to maintain satisfactory air quality. A heat balance calculation may then be used to determine the output required from the heating system under design condition, which in turn defines the heat output required in each room or zone of the building. This calculation may be done on a steady-state or dynamic basis. As the latter type of calculation can lead to extreme complexity, simplified methods have been devised to deal with dynamic effects, such as those described in CIBSE Guide A[1], section 5.6. Dynamic simulation methods using computers are necessary when dynamic responses need to be modelled in detail. In all cases, however, underlying principles are the same - the required output from the heating system is calculated from consideration of the outflow of heat under design conditions, whether static or dynamic.

3.2 Internal climate requirements

Indoor climate may be defined in terms of temperature, humidity and air movement. The heat balance of the human body is discussed in CIBSE Guide A, section 1.4. The human body exchanges heat with its surroundings through radiation and convection in about equal measure. Thus the perception of thermal comfort depends on the temperature of both the surrounding air and room surfaces. It also depends upon humidity and air movement. When defining temperature for heating under typical occupancy conditions, the generally accepted measure is the dry resultant temperature, given by:

$$t_c = \{t_{ai}\surd(10\,v) + t_r\}/\{1+\surd(10\,v)\} \qquad (3.1)$$

where t_c is the dry resultant temperature (°C), t_{ai} is the inside air temperature (°C), t_r is the mean radiant temperature (°C) and v is the mean air speed (m.s^{-1}).

For $v < 0.1$ m.s^{-1}:

$$t_c = (0.5\ t_{ai} + 0.5\ t_r) \qquad (3.2)$$

As indoor air velocities are typically less than 0.1 mas-1, equation 3.2 generally applies.

Table 3.1 gives recommended winter dry resultant temperatures for a range of building types and activities. These are taken from CIBSE Guide A(1) section 1 and assume typical activity and clothing levels. Clients should be consulted to establish whether there any special requirements, such as non-typical levels of activity or clothing. Guide A section 1 includes methods for adjusting the dry resultant temperature to take account of Such requirements.

For buildings with moderate to good levels of insulation, which includes those constructed since insulation requirements were raised in the 1980, the difference between air and mean radiant temperature is often small enough to be insignificant for the building as a whole. Nevertheless, it is important to identify situations where these temperatures differ appreciably since this may affect the output required from heating appliances. As a general rule, this difference is likely to be significant when spaces are heated non-uniformly or intermittently. For some appliances, e.g. fan heater units, the heat output depends only on the difference between air temperature and heating medium temperature. For other types of appliance, e.g. radiant panels, the emission is affected by the temperature of surrounding surfaces. Section 3.3.3 be1ow dea1s with this subject in greater detai1*

Temperature differences within the heated space may also affect the perception of thermal comfort. Vertical temperature differences are likely to arise from the buoyancy of warm air generated by convective heating. In general it is recommended that the vertical temperature difference should be no more than 3 K between head and feet. If air velocities are higher at floor level than across the upper part of the body, the gradient should be no more than 2 K.m-1. Warm and cold floors may also cause discomfort to the feet. In general it is recommended that floor temperatures are maintained between 19 and 26 °C, but that may be increased to 29 oc for under-floor heating systems.

Asymmetric thermal radiation is a potential cause of thermal discomfort. I~ typically arises from:

— proximity to cold surfaces, such as windows
— proximity to hot surfaces, such as heat emitters, light sources and overhead radiant heaters
— exposure to solar radiation through windows.

CIBSE Guide A recommends that radiant temperature asymmetry should result in no more than 5% dissatisfaction, which corresponds approximately to vertical radiant asymmetry (for a warm ceiling) of less than 5 K and horizontal asymmetry (for a cool wall) of less than 10 K. The value for a cool ceiling is 14 K and for a warm wall is 23 K. It also gives recommended minimum comfortable distances from the centre of single glazed windows of different sizes.

In buildings that are heated but do not have full air conditioning, control of relative humidity is possible but unusual unless there is a specific process requirement. Even where humidity is not controlled, it is important to take account of the range of relative humidity that is likely

Table 3.1 Recommended winter dry resultant temperatures for various buildings and activities[1]

Building/room type	Temperature (°C)	Building/room type	Temperature (°C)
Airport terminals		Hotels	
— baggage reclaim	12–19	— bathrooms	26–27
— check–in areas	18–20	— bedrooms	19–21
— customs areas	12–19		
— departure lounges	19–21	Ice rinks	12
Banks, building societies and post offices		Laundries	
— counters	19–21	— commercial	16–19
— public areas	19–21	— launderettes	16–18
Bars, lounges	20–22	Law courts	19–21
Churches	19–21	Libraries	
Computer rooms	19–21	— lendingheference rooms	19–21
Conference/boardrooms	22–23	— reading rooms	22–23
		— store rooms	15
Drawing offices	19–21	Museums and art galleries	
Dwellings		— display	19–21
— bathrooms	26–27	— storage	19–21
— bedrooms	17–19		
— hall/stairs/landing	19–24	Offices	
— kitchen	17–19	— executive	21–23
— living rooms	20–23	— general	21–23
— toilets	19–21	— open plan	21–23
Educational buildings		Public assembly buildings	
— lecture halls	19–21	— auditoria	22–23
— seminar rooms	19–21	— changing/dressing rooms	23–24
— teaching spaces	19–21	— circulation spaces	13–20
Exhibition halls	19–21	— foyers	13–20
Factories		Prison cells	19–21
— heavy work	11–14		
— lightwork	16–19	Railway/coach stations	
— sedentary work	19–21	— concourse (no seats)	12–19
Fire/ambulance stations		— ticket office	18–20
— recreation rooms	20–22	— waiting room	21–22
— watch room	22–23	Restaurant/dining rooms	22–24
Garages		Retail buildings	
— servicing	16–19	— shopping malls	19–24
General building areas		— small shops, department stores	19–21
— corridors	19–21	— supermarkets	19–21
— entrance halls	19–21	Sports halls	
— kitchens (commercial)	15–18	— changing rooms	22–24
— toilets	19–21	— hall	13–16
— waiting areas/rooms	19–21	Squash courts	10–12
Hospitals and health care		Swimming pools	
— bedheaddwards	22–24	— changing rooms	23–24
— circulation spaces (wards)	19–24	— pool halls	23–26
— consulting/treatment rooms	22–24		
— nurses stations	19–22		
— operating theatres	17–19	Television studios	19–21

to be encountered in the building, particularly in relation to surface temperatures and the possibility that condensation could occur under certain conditions.

Also, account should be taken of air movement, which can have a significant effect on the perception of comfort. Where the ventilation system is being designed simultaneously, good liaison between the respective design teams is essential to ensure that localised areas of discomfort are avoided through appropriate location of ventilation outlets and heat emitters, see Guide B2: *Ventilation and air conditioning*[2]. For a building with an existing mechanical ventilation system, heating system design should also take account of the location of ventilation supply outlets and the air movements they produce.

The level of control achieved by the heating system directly affects occupant satisfaction with the indoor environment, see CIBSE Guide A, section 1.4.3.5. Although other factors also contribute to satisfaction (or dissatisfaction), the ability of the heating system and its controls to maintain dry resultant temperature close to design conditions is a necessary condition for satisfaction. Further guidance on comfort in naturally ventilated buildings may be found in CIBSE Applications Manual AM10: *Natural ventilation in non-domestic buildings*[3]. The effect of

Table 3.2 Suggested design temperatures for various UK locations

Location	Altitude (m)	Design temperature*/ °C	
		Low thermal inertia	High thermal inertia
Belfast (Aldegrove)	68	−3	−1.5
Birmingham (Elmdon)	96	−4.5	−3
Cardiff (Rhoose)	67	−3	−2
Edinburgh (Turnhouse)	35	−4	−2
Glasgow (Abbotsinch)	5	−4	−2
London (Heathrow)	25	−3	−2
Manchester (Ringway)	75	−4	−2
Plymouth (Mountbatten)	27	−1	0

* Based on the lowest average temperature over a 24- or 48-hour period likely to occur once per year on average (derived from histograms in Guide A, section 2.3)

temperatures on office worker performance is addressed in CIBSE TM24: *Environmental factors affecting office worker performance*[4].

Close control of temperature is often impractical in industrial and warehouse buildings, in which temperature variations of ±3 K may be acceptable. Also, in such buildings the requirements of processes for temperature control may take precedence over human comfort.

3.3 Design room and building heat loss calculation

3.3.1 Calculation principles

The first task is to estimate how much heat the system must provide to maintain the space at the required indoor temperature under the design external temperature conditions. Calculations are undertaken for each room or zone to allow the design heat loads to be assessed and for the individual heat emitters to be sized.

3.3.2 External design conditions

The external design temperature depends upon geographical location, height above sea level, exposure and thermal inertia of the building. The method recommended in Guide A is based on the thermal response characteristics of buildings and the risk that design temperatures are exceeded. The degree of risk may be decided between designer and client, taking account of the consequences for the building, its occupants and its contents when design conditions are exceeded.

CIBSE Guide A section 2.3 gives guidance on the frequency and duration of extreme temperatures, including the 24- and 48-hour periods with an average below certain thresholds. It also gives data on the coincidence of low temperatures and high wind speeds. The information is available for a range of locations throughout the UK for which long term weather data are available.

The generally adopted external design temperature for buildings with low thermal inertia (capacity), see section 3.3.7, is that for which only one day on average in each heating season has a lower mean temperature. Similarly for buildings with high thermal inertia the design temperature selected is that for which only one two-day spell on average in each heating season has a lower mean temperature. Table 3.2 shows design temperatures derived on this basis for various location in the UK. In the absence of more localised information, data from the closest tabulated location may be used, decreased by 0.6 K for every 100 m by which the height above sea level of the site exceeds that of the location in the table. To determine design temperatures based on other levels of risk, see Guide A, section 2.3.

It is the mass in contact with the internal air which plays a dominant role in determining whether a particular structure should be judged to be of low or high thermal inertia. Where carpets and false ceilings are installed, they have the effect of increasing the speed of response of the zone, which makes it behave in a manner more akin to that of a structure of low thermal inertia. Practical guidance may be found in Barnard et al[5] and in BRE Digest 454[6] . In critical cases, dynamic thermal modelling should be undertaken.

The thermal inertia of a building may be determined in terms of a thermal response factor, f_r , see Guide A, section 5.6.3. Guide A, section 2.3.1, suggests that for most buildings a 24-hour mean temperature is appropriate. However, a 48-hour mean temperature is more suitable for buildings with high thermal inertia (i.e. high thermal mass, low heat loss), with a response factor □ 6.

3.3.3 Relationship between dry resultant, environmental and air temperatures

As noted above, thermal comfort is best assessed in terms of dry resultant temperature, which depends on the combined effect of air and radiant temperature. However, steady-state heat loss calculations should be made using environmental temperature, which is the hypothetical temperature that determines the rate of heat flow into a room by both convection and radiation. For tightly built and well insulated buildings, differences between internal air temperature (t_{ai}), mean radiant temperature (t_r), dry resultant temperature (t_c) and environmental temperature (t_e) are usually small in relation to the other approximations involved in plant sizing and may be neglected under steady-state conditions. This will apply to buildings built to current Building Regulations with minimum winter ventilation. However, where U-values are higher, e.g. in old buildings, or where there is a high ventilation rate either by design or due to leaky construction, there may be significant differences.

An estimate of the air temperature required to achieve a particular dry resultant temperature can be made using equation 5.11 in CIBSE Guide A. The difference between air and dry resultant temperature is likely to be greater in a thermally massive building that is heated intermittently for short periods only, such as some church buildings. In such cases, radiant heating can quickly achieve comfortable conditions without having to raise the temperature of the structure. Radiant heating can also be effective in buildings that require high ventilation rates, especially when they have high ceilings, a situation that typically occurs in industrial buildings. In this case, comfort conditions can be achieved in working areas without having to heat large volumes of air at higher levels, typically by exploiting heat absorbed by the floor and re-radiated at low level.

3.3.4 Structural or fabric heat loss

Structural heat loss occurs by conduction of heat through those parts of the structure exposed to the outside air or adjacent to unheated areas, often referred to as the 'building envelope'. The heat loss through each external element of the building can be calculated from:

$$\phi_f = UA\ (t_{en} - t_{ao}) \tag{3.3}$$

where ϕ_f is the heat loss through an external element of the building (W), U is the thermal transmittance of the building element (W.m^{-2}. K^{-1}), A is the area of the of building element (m$_2$), t_{en} is the indoor environmental temperature (°C) and t_{ao} is the outdoor temperature (°C).

Thermal bridges occur where cavities or insulation are crossed by components or materials with high thermal conductivity. They frequently occur around windows, doors and other wall openings through lintels, jambs and sills and can be particularly significant when a structural feature, such as a floor extending to a balcony, penetrates a wall. This type of thermal bridge may conveniently be treated as a linear feature, characterised by a heat loss per unit length.

Thermal bridging may also occur where layers in a construction are bridged by elements required for its structural integrity. Examples include mortar joints in masonry construction and joists in timber frame buildings. Tabulated U-values may already take account of some such effects but, where U-values are being calculated from the properties of the layers in a construction, it is essential that such bridging is taken into account, especially for highly insulated structures. Several methods exist for calculating the effects of bridging including the 'combined method' specified by BS EN ISO 6946[7] and required by Building Regulations Approved Documents L1 and L2[8] .

Table 3.3 *U*-values for solid ground floors on clay soil

Ratio p_f/A_f	*U*-value (W.m^{2-} K^{-1}) for stated thermal resistance of floor construction R_f (m^{2} K.W^{-1})					
	0	0.5	1.0	1.5	2.0	2.5
0.05	0.13	0.11	0.10	0.09	0.08	0.08
0.10	0.22	0.18	0.16	0.14	0.13	0.12
0.15	0.30	0.24	0.21	0.18	0.17	0.15
0.20	0.37	0.29	0.25	0.22	0.19	0.18
0.25	0.44	0.34	0.28	0.24	0.22	0.19
0.30	0.49	0.38	0.31	0.27	0.23	0.21
0.35	0.55	0.41	0.34	0.29	0.25	0.22
0.40	0.60	0.44	0.36	0.30	0.26	0.23
0.45	0.65	0.47	0.38	0.32	0.27	0.23
0.50	0.70	0.50	0.40	0.33	0.28	0.24
0.55	0.74	0.52	0.41	0.34	0.28	0.25
0.60	0.78	0.55	0.43	0.35	0.29	0.25
0.65	0.82	0.57	0.44	0.35	0.30	0.26
0.70	0.86	0.59	0.45	0.36	0.30	0.26
0.75	0.89	0.61	0.46	0.37	0.31	0.27
0.80	0.93	0.62	0.47	0.37	0.32	0.27
0.85	0.96	0.64	0.47	0.38	0.32	0.28
0.90	0.99	0.65	0.48	0.39	0.32	0.28
0.95	1.02	0.66	0.49	0.39	0.33	0.28
1.00	1.05	0.68	0.50	0.40	0.33	0.28

Section 3 of CIBSE Guide A gives detailed information on thermal bridging and includes worked examples of the calculation required for both the methods referred to above. Other thermal bridging effects may be taken into account using the methods given in BS EN ISO 10211[9,10] .

Heat losses through ground floors need to be treated differently from other losses as they are affected by the mass of earth beneath the floor and in thermal contact with it. A full analysis requires 3-dimensional treatment and allowance for thermal storage effects but methods have been developed for producing an effective *U*-value for the whole floor. The standard for the calculation of *U*-va1ues for ground floors and basements is BS EN ISO 13370[11] . The recommended method is described in detail in CIBSE Guide A section 3; the following is a brief description of the method for solid ground floors in contact with the earth.

Table 3.3 gives *U*-values for solid ground floors on clay (thermal conductivity = 1.5 W.m^{-1}. K^{-1}), for a range of values of the ratio of the exposed floor perimeter p_f (m) and floor area A_f (m^2). The *U*-values are given as a function of the thermal resistance of the floor construction, R_f , where R_f = 0 for an uninsulated floor. CIBSE Guide A section 3 includes tables for soils having different conductivity and gives equations for calculating the U-values for other types of ground floors. Losses are predominantly from areas close to the perimeter and hence large floors have low average U-values. Therefore large floors may not require to be insulated to satisfy the Building Regulations. However, the mean value should not be applied uniformly to each ground floor zone and the heat losses should be calculated separately for individual perimeter rooms.

U-values for windows are normally quoted for the entire opening and therefore must include heat lost through both the frame and the glazing. Indicative *U*-values for typical glazing/frame combinations are given in Building

Regulations Approved Documents L1 and L2[8]. For advanced glazing, incorporating low emissivity coatings and inert gas fillings, the performance of the frame can be significantly worse than that of the glazing. In such cases, *U*-values should be calculated individually using the methods given in BS EN ISO 10077[12] or reference made to manufacturers' certified *U*-values.

The rate of fabric heat loss for the whole building may be calculated by summing the losses calculated for each element. The area of each element may be based on either internal or external measurement; however, if internal measurements are used, they should be adjusted to take account of intermediate floors and party walls. Measurements used in calculations to show compliance with the Building Regulations should be based on overall internal dimensions for the whole building, including the thickness of party walls and floors.

U-values for typical constructions are given in Guide A, Appendix 3.A8. For other constructions the *U*-value must be calculated by summing the thermal resistances for the various elements. For each layer in a uniform plane, the thermal resistance is given by:

$$R_i = d/\lambda \qquad (3.4)$$

where R_i is the thermal resistance of the element ($m^2.K.W^{-1}$), d is the thickness of the element (m) and λ is the thermal conductivity ($W.m^{-1}.K^{-1}$).

Values of thermal conductivity of the materials used in the various building elements can be obtained from manufacturers or from CIBSE Guide A, Appendix 3.A7. The thermal resistances of air gaps and surfaces should also be taken into account using the values given in CIBSE Guide A, Table 3.53.

The total thermal resistance of the element is calculated by adding up the thermal resistances of its layers:

$$R = R_{si} + R_1 + R_2 \,..... + R_a + R_{se} \qquad (3.5)$$

where R_{si} is the internal surface resistance ($m^2.K.W^{-1}$), R_1, R_2 etc. are the thermal resistances of layers 1, 2 etc. ($m^2.K.W^{-1}$), R_a is the thermal resistance of the airspace ($m^2.K.W^{-1}$) and R_{se} is the external surface resistance ($m^2.K.W^{-1}$).

The U-value is the reciprocal of the thermal resistance:

$$U = 1/R \qquad (3.6)$$

Where adjacent rooms are to be maintained at the same temperature, there are neither heat losses nor heat gains either via the internal fabric or by internal air movement. However, where the design internal temperatures are not identical, heat losses between rooms should be taken into account in determining the heat requirements of each room.

3.3.5 Ventilation heat loss

Ventilation heat loss depends upon the rate at which air enters and leaves the building, the heat capacity of the air and the temperature difference between indoors and outdoors. The heat capacity of air is approximately constant under the conditions encountered in a building. The volume of air passing through the building depends upon the volume of the building and the air change rate, which is usually expressed in air changes per hour (h^{-1}) The ventilation heat loss rate of a room or building may be calculated by the formular:

$$\phi_v = q_m (h_{ai} - h_{ao}) \qquad (3.7)$$

where ϕ_v is the heat loss due to ventilation (W), q_m is the mass flow rate of ventilation air ($kg.s^{-1}$), h_{ai} is the enthalpy of the indoor air ($J.kg^{-1}$) and h_{ao} is the enthalpy of the outdoor air ($J.kg^{-1}$).

Where the moisture content of the air remains constant, only sensible heat needs to be considered so the ventilation heat loss can be given by:

$$\phi_v = q_m c_p (t_{ai} - t_{ao}) \qquad (3.8)$$

where c_p is the specific heat capacity of air at constant pressure ($J.kg^{-1}.K^{-1}$), t_{ai} is the inside air temperature (ºC) and t_{ao} is the outside air temperature (°C).

By convention, the conditions for the air are taken as the internal conditions, for which the density will not differ greatly from p = 1.20 $kg.m^{-3}$, and the specific heat capacity c_p = 1.00 $kJ.kg^{-1}.K^{-1}$. This leads to the following simplifications:

$$\phi_v = 1.2 q_v (t_{ai} - t_{ao}) \qquad (3.9)$$

or:

$$\phi_v = (NV/3)(t_{ai} - t_{ao}) \qquad (3.10)$$

where ϕ_v is the heat loss due to ventilation (W), q_v is the volume flow rate of air ($litre.s^{-1}$), t_{ai} is the inside air temperature (°C), t_{ao} the outside air temperature (°C), N is the number of air changes per hour (h^{-1}) and V is the volume of the room (m^3).

Ventilation heat losses may be divided into two distinct elements:

- purpose provided ventilation, either by mechanical or natural means
- air infiltration.

The amount of purpose provided ventilation is decided according to how the building is to be used and occupied. In most buildings, ventilation is provided at a rate aimed at ensuring adequate air quality for building occupants but in some industrial buildings it must be based on matching process extract requirements. Mechanical ventilation is controlled, the design amount known, and the heat loss easily calculated. Ventilation requirements may be specified either in volume supply ($litre.s^{-1}$) or in air changes per hour (h^{-1}). Recommended supply rates for a range of buildings and building uses are given in CIBSE Guide A section 1(1),extracts from which are given in Table 3.4. More detailed guidance on ventilation is given in CIBSE Guide B2: *Ventilation and air conditioning*[2].

When heat recovery is installed, the net ventilation load becomes:

$$\phi_v = 1.2 q_v (t_{a2} - t_{ao}) \qquad (3.11)$$

or:

$$\phi_v = q_m (h_{a2} - h_{ao}) \qquad (3.12)$$

Table 3.4 Recommended fresh air supply rates for selected buildings and uses[1]

Building/use	Air supply rate
Public and commercial buildings (general use)	8 litres^{-1}· person^{-1}
Hotel bathrooms	12 litre.s^{-1}· person^{-1}
Hospital operating theatres	650 to 1000 m^3·s^{-1}
Toilets	>5 air changes per hour
Changing rooms	10 air changes per hour
Squash courts	4 air changes per hour
Ice rinks	3 air changes per hour
Swimming pool halls	15 litre.s^{-1}· m^{-2} (of wet area)
Bedrooms and living rooms in dwellings	0.4 to 1 air changes per hour
Kitchens in dwellings	60 litre·s^{-1}
Bathrooms in dwellings	15 litre·s^{-1}

where t_{a2} is the extract air temperature after the heat recovery unit (°C) and h_{a2} is the extract air enthalpy after the heat recovery unit (J.kg^{-1}).

Air infiltration is the unintentional leakage of air through a building due to imperfections in its fabric. The air leakage of the building can be measured using a fan pressurisation test, which provides a basis for estimating average infiltration rates. However, infiltration is uncontrolled and varies both with wind speed and the difference between indoor and outdoor temperature, the latter being particularly important in tall buildings. It is highly variable and difficult to predict and can therefore only be an estimate for which a suitable allowance is made in design. Methods for estimating infiltration rates are given in CIBSE Guide A[1] , section 4. Table 3.5 gives empirical infiltration allowances for use in heat load calculations for existing buildings where pressurisation test results are not available. As air infiltration is related to surface area rather than volume, estimates based on air change rate tend to exaggerate infiltration losses for large buildings, which points to the need for measurement in thosecases.

The air infiltration allowances given in Table 3.5 are applicable to single rooms or spaces and are appropriate for the estimation of room heat loads. The load on the central plant will be somewhat less (up to 50%) than the total of the individual room loads due to infiltration diversity.

Table 3.5 Recommend allowances for air infiltration for selected building types[1]

Building/room type	Air infiltration allowance (ACH)	Building/room type	Air infiltration allowance (ACH)
Art galleries and museums	1	Hospitals (continued):	
Assembly and lecture halls	0.5	- wards and patient areas	2
		- waiting rooms	1
Banking halls	1 to 1.5	Hotels:	
Bars	1	- bedrooms	1
Canteens and dining rooms	1	- public rooms	1
		- corridors	1.5
Churches and chapels	0.5 to 1	- foyers	1.5
Dining and banqueting halls	0.5	Laboratories	1
Exhibition halls	0.5	Law courts	1
Factories:		Libraries:	
- up to 300 m^3 volume	1.5 to 2.5	- reading rooms	0.5 to 0.7
- 300 m^3 to 3000 m^3	0.75 to 1.5	- stack rooms	0.5
- 3000 m^3 to 10,000 m^3	0.5 to 1.0	- storerooms	0.25
- over 10,000 m3	0.25 to 0.75	Offices:	
Fire stations	0.5 to 1	- private	1
		- general	1
Gymnasia	0.75	- storerooms	0.5
Houses, flats and hostels:		Police cells	5
- living rooms	1		
- bedrooms	0.5	Restaurants, cafes	1
- bed-sitting rooms	1	Schools, colleges:	
- bathrooms	2	- classrooms	2
- lavatories, cloakrooms	1.5	- lecture rooms	1
- service rooms	0.5	- studios	1
- staircases, corridors	1.5		
- entrance halls, foyers	1.5	Sports pavilion changing rooms	1
- public rooms	1	Swimming pools:	
Hospitals:		- changing rooms	0.5
- corridors	1	- pool hall	0.5
- offices	1	Warehouses:	
- operating theatres	0.5	- working and packing areas	0.5
- storerooms	0.5	- storage areas	0.2

Building Regulations Approved Document L2[8] recommends that air permeability measured in accordance with CIBSE TM23: *Testing buildings for air leakage*[13] should not be greater than 10 $m^3 \cdot h^{-1}$ per m^2 of external surface area at a pressure of 50 Pa. It also states that pressurisation tests should be used to show compliance with the Regulations for buildings with a floor area of 1000 m^2 or more. For buildings of less than 1000 m2, pressurisation testing may also be used, but a report by a competent person giving evidence of compliance based on design and construction details may be accepted as an alternative.

CIBSE TM23: *Testing buildings for air leakage*[13] describes the two different parameters currently used to quantify air leakage in buildings, i.e. air leakage index and air permeability. Both are measured using the same pressurisation technique, as described in TM23, and both are usually expressed in terms of volume flow per hour ($m^3 \cdot h^{-1}$) of air supplied per m^2 of building envelope area. They differ in the definition of building envelope area to which they refer; the solid ground floor is excluded from the definition of envelope used for the air leakage index, but is included for air permeability. Air permeability is used in the Building Regulations and the European Standard BS EN 13829[14]. However, the air leakage index was used for most of the measurements used to produce the current database of results.

TM23 provides a simple method of estimation of air infiltration rate from the air permeability. This should be used with caution for calculation of heat losses since it currently applies only to houses and offices and does not include additional infiltration losses related to the use of the building.

3.3.6 Calculation of design heat loss for rooms and buildings

The design heat loss for each zone or room is calculated by summing the fabric heat loss for each element and the ventilation heat loss, including an allowance for infiltration. The calculations are carried out under external conditions chosen as described in section 3.3.2

$$\phi = \Sigma(\phi_f) + \phi_v \qquad (3.13)$$

where ϕ is the total design heat loss (W), ϕ_f is the fabric heat loss (W) and ϕ_v is the ventilation heat loss (W).

Section 4.7 describes how the calculated heat loss may be used in sizing system components, including both heat emitters and boilers.

The recommended allowance for infiltration is important and may constitute a significant component of the total design heat loss. While this allowance should be used in full for sizing heat emitters, a diversity factor should be applied to it when sizing central plant. CIBSE Guide A[1], section 5.8.3.5, notes that infiltration of outdoor air only takes place on the windward side of a building at any one time, the flow on the leeward side being outwards. This suggests that a diversity factor of 0.5 should be applied to the infiltration heat loss in calculating total system load. The same section of Guide A gives overall diversity factors ranging from 0.7 to 1.0 for the total load in continuously heated buildings.

3.3.7 Thermal capacity

Thermal capacity (or thermal mass) denotes the capacity of building elements to store heat, which is an important determinant of its transient or dynamic temperature response. High thermal capacity is favoured when it is desirable to slow down the rate at which a building changes temperature, such as in reducing peak summertime temperatures caused by solar gains, thereby reducing peak cooling loads.

High thermal capacity reduces both the drop in temperature during periods when the building is not occupied and the rate at which it re-heats. When buildings are not occupied at weekends, then the effect of heating up from cold on a Monday morning needs to be considered; in this case a greater thermal capacity will require either a higher plant ratio or a longer pre-heat period. Full treatment of the effects of thermal capacity requires the use of dynamic modelling, as described in CIBSE A[1] section 5.6, or the use of a computer-based dynamic energy simulation. Simplified analysis can be undertaken using the concept of thermal admittance (Y-value), which is a measure of the rate of flow between the internal surfaces of a structure and the environmental temperature inthespaceitencloses, see section 4.7.

3.4 'Buildability', 'commissionability' and 'maintainability'

All design must take account of the environment in which the system will be installed, commissioned and operated, considering both safety and economy.

The Construction (Design and Management) Regulations 1994[15] (CDM Regulations) place an obligation on designers to ensure that systems they design and specify can be safely installed and maintained. The Regulations require that a designer must be competent and have the necessary skills and resources, including technical facilities. The designer of an installation or a piece of equipment that requires maintenance has a duty to carry out a risk assessment of the maintenance function. Where this assessment shows a hazard to the maintenance operative, the designer must reconsider the proposals and try to remove or mitigate the risk.

Apart from matters affecting safety, designers must take account of maintenance cost over the lifetime of the systems they specify. In particular, it is important to ensure that the client understands the maintenance requirements, including cost and the need for skills or capabilities. The CIBSE's *Guide to ownership, operation and maintenance of building services*[16] contains guidance on maintenance issues that need to be addressed by the building services designer.

Part L of the Building Regulations[8] requires the provision of a 'commissioning plan that shows that every system has been inspected and commissioned in an appropriate sequence'. This implies that the designer must consider which measurements are required for commissioning and provide the information required for making and using those measurements. Also, the system must be designed so that the necessary measurements and tests can be carried out, taking account of access to the equipment and the health and safety those making the measurements. Approved Document L2 states that one way of demonstrating compliance would be to follow the guidance given in CIBSE Commissioning Codes[17-21], in BSRIA Commissioning Guides[22-27] and by the Commissioning Specialists Association[28]. The guidance on balancing given in section 4.3.2 is also relevant to this requirement.

3.5 Energy efficiency targets

New buildings and buildings undergoing major refurbishment must comply with the requirements of Part L1 (dwellings) or Part L2 (buildings other then dwellings) of the Building Regulations[8] (or the equivalent regulations that apply in Scotland[29] and Northern Ireland[30]). These requirements may be expressed either in U-values or as energy targets, typically calculated in terms of energy use per year according to a closely specified procedure. For example, the Standard Assessment Procedure for the Energy Rating of Dwellings[31] (SAP) describes how such a calculation may be done for dwellings in order to comply with Part L. SAP is also used in other contexts, for example to assess or specify the performance of stocks of houses owned by local authorities and housing associations. The Building Regulations in the Republic of Ireland offer a heat energy rating as a way of showing compliance with energy requirements for dwellings. It should be remembered that the Building Regulations set minimum levels for energy efficiency and it may economic to improve upon those levels in individual cases.

Energy targets for non-domestic buildings include those described in CIBSE Building Energy Codes **1** and 2. Energy benchmarks have also been developed for certain types of buildings; for example, Energy Consumption Guide 19[32] (ECON 19) gives typical performance levels achieved in office buildings. A method for estimating consumption and comparing performance with the ECON 19 benchmarks is described in CIBSE TM22: *Energy assessment and reporting methodology*[33]. Building Regulations Approved Document L[8] includes a carbon performance rating (CPR) as one way of showing compliance with the Regulations for office buildings. The BRE Environmental Assessment Method[34] (BREEAM) includes a broad range of environmental impacts but energy use contributes significantly to its overall assessment.

See CIBSE Guide F: *Energy efficiency in buildings* for detailed guidance on energy efficiency.

3.6 Life cycle issues

The designer's decisions will have consequences that persist throughout the life of the equipment installed, including durability, availability of consumable items and spare parts, and maintenance requirements. Consideration should also be given to how the heating system could be adapted to changes of use of the building. The combined impact may be best assessed using the concept of life cycle costs, which are the combined capital and revenue costs of an item of plant or equipment throughout a defined lifetime.

The capital costs of a system include initial costs, replacement costs and residual or scrap value at the end of the useful life of the system. Future costs are typically discounted to their present value. Revenue costs include energy costs, maintenance costs and costs arising as a consequence of system failure.

Life cycle costing is covered by BS ISO 156861-1[35] and guidance is given by HM Treasury[36], the Construction Client's Forum[37], BRE[38] and the Royal Institution of Chartered Surveyors[39]. See also CIBSE's *Guide to ownership, operation and maintenance ofbuilding services*[16].

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4 System selection

4.1 Choice of heating options

This section deals with the attributes of particular systems and sub-systems, and the factors that need to be taken into consideration in their specification and design.

4.1.1 Heat emitters

The general characteristics of heat emitters need to be considered, with particular emphasis on the balance between convective and radiative output appropriate to the requirements of the building and activities to be carried out within it. As noted in section 3, well insulated buildings tend to have only small differences between air and mean radiant temperatures when they are in a steady-state. Nevertheless there can be situations in which it is better to provide as much output as possible in either convective or radiant form. For example, radiant heating may be desirable in heavyweight buildings that are occupied intermittently, such as churches, or in buildings with high ceilings, where the heat can be better directed to fall directly on occupants without having to warm the fabric of the building. The characteristics of particular heat emitters are discussed in the following sections.

4.1.2 Location of heat emitters

As it is generally desirable to provide uniform temperatures throughout a room or zone, careful consideration should be given to the location of heat emitters. Their position can contribute to the problem of radiant asymmetry described in section 3.2, and can significantly affect the comfort of particular areas within a room. For example, it may be beneficial to locate emitters to counteract the radiative effects or down-draughts caused by cool surfaces. When single glazing is encountered, it is particularly important to locate radiators beneath windows, but it can still be desirable to do so with double glazing. It is best to locate heat sources on external walls if the walls are poorly insulated.

4.1.3 Distribution medium

The medium for distributing heat around the building needs also to be considered, taking account of requirements for heat emitters. Air and water are the commonest choices but steam is still used in many existing buildings and refrigerant fluids are used in heat pumps. Electricity is the most versatile medium for distribution as it can be converted to heat at any temperature required at any location. However, consideration of primary energy, CO_2 emissions and running cost tend to militate against the use of electricity. Gas and oil may also be distributed directly to individual heaters.

The choice of distribution medium must take account of the balance between radiant and convective output required. When air is used for distribution, the opportunity for radiant heat output is very limited but water and steam systems can be designed to give output that is either predominantly convective or with a significant radiative component. However, when highly directed radiant output is required then only infrared elements powered by electricity or directly fired by gas are applicable. The relative merits of various distribution media are described briefly in Table 4.1.

Table 4.1 Characteristics of heat distribution media

Medium	Principal characteristics
Air	The main advantage of air is that no intermediate medium or heat exchanger is needed. The main disadvantage is the large volume of air required and the size of ductwork that results. This is due to the low density of air and the small temperature difference permissible between supply and return. High energy consumption required by fans can also be a disadvantage.
Low pressure hot water (LPHW)	LPHW systems operate at low pressures that can be generated by an open or sealed expansion vessel. They are generally recognised as simple to install and safe in operation but output is limited by system temperatures restricted to a maximum of about 85 °C.
Medium pressure hot water (MPHW)	Permits system temperatures up to 120 °C and a greater drop in water temperature around the system and thus smaller pipework. Only on a large system is this likely to be of advantage. This category includes pressurisation up to 5 bar absolute.
High pressure hot water (HPHW)	Even higher temperatures are possible in high pressure systems (up to 10 bar absolute), resulting in even greater temperature drops in the system, and thus even smaller pipework. Due to the inherent dangers, all pipework must be welded and to the standards applicable to steam pipework. This in unlikely to be a cost-effective choice except for the transportation of heat over long distances.
Steam	Exploits the latent heat of condensation to provide very high transfer capacity. Operates at high pressures, requiring high maintenance and water treatment. Principally used in hospitals and buildings with large kitchens or processes requiring steam.

4.2 Energy efficiency

See section 2 above. The practical realisation of energy efficiency depends not only on the characteristics of the equipment installed but also on how it is controlled and integrated with other equipment. The following sections describe aspects of energy efficiency that need to be taken into account in heating system design.

4.2.1 Thermal insulation

For new buildings, satisfying the Building Regulations will ensure that the external fabric has a reasonable and cost-effective degree of insulation (but not necessarily the economic optimum), and that insulation is applied to hot water storage vessels and heating pipes that pass outside heated spaces.

In existing buildings, consideration should be given to improving the thermal resistance of the fabric, which can reduce the heat loss significantly. This can offer a number of advantages, including reduced load on the heating system, improved comfort and the elimination of condensation on the inner surfaces of external walls and ceilings. In general, decisions on whether or not to improve insulation should be made following an appraisal of the costs and benefits, taking account both of running costs and the impact on capital costs of the heating system.

Where a new heating system is to be installed in an existing building, pipe and storage vessel insulation should meet the standards required by Parts L1/L2 of the Building Regulations[1] . This should apply when parts of an existing system are to be retained, constrained only by limited access to sections of existing pipework.

4.2.2 Reducing air infiltration

See section 3.3.5 above. Infiltration can contribute substantially to the heating load of the building and cause discomfort through the presence of draughts and cold areas. As for fabric insulation, the costs and benefits of measures to reduce infiltration should be appraised on a life-cycle basis, taking account of both running costs and capital costs.

4.2.3 Seasonal boiler efficiency

Boiler efficiency is the principal determinant of system efficiency in many heating systems. What matters is the average efficiency of the boiler under varying conditions throughout the year, known as 'seasonal efficiency'. This may differ significantly from the bench test boiler efficiency, although the latter may be a useful basis for comparison between boilers. Typical seasonal efficiencies for various types of boiler are given in Table 4.2. For domestic boilers, seasonal efficiencies may be obtained from the SEDBUK[2] database.

Many boilers have a lower efficiency when operating at part load, particularly in an on/off control mode, see Figure 4.1. Apart from the pre-heat period, a boiler spends most of its operating life at part load. This has led to the increased popularity of multiple boiler systems since, at 25% of design load, it is better to have 25% of a number of

Table 4.2 Typical seasonal efficiencies for various boiler types[3]

Boiler/system	Seasonal efficiency(%)
Condensing boilers:	
- under-floor or warm water system	90
- standard size radiators, variable temperature circuit (weather compensation)	87
- standard fixed temperature emitters (83/72 °C flow/return)*	85
Non-condensing boilers:	
- modern high-efficiency non-condensing boilers	80-82
- good modern boiler design closely matched to demand	75
- typical good existing boiler	70
- typical existing oversized boiler (atmospheric, cast-iron sectional)	45-65

* Not permitted by current Building Regulations

small boilers operating at full output, rather than one large boiler operating at 25% output.

Condensing boilers operate at peak efficiency when return water temperatures are low, which increases the extent to which condensation takes place. This can occur either at part or full load and depends principally on the characteristics of the system in which it is installed. Condensing boilers are particularly well suited to LPHW systems operating at low flow and return temperatures, such as under-floor heating. They may also be operated as lead boilers in multiple boiler systems.

4.2.4 Efficiency of ancillary devices

Heating systems rely on a range of electrically powered equipment to make them function, including pumps, fans, dampers, electrically actuated valves, sensors and controllers. Of these, pumps and fans are likely to consume by

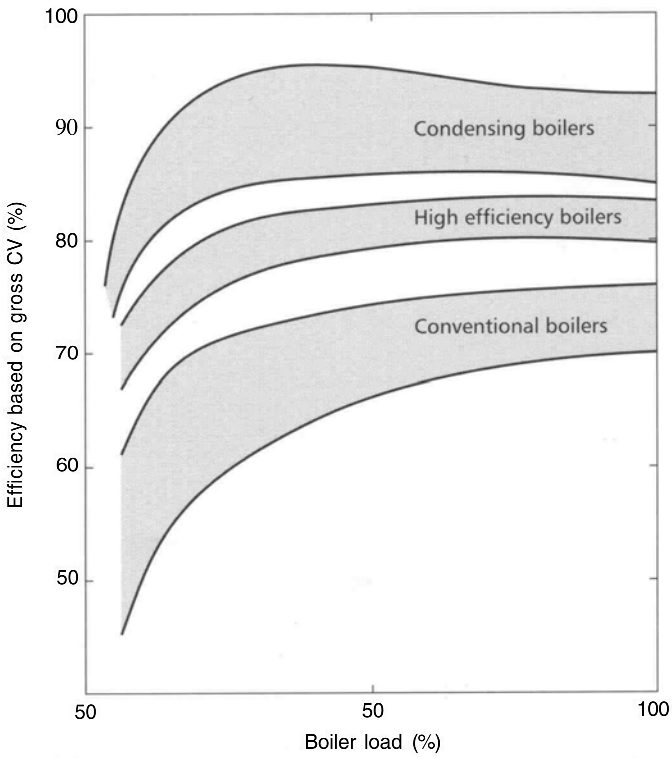

Figure 4.1 Typical seasonal LTHW boiler efficiencies at part load[4]

far the most energy, but even low electrical consumption may be significant if it is by equipment that is on continuously. Also, it is important to remember that the cost per kWh of electricity is typically four times that of fuels used for heating, so it is important to avoid unnecessary electrical consumption.

For pumps and fans, what matters is the overall efficiency of the combined unit including the motor and the drive coupling. Fan and pump characteristics obtained from manufacturers should be used to design the system to operate around the point of maximum efficiency, taking account of both the efficiency of the motors and of the coupling to the pump or fan. Also, it is important that the drive ratios are selected to give a good match between the motor and the load characteristic of the equipment it is driving.

Pumping and fan energy consumption costs can be considerable and may be a significant proportion of total running costs in some heating systems. However, it may be possible to reduce running costs by specifying larger pipes or ductwork. Control system design can also have a significant impact on running costs. Pumps and fans should not be left running longer than necessary and multiple speed or variable speed drives should be considered where a wide flow range is required.

4.2.5 Controls

Heating system controls perform two distinct functions:

- they maintain the temperature conditions required within the building when it is occupied, including pre-heating to ensure that those conditions are met at the start of occupancy periods
- they ensure that the system itself operates safely and efficiently under all conditions.

The accuracy with which the specified temperatures are maintained and the length of the heating period both have a significant impact on energy efficiency and running costs. A poorly controlled system will lead to complaints when temperatures are low. The response may be raised set-points or extended pre-heat periods, both of which have the effect of increasing average temperatures and energy consumption. Controls which schedule system operation, such as boiler sequencing, can be equally important in their effect on energy efficiency, especially as the system may appear to function satisfactorily while operating at low efficiency.

4.2.6 Zoning

Rooms or areas within buildings may require to be heated to different temperatures or at different times, each requiring independent control. Where several rooms or areas of a building behave in a similar manner, they can be grouped together as a 'zone' and put on the same circuit and controller. For instance, all similar south-facing rooms of a building may experience identical solar gain changes and some parts of the building may have the same occupancy patterns. The thermal responses of different parts of a building need to be considered before assigning them to zones, so that all parts of the zone reach their design internal temperature together. A poor choice of zones can lead to some rooms being too hot and others too cool.

4.2.7 Ventilation heat recovery

A mechanical ventilation system increases overall power requirements but offers potential energy savings through better control of ventilation, and the possibility of heat recovery. The most obvious saving is through limiting the operation of the system to times when it is required, which is usually only when the building is occupied. The extent to which savings are possible depends crucially on the air leakage performance of the building. In a leaky building, heat losses through infiltration may be comparable with those arising from ventilation. In an airtight building, the heat losses during the pre-heat period may be considerably reduced by leaving the ventilation off and adopting a smaller plant size ratio.

Ventilation heat recovery extracts heat from exhaust air for reuse within a building. It includes:

- 'air-to-air' heat recovery, in which heat is extracted from the exhaust air and transferred to the supply air using a heat exchanger or thermal wheel
- a heat pump, to extract heat from the exhaust air and transfer it to domestic hot water.

Air-to-air heat recovery is only possible where both supply air and exhaust air are ducted. High heat transfer efficiencies (up to 90%) can be achieved. Plate heat exchangers are favoured for use in houses and small commercial systems, while thermal wheels are typically used in large commercial buildings. Heat pipe systems offer very high heat efficiency and low running cost. Run-around coils may also be used and have the advantage that supply and exhaust air streams need not be adjacent to each other.

The benefits of the energy saved by heat recovery must take account of any additional electricity costs associated with the heat recovery system, including the effect of the additional pressure drop across the heat exchanger. Assessment of the benefits of heat recovery should also take account of the effect of infiltration, which may by-pass the ventilation system to a large extent. The cost-effectiveness of heat recovery also depends on climate and is greatest when winters are severe.

Heat pumps transferring heat from exhaust ventilation air to heat domestic hot water have widely been used in apartment buildings in Scandinavia. The same principle has been successfully used in swimming pools.

4.3 Hydronic systems

Hydronic systems use hot water for transferring heat from the heat generator to the heat emitters. The most usual type of heat generator for hydronic systems is a 'boiler', misleadingly named as it must be designed to avoid boiling during operation. Hot water may also be generated by heat pumps, waste heat reclaimed from processes and by solar panels, the latter typically being used to produce domestic hot water in summer. Heat emitters take a variety of forms including panel radiators, natural and forced convectors, fan-coil units, and under-floor heating. Hydronic systems normally rely on pumps for circulation, although gravity circulation was favoured for systems designed before around 1950.

Hydronic systems offer considerable flexibility in type and location of emitters. The heat output available in radiant form is limited by the temperature of the circulation water but, for radiators and heated panels, can be sufficient to counteract the effect of cold radiation from badly insulated external surfaces. Convective output can be provided by enclosed units relying on either natural or forced air-convection. Flexibility of location is ensured by the small diameter of the circulation pipework and the wide variety of emitter sizes and types.

In addition to the sizing of emitters and boilers, the design of hydronic systems involves the hydraulic design of the circulation system to ensure that water reaches each emitter at the necessary flow rate and that the pressures around the system are maintained at appropriate levels. System static pressures may be controlled either by sealed expansion vessels or by hydrostatic pressure arising from the positioning of cisterns at atmospheric pressure above the highest point of the circulating system. Both cisterns and pressure vessels must cope with the water expansion that occurs as the system heats up from cold; the design of feed, expansion and venting is crucial to both the safety and correct operation of systems.

4.3.1 Operating temperatures for hydronic systems

The operating temperature of a hydronic heating system both determines its potential performance and affects its design. Systems are generally classified according to the temperature and static pressure at which they operate, see Table 4.3. Low pressure hot water (LPHW) systems may be either sealed or open to the atmosphere and use a variety of materials for the distribution pipework. Also, the operating temperature should be set low enough that exposed heat emitters, such as panel radiators, do not present a burn hazard to building occupants. Medium and high pressure systems are favoured where a high heat output is required, such as in a fan coil system in a large building. High pressure systems are particularly favoured for distribution mains, from which secondary systems extract heat by heat exchangers for local circulation at lower temperatures.

LPHW systems are typically designed to operate with a maximum flow temperature of 82 °C and system temperature drop of 10 °C. A minimum return temperature of 66 °C is specified by BS 5449[5] unless boilers are designed to cope with condensation or are of the electric storage type. For condensing boilers, a low return temperature may be used with the benefit of improved operating efficiency. It may also be noted that the larger the difference between flow and return temperatures ($t_1 - t_2$), the smaller the mass flow required, which tends to reduce pipe sizes and pumping power. The heat flux is given by:

$$\phi = q_m \, c_p \, (t_1 - t_2) \quad (4.1)$$

where ϕ is the heat flux (W), q_m is the mass flow rate (kg.s^{-1}), c_p is the specific heat capacity of the heat transfer fluid (J.kg^{-1}.K^{-1}), t_1 is the flow temperature (°C) and t_2 is the return temperature (°C).

Hence, the mass flow rate is given by:

$$q_m = \phi \,/\, [c_p \, (t_1 - t_2) \quad (4.2)$$

The efficiency of a condensing boiler is more strongly influenced by the return temperature, rather than the flow temperature, which ought to be a further encouragement to use large values of ($t_1 - t_2$). However, a larger temperature difference lowers the mean water temperature of the emitter, which reduces specific output and requires larger surface area. The effect of flow rate and return temperature on heat output is explored more fully in section 5.1.1.

The relationship between emitter output and temperature is dealt with in section 5 and varies according to the type of emitter. In general, it may be noted that output tends to increase disproportionately as the difference between the mean system temperature and the room temperature increases. This favours the use of a high system temperature. However, other factors need to be considered which may favour a lower temperature, including the surface temperature of radiators, boiler operating efficiency and the characteristics of certain heat emitters. For example, under-floor heating is designed to operate with low system temperatures to keep floor surface temperatures below 29 °C.

Table 4.3 Design water temperatures and pressures for hydronic heating systems

Category	System design water temperature (°C)	Operating static pressure (bar absolute)
Low pressure hot water (LPHW)	40 to 85	1 to 3
Medium pressure hot water (MPHW)	100 to 120	3 to 5
High pressure hot water (HPHW)	> 120	5 to 10*

* Account must be taken of varying static pressure in a tall building

4.3.2 System layout and design

Systems must be designed to match their specified design heat load, including domestic hot water provision where required, and to have controls capable of matching output to the full range of variation in load over a heating season. Separate circuits may be required to serve zones of the building with different heat requirements. In addition, there must be provision for hydraulic balancing of circuits and sub-circuits, and for filling, draining and venting of each part of the system.

Distribution systems may be broadly grouped into one-pipe and two-pipe categories. In one-pipe systems, radiators are effectively fed in series, and system temperature varies around the circuit. They have not been extensively used in the UK during the last half-century but are common throughout the countries of the former Soviet Union, East Europe and China. Control of one-pipe systems requires the use of by-passes and 3-port valves. Two-pipe systems operate at nominally the same temperature throughout the circuit but require good balancing for that condition to be achieved in practice. Control of two-pipe systems may employ either 2-port or 3-port valves to restrict flow to individual heat emitters.

Draft European Standard prEN 12828 [6] deals with the design of hydronic heating systems with operating temperatures up to 105 °C and 1 MW design heat load. It covers heat supply, heat distribution, heat emitters, and control systems. BS 5449 [5] describes systems specifically for use in domestic premises, although it contains much that is applicable to small systems in other buildings. Detailed guidance on the design of domestic systems is given in the HVCA's *Domestic Heating Design Guide* [7].

4.3.2.1 Hydraulic design

Hydraulic design needs to take account of the effect of water velocity on noise and erosion, and of the pressure and flow characteristics of the circulation pump. CIBSE Guide C [8], section 4.4, contains tables showing pressure loss against flow rate for common tube sizes and materials. Flow velocities may be determined by consideration of pressure drops per metre of pipe run (typically in the range of 100 to 350 $Pa.m^{-1}$). Alternatively, flow velocities may be considered directly, usually to be maintained in the range 0.75 to 1.5 $m.s^{-1}$ for small-bore pipes (< 50mm diameter) and between 1.25 and 3 $m.s^{-1}$ for larger pipes.

Pumps should be capable of delivering the maximum flow required by the circuit at the design pressure drop around the circuit of greatest resistance, commonly known as the index circuit. If variable speed pumping is to be used, the method of controlling pump speed should be clearly described and the pump should be sized to operate around an appropriate part of its operating range.

The location and sizing of control valves need to take account of pressure drops and flows around the circuit to ensure that they operate with sufficient valve authority, see section 5.1.5.

4.3.2.2 Balancing

The objective of balancing is to ensure that each emitter receives the flow required at the design temperature. Balancing may be carried out most precisely by measuring and adjusting flow to individual parts of the circuit, but can also be carried out by observing temperatures throughout the system. Temperature-based balancing is commonly used on domestic systems but has the disadvantage that the adjustments must be made and checked when the system has reached a steady-state, which may take a considerable time.

It is important to take account of the need for balancing at the design stage, including the location of measuring stations around the system, the equipment needed to achieve balancing, and the procedures for carrying it out. Balancing by flow requires a provision for flow measurement and, in all cases, appropriate valves must be installed to control the flow to particular parts of the circuit. Balancing procedures, including a technical specification for commissioning the system, and the responsibilities of the various parties involved should be clearly identified at the outset. Flow measurement and regulating devices used for balancing are described in section 5.1.6.

The design of pipework systems can have a considerable effect on the ease with which balancing can be achieved. Reverse return circuits, which ensure that each load has a similar circuit length for its combined flow and return path, can eliminate much of the inequality of flow that might otherwise need to be rectified during balancing. Distribution manifolds and carefully selected pipe sizes can also assist with circuit balancing. It is important to avoid connecting loads with widely differing pressure drops and heat emitting characteristics (e.g. panel radiators and fan coil units) to the same sub-circuit.

Detailed guidance on commissioning may be found in CIBSE Commissioning Code W: *Water distribution systems*[9] and BSRIA Application Guide: *Commissioning of water systems in buildings*[10]. Guidance for systems with variable speed and multiple pumps may be found in the BSRIA Application Guide: *Commissioning of variable speed and multiple pumping systems*[11].

4.3.3 Choice of heat source

The choice of heat source will depend on the options available. These are outlined below.

4.3.3.1 Boilers

Boilers are available in a large range of types and sizes and, unless they are connected to a community heating system (see 4.3.3.4 below), almost all hydronic heating systems rely on one or more boilers. Boiler efficiency has improved markedly over the past two decades. Technical developments have included the use of new materials to reduce water content and exploit the condensing principle, gas-air modulation to improve combustion efficiency and modularisation to optimise system sizing. These developments have resulted in considerable improvements in performance at part load, with considerable benefit to seasonal efficiency.

Condensing boilers have efficiencies of up to 92% (gross calorific value) and are no longer much more expensive than other boilers. Neither are they so widely differentiated from non-condensing boilers in their performance, as the latter have improved considerably in their efficiency. Seasonal efficiency is the principal characteristic affecting the running cost of a boiler (or boiler system). In considering whole life cost, the lifetime of components should be taken into account.

'Combination' boilers provide an instantaneous supply of domestic hot water in addition to the usual boiler function. Their main advantage lies in the space they save, as they need no hot water storage cylinder or associated storage cistern. Also, they typically incorporate an expansion vessel for sealed operation, so that they need no plumbing in the loft space; this is particularly advantageous in flats where it may be difficult to obtain sufficient head from an open system. A further advantage is the elimination of heat losses from the hot water stored in the cylinder. Combination boilers have gained a large share of the market for boilers installed in housing over the past decade. However, the limitations of combination boilers should also be understood by both the installer and the client. The maximum flow rate at which hot water can be drawn is limited, especially over a prolonged period or when more than one point is being served simultaneously. Combination boilers are also susceptible to scaling by hard water, as the instantaneous water heating function requires the continual passage of water direct from the mains through a heat exchanger.

4.3.3.2 Heat pumps

Heat pumps have a number of different forms and exploit different sources of low grade heat. World wide, the heat pumps most widely used for heating are reversible air-to-air units that can also be used for cooling. Such units are typically found where there is significant need for cooling and the need for heating is limited. In the UK climate, electrically driven air-to-air heat pumps are not frequently installed solely to provide heating, which may be explained by the relatively high price of electricity in relation to gas. Heat pumps offer a particularly attractive option for heating when there **is** a suitably large source of low grade heat, such as a river, canal or an area of ground. Gas-fired ground source heat pumps currently being evaluated for use in housing as a boiler replacement are reported to have a seasonal coefficient of performance of around 1.4.

4.3.3.3 Solar panels

Solar water heating panels are widely used around the world to provide domestic hot water, particularly where sunshine is plentiful and fuel is relatively expensive, but are rarely used for space heating. In the UK climate, a domestic installation can typically provide hot water requirements for up to half the the annual hot water requirements, using either a separate pre-heat storage cylinder or a cylinder with two primary coils, one linked to the solar panel and the other to a boiler. Although technically successful, the economics of such systems are at best marginal in the UK when assessed against heat produced by a gas or oil boiler and they are rarely used in non-domestic buildings. Solar panels are also widely used for heating outdoor swimming pools in summer, for which they are more likely to be cost effective.

4.3.3.4 Community heating

If available, consideration should be given to utilising an existing supply of heat from a district or local heat supply ('community heating'). Heat supplied in this way may be of lower cost and may also have significantly lower environmental impact, especially if it is generated using combined heat and power (CHP) or makes use of heat from industrial processes or waste combustion. The low net co_2 emissions from heat from such sources can contribute significantly to achieving an environmental target for a building. Detailed guidance on the evaluation and implementation of community heating may be found in *Guide to community heating and CHP*[12], published under the government's Energy Efficiency Best Practice programme.

4.3.3.5 Stand-alone CHP systems

Where there is no suitable existing supply of heat, the opportunity for using a stand-alone combined heat and power (CHP) unit should be evaluated. The case for using CHP depends on requirements both for heat and electricity, their diurnal and seasonal variability and the extent to which they occur simultaneously. The optimum CHP plant capacity for a single building needs to be determined by an economic assessment of a range of plant sizes and in general will result in only part of the load being met by CHP, the rest being met by a boiler. It is important to have a reasonable match between the generated output and electricity demand, as the value of the electricity generated tends to dominate the economic analysis; the optimum ratio of heat demand to power demand generally lies between 1.3:1 and 2:1. There may be opportunities for exporting electricity. The best price for exported electricity is likely to be obtained from consumers who can link directly to the system rather than from a public electricity supplier. Where standby power generation is required to reduce dependency of public supplies of electricity, it may be particularly advantageous to install a CHP unit, thereby avoiding the additional capital cost of a separate standby generator. CIBSE Applications Manual AM12: *Small-scale combined heat and power for buildings*[13], gives detailed guidance on the application of CHP in buildings.

4.3.4 Choice of heat emitter

Hydronic systems are capable of working with a wide variety of heat emitters, offering a high degree of flexibility in location, appearance and output characteristics. This section deals with some of the principal characteristics of emitters affecting their suitability for particular situations.

4.3.4.1 Radiators

Radiators, usually of pressed steel panel construction, are the most frequent choice of emitter. They are available in a wide variety of shapes, sizes and output ranges, making it possible to obtain a unit (or units) to match the heat requirements of almost any room or zone.

Despite their name, radiators for hydronic systems usually produce more than half their output by convection, often aided by fins added to increase their surface area. Details on the heat output available from radiators are given in section 5.1.1.

4.3.4.2 Natural convectors

Wall-mounted natural convectors may be used instead of radiators. They may also be used where there is insufficient space for mounting radiators, for example in base-board or trench heating configurations. The output from natural convectors varies considerably with design and manufacturer's data for individual emitter types should be used. Details of how the heat output from natural convectors varies with system temperature are given in section 5.1.1.

4.3.4.3 Fan coil heaters

Fan coil units produce high heat outputs from compact units using forced air circulation. Their output may be considered to be entirely convective and is approximately proportional to temperature difference. Where systems contain a mixture of natural and forced air appliances, the different output characteristics of the two types should be taken into account, particularly with regard to zoning for control systems.

4.3.4.4 Floor heating

Floor heating (also referred to as under-floor heating) uses the floor surface itself as a heat emitter. Heat may be supplied either by embedded electric heating elements or by the circulation of water as part of a hydronic system, involving appropriately spaced pipes positioned beneath the floor surface. The pipes may be embedded within the screed of a solid floor or laid in a carefully controlled configuration beneath a suspended floor surface. Insulation beneath the heating elements is clearly very important for good control of output and to avoid unnecessary heat loss.

The heat emission characteristics of floor heating differ considerably from those of radiator heating. Floor surface temperature is critical to comfort, as well as to heat output. The optimum floor temperature range for comfort lies between 21 and 28 °C depending on surface material (see Table 5.9), so systems are normally designed to operate at no higher than 29 °C in occupied areas. Higher temperatures are acceptable in bathrooms and close to external walls with high heat loss, such as beneath full-length windows.

The design surface temperature is controlled by the spacing between pipes and the flow water temperature. It is also affected by floor construction, floor covering and the depth of the pipes beneath the floor surface; detailed design procedures are given by system manufacturers. In practice, systems are usually designed to operate at flow temperatures of between 40 and 50 °C, with a temperature drop of between 5 and 10 K across the system. Maximum heat output is limited by the maximum acceptable surface temperature to around 100 $W.m^{-2}$ for occupied areas. The overall design of floor heating systems should be undertaken in accordance with the European Standard BS EN 1264[14]. See also section 5.1.1.

Floor heating may be used in conjunction with radiators, for example for the ground floor of a house with radiators on upper floors. Separate circuits are required is such cases, typically using a mixing valve to control the temperature of the under-floor circuit. Floor heating is best suited to well insulated buildings, in which it can provide all the required heating load.

4.3.5 Pumping and pipework

The hydraulic requirements for a system are derived from parameters such as system operating temperature and the heat output required from emitters, which affect pipework layout. The design also needs to take account of the effect of water velocity on noise and corrosion, and the pressure and flow characteristics required of the circulation pump. The key design decisions include:

- system pressures
- whether to use an open or a sealed pressurisation method
- which material to use for pipes
- the flow velocity to be used
- how the system is to be controlled
- filling and air removal arrangements
- pumping requirements, i.e. variable or fixed flow rate.

Details of the characteristics of pipework and pumps are dealt with in sections 5.1.3 and 5.1.4.

4.3.6 Energy storage

Energy storage may either be used to reduce peak loads or to take advantage of lower energy prices at certain times of day. Heat is stored using either solid cores or hot water vessels. The most common application of thermal storage is in dwellings, in which solid core storage is charged with heat at off-peak rates for a 7 or 8 hour period. Guidance for the design of such systems is contained in Electricity Association publication *Design of mixed storage heater/direct systems*[15].

Systems relying on hot water storage vessels are also available for use in dwellings. The three main types are as follows:

- *Combined primary storage units (CPSU):* provide both space and water heating from within a single appliance, in which a burner heats a thermal store. The water in the thermal store is circulated to radiators to provide space heating, while a heat exchanger is used to transfer heat to incoming cold water at mains pressure to provide a supply of domestic hot water.
- *Integrated thermal stores:* also provide both space and water heating from within a single appliance. However, they differ from CPSUs in that a separate boiler is used to heat the primary water.
- *Hot-water-only thermal stores:* use thermal storage only for production of domestic hot water. As for the two types described above, the domestic hot water is provided by a heat exchanger working at mains pressure.

Also, some models of combination boiler contain a small thermal store to overcome the limitation on flow rates for domestic hot water, see section 4.3.3.

Thermal storage for larger buildings must rely on purpose-designed storage vessels with capacity and storage temperature optimised for the heat load. Other design parameters that must be considered are insulation of the storage vessel, arrangements for dealing with expansion and the control strategy for coupling the store to the rest of the system.

4.3.7 Domestic hot water

Whether or not to produce domestic hot water from the same system as space heating is a key decision to be taken before detailed design proceeds. In housing, where demand for hot water is a substantial proportion of the total heat load, a hydronic heating system is usually the most convenient and satisfactory means of producing hot water, using either a hot water storage cylinder or a combination boiler.

In buildings other than housing, the case for deriving domestic hot water from a hydronic heating system depends greatly on circumstances. The demand for hot water and the locations with the building where it is required will affect the relative costs of independent heat generation and connection to the space heating system. In general, independent hot water generation is the more economical choice when relatively small amounts of hot water are required at positions distant from the boiler.

Circulating hot water circuits that require long pipe runs and operate for extended periods solely to provide hot water can waste large amounts of energy, particularly during summer months when no space heating is required. In commercial buildings, toilet areas are often best served by independent gas or electric water heaters.

4.3.8 Control for hydronic systems

Hydronic heating systems are capable of very close control over environmental conditions using a range of strategies. The choice of control system type will depend on the closeness of control required, the number of different zones that must be controlled independently and the times at which the building will be occupied and require heating. The design must also take account of the characteristics of both heat generators and emitters.

A typical control system for a hydronic heating system in a dwelling or small building consists of a programmer, which may incorporate a timeswitch or optimum start/stop functions, a room thermostat for each zone, motorised valves to control the flow to each zone and, if necessary, a frost protection thermostat. Where domestic hot water is also provided by the system, a thermostat and motorised valve to control the temperature of the hot water storage cylinder are also needed. Controls should be wired in such a way that the boiler operates only when a space heating or cylinder thermostat is calling for heat. Thermostatic radiator valves (TRVs) may be used to control individual rooms within a zone. Pump 'over-run' (i.e. delay in switching off a pump) may also be provided by the system or may be incorporated in the boiler controls.

Hydronic systems in larger buildings are likely to have more complex controls, including optimum start, and often incorporate weather compensation in which the system flow temperature is controlled in response to external temperature, according to a schedule derived for the building. Where there are multiple or modular boilers, sequence control is required for the boilers. Variable speed pumping may also be used. The pump speed is usually controlled to maintain a constant pressure differential across a point in the circuit as flow reduces in response to 2-port valve and TRV positions. Care is needed in the choice of valves used for control to ensure good 'valve authority', which means that they are sized appropriately in relation to the pressure drops around the circuit.

Comprehensive guidance on control system design is given in CIBSE Guide H [16] and the characteristics of control system components are given in section 5.1.5.

4.3.9 Water expansion

The density of water reduces significantly as temperature rises which results in significant expansion as a hydronic system warms up from cold. This must be accommodated without an excessive rise in system pressure. Table 4.4 shows the percentage expansion, calculated with reference to 4 °C at start-up for a range of operating temperatures using the expression:

$$(\Delta V/V_4) = (p_4/p)-1 \qquad (4.3)$$

where AV is the change in volume resulting from change in temperature (m^3), V_4 is the volume at 4 °C (m^3), p_4 is the density at 4 °C ($kg.m^{-3}$) and p is the density ($kg.m^{-3}$) at a given temperature.

Allowance may also be made for the expansion of the pipework, but this is small for most materials.

All hydronic systems must have provision for maintaining system operating pressure within a range that ensures safety and effective operation of the system. For low pressure systems this may be achieved by the use of a cistern positioned to maintain pressure by gravity, or by a sealed expansion vessel in which a volume of pressurised gas is separated from the primary water by a diaphragm. In both cases, the system must be able to cope with the expansion of the primary water as the system heats up from cold to its design temperature.

An open system, relying on hydrostatic pressurisation normally has separate feed and open safety vent pipes, with the latter positioned to provide an unrestricted path for the relief of pressure and the escape of steam if the boiler thermostat were to fail and the system overheat. The open safety vent pipe should rise continuously from its point of connection, contain no valves or restrictions and discharge downwards into the feed and expansion cistern. BS 5449[5] recommends that cistern capacity should be at least 5% of system volume to give an adequate margin of safety in operation.

Sealed pressurisation equipment for low pressure systems consists of an expansion vessel complying with BS 4814[17] , a pressure gauge, a means for filling, and a non-adjustable safety valve. Boilers fitted to sealed systems must be approved for the purpose by their manufacturer and must incorporate a high limit thermostat and a safety/pressure relief valve. The expansion vessel contains a diaphragm, which separates the system water from a volume of gas (air or nitrogen). When the system water expands, it enters the vessel, compressing the gas. The vessel must have sufficient volume to accommodate the change in system volume without an excessive increase in

Table 4.4 Percentage expansion of water heating up from 4 °C

Temperature (°C)	Expansion (%)
40	0.79
50	1.21
60	1.71
70	2.27
80	2.90
90	3.63
100	4.34
110	5.20
120	6.00
130	7.00
140	8.00
150	9.10
160	10.2
170	11.4
180	12.8
190	14.2
200	15.7

pressure. BS 7074[18] gives guidance on expansion vessel sizing, initial system pressure and safety valve settings. The expansion vessel should be connected to the return circuit just prior to the pump inlet.

A sealed system has the considerable advantage of eliminating the need for a feed and expansion cistern, placed at a suitable level, and the associated pipework. In housing, this can mean the elimination of pipework and cisterns in the roof space, reducing the risk of frost damage and condensation. A sealed system is also much less prone to corrosion since there is no opportunity for the introduction of air into the system under normal operation. An example calculation for sizing a sealed expansion vessel is given in Appendix Al. 1.

Medium and high pressure systems may use a variety of techniques to maintain working pressure:

- pressurisation by expansion of water, in which the expansion of the water in the system is itself used to charge a pressure vessel
- pressurisation by an elevated header tank
- gas pressurisation with a spill tank, in which a pressure cylinder is partly filled with water and partly with a gas (usually nitrogen)
- hydraulic pressurisation with spill tank, in which pressure is maintained by a continuously running pump.

4.4 Steam systems

4.4.1 Characteristics of steam systems

Steam systems use dry saturated steam to convey heat from the boiler to the point of use, where it is released by condensation. Control of heat output is generally by variation of the steam saturation pressure within the emitter. The resulting condensate is returned to the feed tank, where it becomes a valuable supply of hot feed-water for the boiler. The flow of steam is generated by the pressure drop that results from condensation. Condensate is returned to the lowest point in the circuit by gravity.

Steam offers great flexibility in application and is long established as a medium for heating in buildings. However, it is not frequently chosen as a medium for heating buildings when that is the sole requirement. This is because of more stringent safety requirements and more onerous maintenance requirements than are required for LTHW systems. It is much more likely to be appropriate when there are other requirements for steam, such as manufacturing processes or sterilisation. In such cases, steam may be the most satisfactory medium both for space heating and for domestic hot water generation. In many cases, it will be appropriate to use steam to generate hot water in a heat exchanger for distribution in a standard hydronic heating system.

4.4.2 Types of system/system design

4.4.2.1 Typical steam circuit

A typical steam circuit is shown in Figure 4.2, showing a main pipe carrying steam from the crown valve of the boiler and a second pipe returning condensate to the feed tank. Branch pipes connect individual pieces of equipment or loads to the mains. Condensate from the feed tank is returned to the boiler by the feed pump, which is controlled to maintain the water level in the boiler. Treated water is supplied to the feed tank as required to make up for losses incurred through leaks or venting.

4.4.2.2 Calculation of system loads

The heat requirement may be calculated in the same way as for a hydronic heating system. This may then be converted to a mass flow rate for steam at the design temperature and pressure using steam tables, see CIBSE Guide C[8], which give the specific enthalpy of evaporation in $kJ.kg^{-1}$. A correction should be made for the dryness of the steam, which is typically around 95% and will increase the required mass flow rate pro rata.

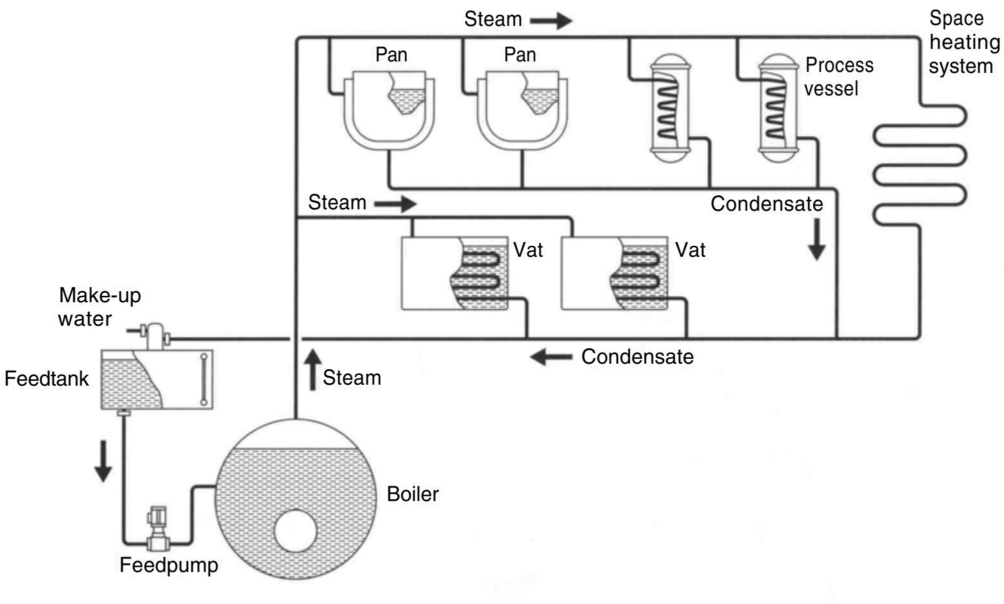

Figure 4.2 Typical steam circuit (courtesy of Spirax-Sarco Ltd)

4.4.2.3 Working pressure

The working pressure at which steam must be circulated depends upon:

- the pressure required where each piece of plant is connected
- the pressure drop along the distribution pipework due to resistance to flow
- pipe heat losses.

As steam at high pressure occupies less volume per unit of mass than steam at low pressure, smaller distribution pipework can be used to achieve a given mass flow rate. This leads to lower capital cost for the pipework and the associated valves, flanges and pipe insulation. Higher pressure also offers the advantages of drier steam at the point of use and increased thermal storage in the boiler. The usual practice is to convey steam to the points of use at high pressure and to provide pressure reduction at the point of use.

4.4.2.4 Pipework sizing

Oversized pipework results in excessive capital costs, greater than necessary condensate formation, and poor steam quality. Undersized pipework causes excessive steam velocity and higher pressure drops, which can cause steam starvation at the point of use as well as a greater risk of erosion and noise. Pipe sizing may be carried out from consideration of the steam velocity required to match the loads around the circuit. In practice, limiting the velocity to between 15 and 25 $m.s^{-1}$ will avoid excessive pressure drops and problems with noise and erosion. Velocities of up to 40 $m.s^{-1}$ may be acceptable in large mains. Sizing may also be carried out from consideration of the steam pressure required at particular pieces of plant.

4.4.2.5 Pressure reducing sets

Steam distributed at a higher pressure than the equipment served requires pressure reduction. The main component in a pressure reducing set is the reducing valve, often a spring loaded diaphragm or bellows type. Simple direct acting reducing valves can be used where the load is small or remains fairly steady. For larger and varying loads a more elaborate, pilot-operated valve may be necessary.

To prevent water or dirt entering the reducing valve it is good practice to install a baffle-type separator and strainer upstream of the valve. Pressure gauges are usually fitted either side of the reducing valve to set the valve initially and to check its operation in use.

It is essential to fit a pressure relief or safety valve on the downstream side of the reducing valve. The relief valve and its discharge pipe must be sized and located to discharge steam safely at the upstream pressure for the maximum capacity of the reducing valve, should it fail wide open.

4.4.2.6 Steam trapping and air venting

Condensation occurs whenever heat is transferred to a load and it must be removed for return to the feed tank. The principal function of a steam trap is to discharge condensate while preventing the escape of dry steam. Air is present within steam supply pipes and steam equipment when the system is started and may also be introduced at other times in solution in the feed water. Air must be removed since it both reduces the capability of a steam system to supply heat and causes corrosion. Some types of steam traps are also designed to remove air and other non-condensing gases from systems. Specialised automatic air vents are fitted at remote points to achieve full air removal.

Condensation takes place in steam mains even when they are well insulated and provision must be made for drainage. Steam mains should be installed with a fall of not less than 100 mm in 10 m in the direction of stearn flow, with collection points arranged as shown in Figure 4.3 using appropriate steam traps. Where possible, branch connections should be taken from the top of the main to avoid the entry of condensate. Low points in branch lines, such as those that occur in front of a control valve, will also accumulate condensate and need provision for trapping and drainage. Steam traps must be sized to remove condensate at the rate needed for cold start-up. A general rule of thumb is to size the condensate return system for twice the mean condensing rate at the operating differential pressure. The characteristics of steam traps and their suitability for particular applications are described in section 5.2.2.

4.4.2.7 Condensate handling

Effective condensate removal and return to the boiler is essential for steam systems to operate properly. As mentioned above, it is important to trap the steam main at low points along its length to ensure that dry steam is available at the point of use.

Temperature control of steam process equipment and heat exchangers is usually achieved by throttling the flow of steam. Consequently, steam pressure falls inside the exchanger. When the steam pressure inside the exchanger is equal to, or lower than the pressure at the outlet side of the steam trap, condensate will not flow. To prevent the

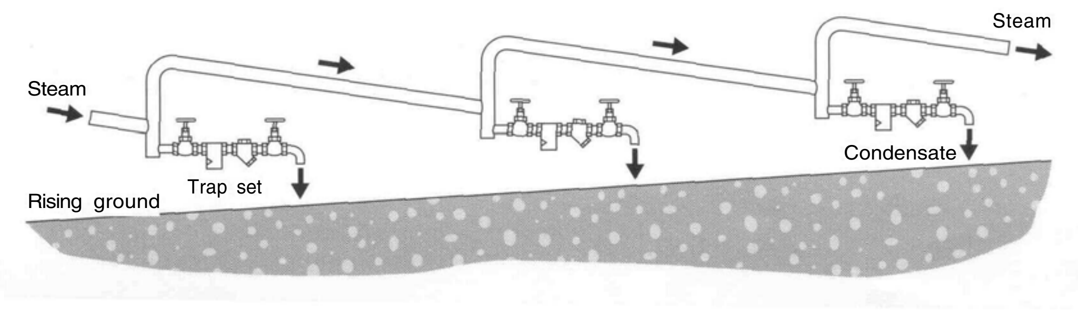

Figure 4.3 Steam main on rising ground showing drainage (courtesy of Spirax-Sarco Ltd)

exchanger from flooding with condensate it is necessary to locate the trap below the exchanger outlet to provide a hydrostatic head to enable condensate to pass through the trap by gravity, the outlet side of the trap normally being kept at atmospheric pressure. A vacuum breaker is often fitted at the steam inlet point of the heat exchanger to admit air in the event that steam pressure inside the exchanger falls below atmospheric pressure. If condensate is to return to the boiler feed tank through pipework at a higher level than the trap, as is usually the case, then the condensate must be pumped, see below.

4.4.2.8 Condensate pumping

A condensate pump set usually comprises an open vented vessel mounted above one or more electric motor pumps or pressure operated lifting pumps, the latter most often using steam but compressed air or other gas may also be used. Condensate from steam traps is piped to discharge into the receiver vessel by gravity.

Electric pumps are usually switched on and off by level controls in the receiver vessel. Special measures regarding electric pumps need to be taken with high pressure steam systems, where condensate temperatures can equal or exceed 100°C.

Pressure operated pumps work by displacing a volume of collected condensate in the pump body. Check valves are fitted on the condensate inlet and outlet of the pump to ensure correct water flow. When the pump body is full of condensate from the receiver an internal mechanism opens the pressurising gas inlet valve. The condensate is pushed through the outlet check valve. At the end of the discharge stroke the mechanism closes the inlet valve and opens an exhaust valve. The 'used' pressurising gas within the pump body then vents either to atmosphere or to the space from which the condensate is being drained. When the pressures are equalised, more condensate can flow by gravity from the receiver into the pump body, and the cycle repeats.

4.4.2.9 Condensate return mains

There are essentially two types of condensate return: gravity and pumped. Traps draining a steam main or device that is always at full steam pressure can vertically lift condensate a limited distance before discharging into a gravity return main laid to fall towards the boiler feed tank. As mentioned above, traps draining heat exchange equipment normally discharge condensate by gravity into a vented receiver from where it is pumped into a separate return main. Gravity condensate return lines carry both condensate and incondensable gases, together with flash steam from the hot condensate. The pipework should be sufficiently large to convey all the liquid, gases and flash steam. An adequately sized pipeline is capable of accepting condensate discharged from traps with different upstream pressures. However, if the pipeline is too small, excessive velocities and pressure drops may arise, particularly where condensate at high pressure and temperature enters the line, giving off flash steam. Such situations often give rise to water-hammer.

Pumped condensate pipes carry only water and can be sized for higher velocities than gravity lines. Trap discharge pipes should not connect directly into pumped condensate pipelines. Flash steam released from additional condensate flowing into a flooded pipe will invariably result in water-hammer.

4.4.2.10 Safety

Every steam boiler must be fitted with a safety valve to protect it from excessive pressure. The safety valve must:

- have a total discharge capacity at least equal to the capacity of the boiler
- achieve full discharge capacity within 110% of the boiler design pressure
- have a minimum valve seat bore of 20 mm
- be set at a pressure no higher than the design pressure of the boiler and with an adequate margin above the normal working pressure of the boiler.

Boilers with a capacity of more than 3700 $kg.h^{-1}$ must have at least two single safety valves or a one double safety valve. All boilers must also be fitted with:

- a stop valve (also known as a crown valve) to isolate the boiler from the plant
- at least one bottom blow-down valve to remove sediment
- a pressure gauge
- a water level indicator.

4.4.3 Guidance and standards

There are many standards and guidance documents relevant to steam systems, including the following:

- Statutory Instrument 1989 No. 2169, The Pressure Systems and Transportable Gas Containers Regulations 1989[19] : provides the legal framework for pressurised vessels.
- BS 1113[20] : covers the design and manufacture of water-tube steam generating plant
- BS 2790[21] : covers the design and manufacture of shell boilers of welded construction , including aspects such as stop valves
- BS 6759-1[22] : covers the specification of safety valves
- BS 759: Part 1[23] : covers valves, mountings and fittings for steam boilers above 1 bar gauge
- Health and Safety Executive PM60[24] : covers bottom blow-down
- BS 1780: Part 2[25] : cover pressure gauges
- BS 3463[26] : covers level indicators
- BS 806[27] : covers drainage of steam lines
- Health and Safety Executive PM5[28] : covers boiler operation.

4.5 Warm air systems

4.5.1 Characteristics of warm air heating

Warm air heating can be provided either by stand-alone heaters or distributed from central air-handling plant; in many cases the same plant is used for summertime cooling/ventilation. Almost all the heat output is provided in convective form so the room air temperature is usually greater than the dry resultant temperature. Warm air systems generally have a much faster response time than hydronic systems. For example, a typical factory warm air system will bring the space up to design temperature within 30 minutes. Warm air systems can cause excessive temperature stratification, with warm air tending to collect at ceiling level. This may be particularly unwelcome in buildings with high ceilings, although it can be overcome by the use of destratification systems.

Warm air systems may be used to provide full heating to a space or simply supply tempered 'make-up' air to balance the heat loss and air flow rate from exhaust ventilation systems. A slight excess air flow can be used to pressurise the heated space slightly and reduce cold draughts.

4.5.2 Layout and design

Warm air systems for housing are often based on stub ducts, radiating from a centrally located furnace. This minimises the length of ductwork required and simplifies installation. Systems used in larger houses, especially in North America, typically rely on long lengths of ductwork distributing heat from a furnace located in a basement. Systems for large commercial buildings are described in CIBSE Guide B2[29] . Such systems typically use ductwork, which may also provide ventilation air and cooling.

For industrial and warehouse buildings, heating is often provided by dedicated warm air heaters.

Most commonly a distributed system using individual warm air heaters rated at between about 20 kW and 300 kW is used. Efficiency is high at about 80% gross. Traditionally these heaters have been floor standing, oil or gas fired and of high output. This minimises initial cost and floor space requirements but provides fairly coarse control of conditions. Current practice typically uses suspended gas fired heaters, rated at up to 100 kW. These are quieter, avoid loss of floor space and provide better heat distribution.

It is necessary to use a de-stratification system (punkah fans or similar) to avoid excess heat loss through the roof and poor comfort at floor level due to temperature stratification, particularly when using suspended heaters. A well designed system can limit temperature differences arising from stratification to only a few degrees, even in buildings with high ceilings.

In tall industrial and warehouse buildings, specialist central plant warm air heating systems are also used. They typically rely on high-temperature, high-velocity primary air supply at high level, supplemented by induction of room air at discharge points to provide good air circulation and even temperatures in the occupied zone.

Electric warm air unit heaters are typically only used in restricted circumstances, such as air curtains at entrance doors, due to their relatively high running cost. Air curtains are described in BSRIA Application Guide AG2/97: *Air curtains - commercial applications*[30] .

Direct fired (flueless) gas warm air heating is sometimes used due to its high efficiency (100% net, 92% gross). While this benefit makes it attractive, particularly if a high ventilation rate is needed, the dispersion of combustion gases into the heated space means that it must be used with care. In particular the ventilation requirements of BS 6230[31] should be met to ensure that co_2 levels are kept low enough to avoid adverse effects on health and comfort.

Care should be taken to ensure that even these low levels of diluted products of combustion do not have adverse an effect on items stored in the heated space, such as premature yellowing of paper and some fabrics due to NO_x levels.

4.5.3 Control of warm air heating

For central plant providing heating and ventilation, the heating component generally places no extra demands on the control system, although care should be taken to ensure that the sensor locations accurately reflect zone temperatures in the heating mode.

For individual warm air heaters it is usual to provide a separate thermostat or sensor to control each heater although, exceptionally, up to four small heaters in one space may be controlled together. Time control is usually by simple time-switch, since the fast response of warm air heaters makes optimum start/stop of limited benefit

4.5.4 Restrictions on use

Flueless appliances may only be used in accordance with the requirements of the Building Regulations Part J[32] . Noise generated by warm air distribution may also restrict the use of warm air heating in some circumstances.

4.6 Radiant systems

4.6.1 Characteristics of radiant heating

In general, systems are considered to be radiant when more than 50% of their output is radiant, which corresponds broadly to those with emitter temperatures greater than 100 °C. This definition includes medium temperature systems, such as high pressure hydronic systems, steam systems and air heated tubes, which operate at temperatures up to 200 °C. High temperature radiant systems, such as those with electric radiant elements or gas heated plaques, produce a higher proportion of their output in radiant form and are particularly effective when heat output needs to be focussed and directed to specific locations.

Radiant heating is particularly useful in buildings with high air change rates or large volumes that do not require uniform heating throughout, e.g., factories, and intermittently heated buildings with high ceilings. The key characteristics of radiant heating are as follows:

- Heat transfer occurs by radiation directly on surfaces, including building occupants and the internal surfaces of buildings and fittings. The surrounding air need not be heated to the same temperature as would be required with convective heating.
- A rapid response can be achieved because the effect of the thermal inertia of the building is bypassed by direct radiation.
- After an initial warm-up period, radiant heating directed downwards towards floor level is augmented by re-radiation and convection from surfaces at the level occupied by people.
- Radiant asymmetry is a potential problem and may place restrictions on design.

Radiant heating can require less energy than convective heating because it enables comfort conditions to be achieved at lower air temperatures. As a general rule it is likely to have an advantage in this respect whenever ventilation heat losses exceed fabric heat losses. Further savings may be achieved when only some zones within a large open area require heating and local radiant temperature can be raised by well directed radiant heat. In such cases, large volumes of surrounding air may be left at much lower temperatures without a detrimental effect on dry resultant temperature in the working zones.

4.6.2 Layout and design of radiant heating systems

There are two basic approaches to radiant heating design:

- *Spot heating:* applies to the situation described in the preceding paragraph, in which the intention is to heat only a small part of a larger space. In such cases, comfort depends mainly on direct radiant output from the heaters and there is little effect on the overall air temperature in the building.
- *Total heating:* applies to situations in which the whole space must be heated to a uniform temperature.

Detailed guidance on the design of radiant heating systems is given in BSRIA Application Guide AG3/96: *Radiant heating*(33) .

For spot heating, standard heat loss calculations are not appropriate for calculating the output required from emitters. Relatively high levels of irradiance are required to produce the necessary dry resultant temperature and it is necessary to determine the distribution of radiant energy within the space. To achieve this, it is necessary to know the directional characteristics of each heat emitter. For an air temperature of 15 °C, the maximum irradiance recommended(33) at floor level is 80 $W.m^{-2}$, which places limitations on the mounting height of emitters. Total spherical irradiance at 1.8 m above floor level is recommended not to exceed 240 $W.m^{-2}$. These figures are considered conservative for industrial heating applications and may be exceeded with caution. However, account should be taken of temperatures reached on surfaces close to heaters, for example on the tops of shelving. When considering the use of spot radiant heating, it is important to consider relative humidity of the air in the building. Contact between moist air and cold surfaces away from the heated areas may cause problems with condensation, particularly where flueless gas radiant heaters are used.

When designing for total radiant heating relying on low and medium temperature emitters, the procedure is similar to that required for other heating systems, involving consideration of fabric and ventilation heat loss and the calculation of total heat output required. Designs typically assume that air temperature will be around 3 °C below dry resultant temperature.

4.6.3 Control of radiant heating

The sensing of temperature for the control of radiant heating presents difficulties both in sensing dry resultant temperature and in finding an appropriate location for the sensor. A black-bulb thermometer needs to be located centrally in a zone to avoid influence by proximity to a wall. Hemispherical black-bulb sensors are available for wall mounting, but are often difficult to set in relation to perceived comfort conditions.

Air temperature sensors may be used to control radiant heating, particularly where total heating is provided. However, they tend to underestimate dry resultant temperature during warm up and cause waste of energy.

4.6.4 Restrictions of use of radiant heating

Physical restrictions on the mounting of radiant emitters apply. High temperature emitters must not be placed where they can come into contact with people or objects that cannot withstand the resulting surface temperatures. Also, the irradiance from emitters limits their proximity to working areas. Consequently, radiant heating may be considered unsuitable for use in buildings with low ceilings. Table 4.5 shows typical restrictions on mounting height for various types of radiant heat emitter.

Table 4.5 Minimum heights for radiant heat emitters (source BSRIA AG3/96[33])

Emitter type	Input rating (kW)	Minimum height (m)
Gas radiant U-tube	13	3.0
	22	3.6
	38	4.3
Gas plaque heater	13.5	4.2
	27	7.0
Gas cone heater	12	3.6
Quartz tube heater	3	3.0
	6	4.5

Despite its obvious advantages for partially heated buildings, 'spot' radiant heating does not offer good control of temperature. It should not be considered, therefore, where close temperature control is required.

4.7 Plant size ratio

4.7.1 Definition of plant size ratio

Heating systems are designed to meet the maximum steady-state load likely to be encountered under design conditions. However, additional capacity is needed to overcome thermal inertia so that the building may reach equilibrium in a reasonable time, particularly if the building is heated intermittently.

Plant size ratio (PSR) is defined as:

$$\text{PSR} = \frac{\text{installed heat emission}}{\text{design heat load}}$$

The design heat load used in the calculation of PSR is the heat loss from the space or building under conditions of external design temperature and internal design temperature. For the purpose of specifying the heating system this condition should be calculated for the time of peak steady state load. The time at which this occurs will depend on the building or space, its services and its occupancy. Peak load normally occurs under one of the following conditions:

- *during occupancy:* taking account of any reliable internal heat gains, fabric heat losses and all ventilation heat losses
- *before occupancy:* taking account of any permanent internal heat gains (but not those occurring only during occupied periods), fabric heat losses and all ventilation losses (unless ventilation systems operate during occupied periods only, in which case only infiltration losses are applicable).

4.7.2 Intermittent heating

Intermittent occupancy permits a reduction in internal temperature while the building is unoccupied and a consequent reduction in fuel consumption. It is important to note that the building continues to lose heat during the off period and requires additional heat to bring the building back up to temperature during the 'pre-heat' period prior to the next period of occupancy. For many buildings, the pre-heat period can constitute the major energy consumption of the building. The shaded area in Figure 4.4 represents the accumulated temperature reduction (in degree-hours), which is directly related to the energy saved by the system due to the reduction in space temperature during the period of non-occupancy. A building having low thermal inertia, which cools to a lower temperature when the heating system is off, will experience greater economy as a result of intermittent heating, than a building of high thermal inertia, see Figure 4.5. However, it should be noted that high thermal inertia is beneficial in that it enables better utilisation of heat gains.

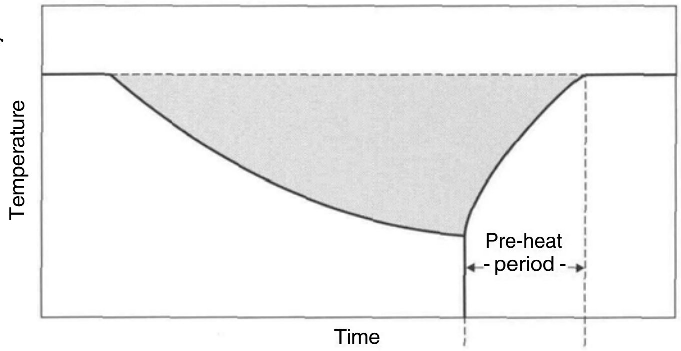

Figure 4.4 Temperature profile of a space during intermittent heating with the pre-heat period optimised to be as short as possible

The necessary plant size ratio required to reach design temperature for a particular building depends on the occupancy and heating pattern. For many buildings, the most demanding situation arises on Monday morning after being unoccupied during the weekend. If the system is shut off completely during the weekend, the building may have to be heated up from a room temperature little higher than the outside temperature. The heating system may also be operated at a set-back temperature when it is not occupied, in which case less energy is required to restore it to design temperature. It may also be observed from Figure 4.5 that a building with low thermal inertia heats up more quickly than one with high thermal inertia and therefore a lower plant size ratio may be employed.

4.7.3 Choice of plant size ratio

The shorter the pre-heat period, the greater is the saving in energy. This implies that the greater the plant size ratio, the greater the economy in energy consumption. However there are several disadvantages in over-sizing the heating system:

- greater capital cost
- more difficult to achieve stability of controls
- except during pre-heat, the plant will run at less than full load, generally leading to a lower seasonal efficiency.

The optimum plant size ratio is difficult to determine as it requires knowledge, or estimates, of:

- the occupancy pattern
- the thermal inertia or thermal response of the building areas

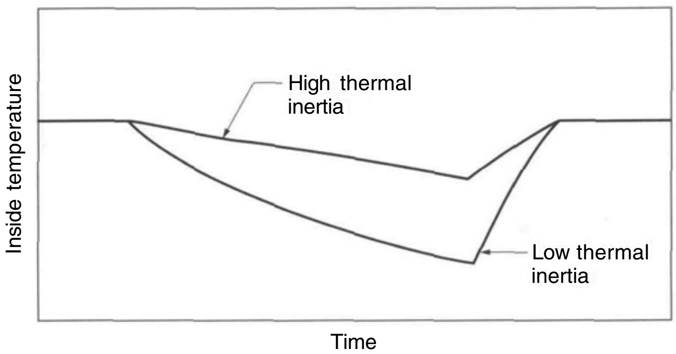

Figure 4.5 Profile of space temperature for buildings of high thermal inertia and low thermal inertia, each having the same plant size ratio

- the design internal temperature
- the minimum permissible internal temperature
- a record of the weather over a typical season
- the current fuel tariffs and estimates of future tariffs over the life of the system
- the capital and maintenance costs of different sizes of equipment.

Section 5 of CIBSE Guide A[34] deals with thermal response, including descriptions of steady-state and dynamic models. Fully functional dynamic models are too complex for hand calculation and in practice must be implemented through carefully developed and validated software. CIBSE Applications Manual AM11[35] gives guidance on the selection of suitable models. For complex buildings, it is recommended that plant size ratio be calculated using a dynamic simulation of the building and the plant.

For less complex buildings, CIBSE Guide A, section 5.8.3.3, describes a method of calculating plant size ratio based on the admittance procedure:

$$F_3 = \frac{24 f_r}{H f_r + (24 - H)} \tag{4.5}$$

where F_3 is the plant size ratio (or 'intermittency factor'), f_r is the thermal response factor (see equation 4.6) and H is the hours of plant operation (including preheat) (h).

The response factor may be calculated from:

$$f_r = \frac{\Sigma\,(A\,Y) + C_v}{\Sigma\,(A\,U) + C_v} \tag{4.6}$$

where f_r is the thermal response factor, $\Sigma\,(A\ Y)$ is the sum of the products of surface areas and their corresponding thermal admittances (W.K^{-1}), $\Sigma\,(A\ U)$ is the sum of the products of surface areas and their corresponding thermal transmittances over surfaces through which heat flow occurs (W.K^{-1}) and C_v is the ventilation heat loss coefficient(W.K^{-1}).

The ventilation heat loss coefficient is given by:

$$C_v = (c_p\, p N V) / 3600 \tag{4.7}$$

where c_p is the specific heat capacity of air (J.kg^{-1}. K^{-1}), p is the density of air (kg.m^{-3}), N is the number of air changes in the space (h^{-1}) and V is the room volume (m^3).

For air at ambient temperatures, $p \approx 1.20$ kg.m^{-3} and $c_p \approx$ 1000 J.kg^{-1}. K^{-1}, hence:

$$C_v \approx N V / 3 \tag{4.8}$$

Table 4.6 shows plant size ratios for a range of heating periods and thermal response factors. Structures with a response factor greater than 4 are referred to as slow response or 'heavyweight', and those with a response factor less than 4 as fast response or 'lightweight'. CIBSE Guide A recommends that when the calculation yields a result of less than 1.2, a plant size ratio of 1.2 should be used.

Plant sizing as described above is based on ensuring that the heating system is able to bring the building up to design temperature in the required time. A more comprehensive approach, including economic appraisal, is described in a paper by Day et al [36]. This proposes a new method for calculating the pre-heat time required, which takes account of the plant capacity in relation to the mean temperature of the whole daily cycle. It goes on to optimise plant size by finding the minimum life cycle cost, taking account of both capital and running costs. The paper also reports conclusions reached from applying the model to a large gas-fired system (750 kW), as follows:

- The greater the thermal capacity of the building, the smaller the optimal plant size ratio. In determining the effective thermal capacity of the building, as a general guide, the first 100 mm of the inner fabric skin should be taken into account.
- For the particular case studied, the optimum plant size ratio was found to be 1.63 but the economic savings which result from this choice do not vary significantly for plant size ratios of ± 10% of the optimum.
- Plant size ratio >2.0 are not justified for most typical buildings.
- Smaller plants have higher values of marginal installed cost (£/extra kW), so the optimum plant size ratio will be lower.

In general, it may be observed that, unless rapid warm-up is essential, plant size ratio should be in the range 1.2 to 2.0. Optimum start control can ensure adequate pre-heat time in cold weather.

Table 4.6 Plant size ratio calculated for different heating periods

Heating hours (including pre-heat period)	Thermal weight		
	Light (f_r = 2)	Medium (f_r = 4)	Heavy (f_r = 8)
6	1.6	-	-
7	1.5	-	-
8	1.5	2.0	-
9	1.5	1.9	-
10	1.4	1.8	2.0
11	1.4	1.7	1.9
12	1.3	1.6	1.8
13	1.3	1.5	1.7
14	1.3	1.5	1.6
15	1.2	1.4	1.5
16	1.2	1.3	1.4

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5 Plant and equipment

5.1 Equipment for hydronic systems

5.1.1 Heat emitters

5.1.1.1 Radiators and convectors

Both radiators and convectors emit heat by virtue of their surface temperatures being greater than the room air temperature and the mean radiant temperature of the surfaces surrounding them. In each case, heat is emitted by both radiation and convection. Even for a 'radiator', the convective component may be well over half the heat emission when fins are included either behind or between panels.

Manufacturers are obliged to quote the nominal output of the emitter under a standard method for testing as specified in BS EN 442-2[1] .

The standard emission is under conditions of 'excess temperature' of 50 K, i.e:

$$\Delta T = (t_m - t_{ai}) = 50 \qquad (5.1)$$

where ΔT is the excess temperature (K), t_m is the mean water temperature within the emitter (°C) and t_{ai} is the temperature of the surrounding air (°C).

The test conditions require that the surrounding mean radiant temperature does not differ significantly from the surrounding air temperature. They also require that the inlet and outlet temperatures should be 75 °C and 65 °C respectively in surroundings at 20 °C. The designer is not obliged to adhere to these temperatures.

The 'water-side' of the heat exchange is given by:

$$\phi = q_m \, c_p \, (t_1 - t_2) \qquad (5.2)$$

where ϕ is the heat emission (W), q_m is the mass flow rate ($kg.s^{-1}$), c_p is the specific heat capacity of water ($J.kg^{-1}.K^{-1}$), t_1 is the inlet temperature (°C) and t_2 is the outlet temperature (°C).

The 'air-side' of the heat exchange is given by:

$$\phi = K_m \, \Delta T^n \qquad (5.3)$$

where K_m is a constant for a given height and design of emitter and n is an index.

The value of c_p for water varies slightly with temperature, see Table 5.1.

The effects of architectural features and surface finish on radiator output are summarised in Table 5.2. In general, it may be observed that heat output is reduced when airflow is restricted, such as by placing a shelf immediately above a radiator, or by an enclosure. It is also reduced by surface finishes with low emissivity, such as metallic paints or plating.

Radiator output is also affected by the form of connection to the system pipework. Testing is commonly done with top and bottom opposite end (TBOE) connections. Other forms of connection produce different outputs which may be corrected for by applying factors obtained from manufacturers.

5.1.1.2 Fan coil heaters

The characteristics of fan coil heaters are described in BS 4856[2] , which gives test methods for heat output and air movement with and without attached ducting, and for noise levels without attached ducting. The heat output from fan coil heaters is approximately linear with the difference between system temperature and room air temperature, corresponding to $n = 1.0$ in equation 5.3.

The output from fan coil units is generally more sensitive to airflow problems than to water circulation and this should be borne in mind both at the design stage and when investigating problems. Other practical difficulties with fan coil units can arise from the use of copper tubing in their fabrication, which can lead to corrosion if traces of sulphides remain following manufacture.

Table 5.1 Values of specific heat capacity and density of water

Temperature /°C	Specific heat capacity c_p/kJ.kg^{-1}.K^{-1}	Density p/ kg.m^{-3}
10	4.193	999.7
20	4.183	999.8
30	4.179	995.6
40	4.179	992.2
50	4.182	988.0
60	4.185	983.2
70	4.191	977.8
80	4.198	971.8
90	4.208	965.3
100	4.219	958.4
110	4.233	950.6
120	4.248	943.4
130	4.27	934.6
140	4.29	925.9
150	4.32	916.6
160	4.35	907.4
170	4.38	897.7
180	4.42	886.5
200	4.50	864.3

Table 5.2 Effects of finishes and architectural features on radiator output

Feature	Effect
Ordinary paint or enamel	No effect, irrespective of colour.
Metallic paint such as aluminium and bronze	Reduces radiant output by 50% or more and overall output by between **10** and **25%.** Emission may be substantially restored by applying two coats of clear varnish.
Open fronted recess	Reduces output by 10%.
Encasement with front grille	Reduces output by 20% or more, depending on design.
Radiator shelf	Reduces output by 10%.
Fresh air inlet at rear with baffle at front	May increase output by up to 10%. This increase should not be taken into account when sizing radiator but should be allowed for in pipe and boiler sizing. A damper should always be fitted.
Distance of radiator from wall	A minimum distance of 25 mm is recommended. Below this emission may be reduced due to restriction of air-flow.
Height of radiator above floor	Little effect above a height of 100 mm. If radiators are mounted at high level, output will depend on temperature at that level and stratification may be increased.

5.1.1.3 Variation of heat emitter output with system water temperature

The variation with mean water temperature depends upon the characteristics of the individual emitter. If correction factors are not given by the manufacturer, then reasonably accurate values can be obtained using equation 5.3 above.

BS EN 442-2 obliges the manufacturer to test the radiator at excess temperatures $\Delta T = 30$ *K,* 50 K and 60 K so as to determine the value of *n.* Thus if the test conditions are not precisely those specified, the experimental readings can be adjusted to correspond to the nominal conditions. The manufacturer is not obliged to publish the value of *n* but some manufacturers give data for both $\Delta T = 50$ *K* and $\Delta T = 60$ *K.* From such data it would be possible to deduce the value of *n* using:

$$n = \frac{\ln(\phi_{60}/\phi_{50})}{\ln(60/50)} \qquad (5.4)$$

where ϕ_{60} is the heat emission at 60 °C (W) and ϕ_{50} is the heat emission at 50 °C (W).

A value of $n = 1.24$ has been obtained from the quoted outputs of one manufacturer, but values of up to 1.33 may be encountered.

Then for any value of ΔT, the output can be determined from:

$$\phi = \phi_{50}\ (\Delta T/50)^n \qquad (5.5)$$

5.1.1.4 Variation of emitter heat output with water flow rate

Although a lower flow rate might cause a slight decrease in the water-side convection coefficient, this small increase in resistance is trivial in comparison with the overall resistance. Thus it is reasonable to consider that the overall heat transfer coefficient will remain constant. A reduction in the mass flow rate of the water has a greater effect on the mean water temperature and it is this that affects the heat emission.

One way of reducing emitter output and reducing pump power consumption is to reduce the pump speed, and hence the mass flow. The effect is considered here, assuming that the flow temperature t_1 remains constant. The mathematics involves equating the water-side and air-side heat transfer equations (equations 5.2 and 5.3). i.e:

$$q_m\ c_p\ (t_1 - t_2) = K_m\ \Delta T^n \qquad (5.6)$$

The mean water temperature, $t_m = (t_1 + t_2)/2$. Therefore, from equation 5.1:

$$\Delta T = \frac{(t_1 + t_2)}{2} - t_{ai} \qquad (5.7)$$

Hence, substituting into equation 5.6:

$$q_m\ c_p\left(t_1 - t_2\right) = K_m\ \left[\tfrac{1}{2}\left(t_1 + t_2\right) - t_{ai}\right]^n \qquad (5.8)$$

Rearranging in terms of the unknown return temperature, t_2, gives:

$$t_2 = t_1 - \frac{K_m}{q_m c_p}\left[\tfrac{1}{2}\left(t_1 + t_2\right) - t_{ai}\right]^n \qquad (5.9)$$

Equation 5.9 contains t_2 on both sides of the equation. Once a starting value is inserted in the right hand side of the equation, the value oft, may be obtained by iteration. Equation 5.2 will then readily yield the heat output.

The example calculation in Appendix A1.2 shows how to calculate the heat output for conditions other than nominal. Although shown for a change in flow rate only, the same technique could be used if using a different flow temperature, tl.

Figure 5.1, which was obtained using the above method, shows the effect on emitter output for flow rates less than nominal. It can be seen that whatever the design value of water temperature drop ($t_1 - t_2$), an appreciable reduction in water flow rate causes little reduction in heat output. Thus, except when full heat output is required (during the pre-heat period), there is no need for the pumps to run at full speed. Similarly it can be seen that increasing the flow above the design flow does not boost the heat output appreciably. A change in flow temperature from 75 °C to 65 °C does not make a significant difference to the shape of the curves.

Table 5.3 Heat emission from plane surfaces by radiation

Surface temp (°C)	Heat emission / W.m^{-2} for stated surface emissivity and enclosure mean radiant temperature (°C)																				
	Surface emissivity = 0.3							Surface emissivity = 0.6							Surface emissivity = 0.9						
	10	12.5	15	17.5	20	22.5	25	10	12.5	15	17.5	20	22.5	25	10	12.5	15	17.5	20	22.5	25
20	16	12	8.3	4.2	0	-4.3	−8.8	33	25	17	8.4	0	−8.7	−18	49	37	25	13	0	−13	−26
30	34	30	26	22	18	14	9.2	69	61	53	44	36	27	18	103	91	79	67	54	41	28
40	54	50	46	42	38	34	29	108	100	92	84	76	67	58	162	151	139	126	114	101	87
50	76	72	68	64	60	55	51	152	144	136	128	120	111	102	228	216	204	192	179	166	153
60	100	96	92	88	84	79	75	200	192	184	176	168	159	150	300	288	276	264	251	238	225
70	126	122	118	114	110	106	101	253	245	237	229	220	211	203	379	367	355	343	330	317	304
80	155	151	147	143	139	134	130	310	302	294	286	278	269	260	465	453	441	429	416	403	390
90	186	182	178	174	170	166	161	372	365	357	348	340	331	322	559	547	535	523	510	497	484
100	220	216	212	208	204	200	195	440	432	424	416	408	399	390	660	649	637	624	612	599	585
120	297	293	289	285	280	276	272	593	586	577	569	561	552	543	890	878	866	854	841	828	815
140	386	382	378	374	370	365	361	772	764	756	747	739	730	721	1160	1150	1130	1120	1110	1100	1080
160	489	485	481	477	473	468	464	978	970	962	954	945	936	928	1470	1450	1440	1430	1420	1400	1390

Table 5.4 Heat emission from plane surfaces by convection

Surface temp. (°C)	Heat emission / W.m^{-2} for stated direction and air temperature (°C)																				
	Horizontal looking down							Vertical							Horizontal looking up						
	10	12.5	15	17.5	20	22.5	25	10	12.5	15	17.5	20	22.5	25	10	12.5	15	17.5	20	22.5	25
20	11	7.9	4.8	2.0	0	−5.8	−14	30	20	12	4.7	0	-4.7	−12	36	25	14	5.8	0	−2.0	
30	27	23	19	15	11	7.9	4.8	75	63	51	40	30	20	12	91	77	62	49	36	25	-4.8
40	45	40	36	31	27	23	19	129	115	101	88	75	63	51	157	140	123	107	91	77	14
50	64	59	54	50	45	40	36	189	174	158	144	129	115	101	230	211	192	174	157	140	62
60	85	80	75	69	64	59	54	255	238	221	205	189	174	158	309	289	269	249	230	211	123
70	107	101	96	90	85	80	75	324	307	289	272	255	238	221	394	372	351	330	309	289	192
80	130	124	118	112	107	101	96	398	379	361	342	324	307	289	484	461	438	416	394	372	269
90	153	147	141	135	130	124	118	476	456	436	417	398	379	361	578	554	530	507	484	461	351
100	177	171	165	159	153	147	141	556	536	516	495	476	456	436	675	651	626	602	578	554	438
120	228	222	215	209	202	196	190	726	705	683	661	640	619	598	882	856	829	803	777	751	530
140	281	274	267	261	254	248	241	907	884	861	838	816	793	771	1100	1070	1050	1020	990	963	726
160	336	329	322	315	308	301	295	1100	1070	1050	1020	1000	977	954	1330	1300	1270	1240	1220	1190	936

1160

Table 5.5 Heat emission from single horizontal steel pipes with a surface emissivity of 0.9 and freely exposed to ambient air at temperatures between 10 and 20 °C

Nominal pipe size /mm	Heat emission / W.m^{-2} for stated temperature difference between surface to surroundings / K																			
	40	45	50	55	60	65	70	75	80	100	120	140	160	180	200	220	240	260	280	300
15	42	48	55	62	69	77	84	92	100	135	173	215	261	311	366	425	490	560	635	717
20	51	59	67	75	84	93	103	112	122	164	211	262	318	380	447	520	600	686	780	881
25	62	71	81	92	102	114	125	137	149	200	257	320	389	465	547	637	735	842	957	1080
32	75	87	99	112	125	138	152	167	181	244	314	391	476	569	670	781	902	1030	1180	1330
40	84	98	111	125	140	155	170	186	203	273	352	438	534	638	753	878	1010	1160	1320	1500
50	106	118	135	152	169	188	206	226	246	331	427	532	648	776	916	1070	1240	1420	1620	1830
65	125	145	165	186	207	230	253	277	301	406	523	653	796	954	1130	1320	1520	1750	2000	2260
80	143	166	189	213	238	263	290	317	345	466	600	750	915	1100	1300	1510	1750	2010	2300	2610
100	179	207	236	266	297	329	362	396	431	582	750	937	1140	1370	1620	1900	2200	2530	2890	3280
125	214	247	281	317	354	392	432	473	515	696	897	1120	1370	1650	1950	2280	2650	3040	3480	3950
150	248	287	327	368	411	456	502	549	598	808	1040	1310	1600	1920	2270	2660	3090	3550	4060	4620
200	319	369	421	474	529	586	646	706	769	1040	1340	1680	2060	2480	2940	3450	4000	4610	5280	6010
250	389	449	512	577	644	714	786	860	937	1270	1640	2050	2520	3030	3600	4220	4900	5650	6470	7370
300	453	524	597	673	751	832	916	1000	1090	1480	1910	2400	2940	3540	4200	4930	5740	6620	7590	8650

Table 5.6 Heat emission from single horizontal copper pipes freely exposed to ambient air at temperatures of 20 °C

Nominal pipe size /mm	Heat emission / W.m^{-2} for stated surface finish and temperature difference between surface and surroundings / K																			
	Painted pipe (ε = 0.95)										Tarnished pipe (ε = 0.5)									
	40	45	50	55	60	65	70	75	80	100	40	45	50	55	60	65	70	80	90	100
8	18	21	24	27	30	33	37	40	43	58	15	17	20	22	25	27	30	33	36	48
10	22	25	29	32	36	40	44	48	52	70	18	21	24	27	30	33	36	39	43	57
15	31	36	41	46	51	57	62	68	74	99	25	29	33	37	41	46	50	55	60	80
22	43	49	56	63	71'	78	86	94	103	138	34	39	45	51	56	62	69	75	81	109
28	53	61	69	78	87	97	106	116	126	170	42	48	55	62	69	76	84	91	99	133
35	64	74	84	95	106	117	129	141	153	206	50	58	66	74	83	92	101	110	120	160
42	75	86	98	111	124	137	151	165	179	242	58	67	77	86	96	107	117	128	139	186
54	93	107	122	138	154	171	188	205	223	301	72	83	94	106	119	131	144	158	171	230
76	125	145	165	186	208	230	253	277	302	407	95	110	126	142	158	175	192	210	229	306
108	171	197	225	253	283	313	345	377	411	554	128	148	169	190	212	235	258	282	307	412
133	205	237	270	305	340	377	415	454	494	668	153	177	201	227	253	280	308	337	366	492
159	240	278	317	357	399	442	486	532	579	783	178	205	234	264	294	326	358	392	426	572

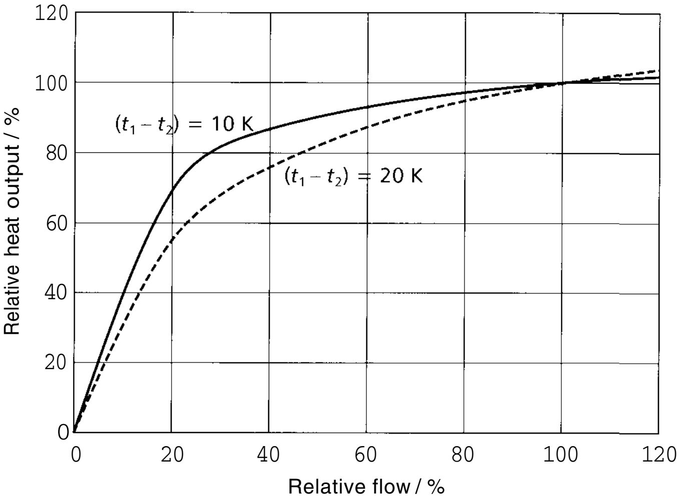


Figure 5.1 Heat emission of a radiator having n = 1.25 and t_1 = 75 °C for design values of $(t_1 - t_2)$ = 10 K and 20 K.

Table 5.7 Correction factors for for Tables 5.5 and 5.6 for heat emission from vertical pipes

Pipe size / mm	Correction factor
8	0.72
10	0.74
15	0.76
20	0.79
25	0.82
32	0.84
40	0.86
50	0.88
65	0.90
80	0.92
100	0.95
125	0.97
150	0.99
200	1.03
250	1.05
300	1.07

Table 5.8 Correction factors for Tables 5.5 and 5.6 for heat emission from horizontal pipes in banks

Number of pipes in bank	Correction factor
2	0.95
4	0.85
6	0.75
8	0.65

5.1.1.5 Plane surfaces

Heat emitted from plane surfaces, e.g. panels or beams, may be estimated using Tables 5.3 and 5.4, which have been calculated using the data given in CIBSE Guide C [3], section 3.3.4. Radiative and convective outputs are given separately to assist where significant differences between air and mean radiant temperature are expected in heated areas. The convective output applies to draught-free conditions; significantly increased output may be available where there is air movement. For example, a local air movement velocity of 0.5 $m.s^{-1}$ could be expected to increase convective output by around 35%. In practice, the heat output from a vertical surface varies with the height of the surface.

5.1.1.6 Heat emission from distribution pipework

Account needs to be taken of the heat emitted from distribution pipework when sizing both emitters and boilers. Large diameter pipes may also be used as heat emitters by design, but this is no longer common practice. Tables 5.5 and 5.6 give heat emissions per metre horizontal run for steel and copper pipes respectively. When pipes are installed vertically, heat emissions are different due to the differences in the boundary layer or air around the pipe surface. Table 5.7 gives correction factors for vertical pipes. When pipes are arranged in a horizontal bank, each pipe directly above another at close pitch, overall heat emission is reduced. Table 5.8 gives correction factors for such installations.

Heat emission from pipes and plane surfaces is covered in detail in CIBSE Guide C [3], section 3.3.

5.1.1.7 Heat emissions from room surfaces

Room surfaces may be designed to emit heat or, in other cases, heat emissions arising from surfaces may need to be taken into account as heat gains in the design of systems. Tables 5.3 and 5.4 may be used for this purpose.

Surface temperatures must be limited to a level that will not cause discomfort to building occupants, taking account of thermal gradients and asymmetrical thermal radiation, see section 3.2. CIBSE Guide A [4], section 1.4.3, notes that local discomfort of the feet can be caused by either high or low temperatures. For rooms in which occupants spend much of their time with bare feet (e.g. changing rooms and bathrooms), it is recommended that floor temperatures should lie within the ranges shown in Table 5.9. For rooms in which normal footware is expected to be worn, the optimal surface temperature for floors is 25 °C for sedentary occupants and 23 °C for standing or walking occupants. Flooring material is considered to be unimportant in these circumstances.

5.1.1.8 Floor heating

The general characteristics of floor heating are described in section 4.3.4 above. The floor surface itself is used as a heat emitter and heat is supplied by the circulation of water as part of a hydronic system, through appropriately spaced pipes positioned beneath the floor surface.

BS EN 1264 [5] deals with floor heating; Part 1 gives definitions and symbols, Part 2 gives a method for the determination of thermal output, Part 3 deals with dimensioning and Part 4 deals with installation.

Much of the equipment required for floor heating systems is the same as that used for other hydronic heating

Table 5.9 Comfortable temperatures for barefoot occupants for typical floor surfaces

Material	Surface temperature range (°C)
Textiles	21 to 28
Pine wood	21.5 to 28
Oakwood	24.5 to 28
Hard thermoplastic	24 to 28
Concrete	26 to 28

systems. However, the heat emitting floor surfaces require careful design to produce the required surface temperatures and heat output. Surface temperature should not exceed 29 °C in general or 35 °C for peripheral areas, which are defined in BS EN 1264 as 'generally an area of 1 m maximum in width along exterior walls' and 'not an occupied area'.

BS EN 1264 gives the heat output available from the floor surface as:

$$\phi = 8.92(t_{fm} - t_i)^{1.1} \tag{5.10}$$

where ϕ is the heat output per unit area of floor (W.m^{-2}), t_{fm} is the average floor temperature (°C) and t_i is the room temperature (°C).

The limitation on surface temperature leads to a corresponding limitation on heat output. For a room temperature of 20 °C, the maximum output is around 100 W.m^{-2} in general and 175 W.m^{-2} at the periphery.

The designer's task is to ensure that the heat flow density at the floor surface is such as to maintain design surface temperatures. Calculations need to take account of the spacing and diameter of embedded pipes, the thickness and heat conductivity of the material between the pipes and the floor surface (including floor covering), and the properties of pipes and any heat conducting devices used to distribute heat within the floor material. BS EN 1264-2 gives procedures for systems with pipes embedded in the floor screed and those with pipes below the screed.

5.1.2 Heat sources

5.1.2.1 Boilers

Boilers intended for use in hydronic systems are available in a wide range of types, constructions and output ranges, and suitable for use with different fuels. Many standards and codes of practice relate to boilers, covering their construction, the combustion equipment required for each type of fuel, and their installation and commissioning. The recommendations of HSE Guidance Note PM5 [6] should be followed in all cases.

Cast iron sectional boilers

Boilers of this type are constructed out of sections joined by barrel nipples, with the number of sections selected to produce the required output. They are normally operated at pressures below 350 kPa and have outputs of up to 1500 kW. Where access is limited, the boiler may be delivered in sections and assembled on site. It is important that water flow be maintained at all times to meet the manufacturer's recommendations, including a period after shutdown to disperse residual heat. Boilers of this type are covered by BS 779 [7].

Low carbon steel sectional boilers

These are similar to cast iron boilers except that their sections are made of steel and similar recommendations apply.

Welded steel and reverse flow boilers

Welded steel and reverse flow boilers are fabricated from steel plate. The combustion chamber is pressurised and a 'blind' rear end reverses the burner discharge back over the flame, in counter-flow. The gases then pass through a circumferential ring of fire tubes around the combustion chamber. This arrangement achieves high efficiency and compactness. They are typically designed for a maximum working pressure of 450 kPa but can be designed to operate at up to 1 MPa, with outputs between 100 kW and 3 MW. Boilers of this type are covered by BS 855 [8].

Steel shell and fire-tube boilers

Steel shell and fire-tube boilers consist of a steel shell and a furnace tube connected to the rear combustion chamber, from which convection tubes are taken to provide two-pass or three-pass operation. Boilers of this type are suitable for pressures up to 1 MPa and are available with outputs up to 12 MW and are often used for steam applications (see also section 5.2). The relevant standard is BS 2790 [9].

Multiple or modular boilers

Multiple or modular boilers are designed to operate in installations in which the number of boilers firing is matched to the load on the system. The result is that the load on each boiler remains high even when the system load is low, leading to higher operating efficiency. Reliability is also improved, as the unavailability of a single boiler does not shut down the entire system. Multiple boilers are typically operated in parallel, under a sequence controller that detects the load on the system and brings individual boilers into the circuit as required. For circuits with two-port valves, where flow is progressively reduced as individual thermostats are satisfied, it is advantageous to use an additional primary circuit decoupled from the load by a common header or buffer vessel. The use of a header allows flow through the boiler circuit to be unaffected by variations in flow to the load. Circuits connected to loads are operated from the header. The use of reverse return pipework is recommended for the boiler side of the header to ensure equal flows through all boilers. A circuit of this type is shown in Figure 5.2, incorporating a 4-module boiler system and two weather-compensated heating circuits.

Condensing boilers

Condensing boilers differ from others in that they are designed to extract extra heat from the combustion gases by causing condensation of the water vapour in the flue gas. A drain to remove condensate is necessary. However, condensing operation cannot be achieved unless the return water temperature is low, typically below 55 °C; the lower the return temperature, the greater the condensation and the higher the efficiency. The materials of construction must be able to withstand the slightly acidic condensate; stainless steel is frequently used for these heat exchangers. Institution of Gas Engineers publication IGE UP/10 [10] gives detailed advice on the use of stainless steel flues and plastic condensate pipes. The relatively cool combustion gases lack buoyancy and it is usual to have additional fan power to drive them through the flue system. Condensing

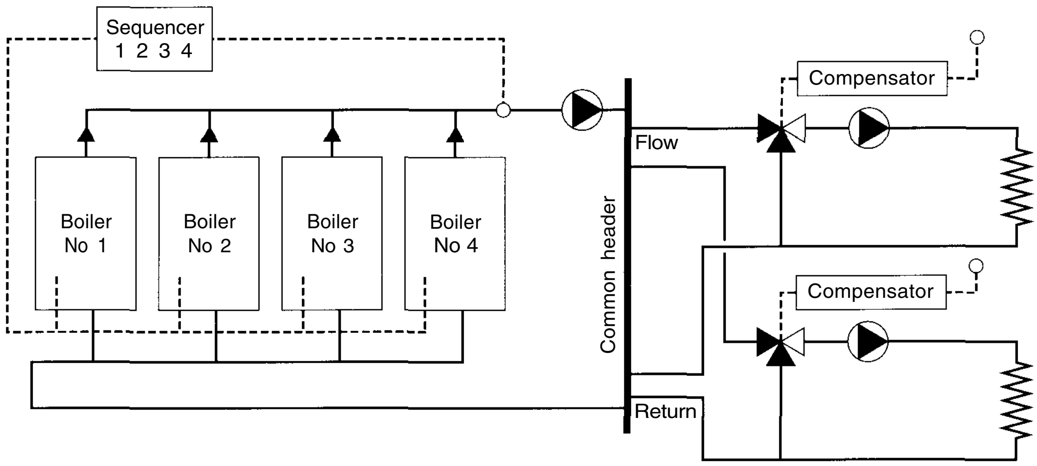


Figure 5.2 Multiple boiler circuit with header and reverse return circulation through boilers (courtesy of Hamworthy Heating Ltd)

boilers should be used only with very low sulphur content fuels.

Low water content boilers

Low water content boilers have compact heat exchangers designed for maximum surface area. Common materials for heat exchangers include aluminium, copper and stainless steel. Both natural and forced draught combustion types are available.

Good water circulation through the heat exchanger is essential during boiler operation and a means of flow sensing is usually required, interlocked with the burner.

Low water content boilers offer rapid heat-up and high efficiency coupled with compact size and light weight. However, life expectancy is usually significantly shorter than for cast iron or steel boilers with larger combustion chambers.

Gas boilers

Gas boilers are available in a large range of types and sizes for use with both natural gas and liquefied petroleum gas (LPG). The properties of both types of gas are described in section 6. Modern appliances are designed and manufactured in compliance with European standards. Under UK gas safety legislation, all new appliances must display a CE mark of conformity; to install appliances not having the CE mark or to modify appliances displaying the mark may be unlawful. Strict requirements for gas safety apply similarly to forced draught and natural draught burners.

Appliance standards deal not only with construction but also cover efficiency and emissions to the atmosphere. However, standards cannot easily cover the quality of the installation, which is the responsibility of competent designers and installers. Guidance on installation is provided in IGE UP/10 [10], which also includes information on ventilation and flues for appliances with a net output above 70 kW.

Gas boilers rely on various different types of burner:

- *Forced draught burners:* typically of the nozzle mix type in which gas and air are separately supplied right up to the burner head, where mixing takes place. The effectiveness of the combustion process relies on the design of the mixing head and the pressure of the air and gas at the head, particularly in achieving low emissions of nitrogen oxides (NO_x) and carbon monoxide (CO). Most burners are made to comply with BS EN 676 [11]. It is rare today to see a burner with a separate pilot since most start at a low fire condition at the main burner. Air proving is essential with a 'no-air' check being made before the fan starts, to check that the proving switch/transistor is operational. The combustion system is normally purged with up to 5 volumes of air in order to remove any traces of gas or remaining products of combustion. The gas safety train to the main burner supply incorporates a low inlet pressure switch, a pressure regulator and two high quality safety shut off valves. Above 1200 kW there is a requirement for either a valve seat condition proving system or a double block and vent valve position proving.

  The turndown range of the burner from high to low depends on the individual manufacturer's designs and the required excess air levels from high to low fire. Many can operate over a range of more than 4 to 1.

  Some larger burners require higher pressures than are available from the gas supply system. In such cases, a gas pressure booster may be required, which is typically provided by a simple centrifugal fan. Overall safety requirements are covered by IGE UP/2 [12]; they include a stainless steel flexible pipe either side of each booster and a pressure switch to cut off the booster at low line pressure.

  It is possible for forced draught burners to operate in dual fuel mode, using an additional nozzle for oil firing. Larger types of dual fuel burner may incorporate a rotary or spinning cup to atomise the oil but many simply rely on high oil pressures at the atomiser.

- *Pre-mix burners:* these differ from forced draught burners principally in that the air for combustion is mixed with the gas before it reaches the burner head. They produce very short intense flames that can work in very compact combustion chambers and, due to lower excess air levels, can achieve higher efficiencies. However, turndown is more restricted than with nozzle mix burners and is typically of the order of 1.5 or 2 to 1 on a single burner head. Larger turndowns are achieved by sequencing burner heads or bars within a single combustion chamber.

- *Natural draught (atmospheric) burners:* these are widely used on gas cookers and small boilers and are often described as 'Bunsen' type. The incoming gas at the injector induces combustion air with which it mixes before reaching the head. The amount of air induced is typically 40 to 50% of what is required and the remainder is drawn in by the combustion process itself. Because of its slow and staged mixing, the flame envelope is larger and requires a larger combustion chamber than forced draught and pre-mix burners. Some boilers of less than 45 kW still use thermo-electric flame safeguards to detect the loss of flame but fully automatic flame rectification and ignition are increasingly becoming standard.

- *Pulse combustion:* air is induced into the combustion system by means of Helmholtz effect. The rapid forward flow of the exploding combustion products within a strong chamber leaves a shock wave behind that induces the gas and air required for the next pulse, which ignites automatically. The cycle continues until the gas supply is turned off. Pulse combustion operates at high pressure and enables very small heat exchangers and flues to be used.

Oil boilers

Burners for oil boilers almost always rely on atomisation, which is carried out mechanically. Oil of various grades is used for firing. Kerosene (Class C2) is commonly used in domestic boilers, gas oil (Class D) is most frequently used in larger heating installations, and fuel oil (Classes E, F and G) is used in some large installations. Guidance on oil boilers may be obtained from OFTEC[13,14] .

- *Pressure jet burners:* most frequently used on smaller boilers, but can operate at outputs up to 4.5 MW. They consist of a fan to provide combustion air and to mix it with atomised droplets of oil produced by a nozzle fed at a high pressure from a fuel pump. Since effective atomisation depends on the flow of oil to the nozzle, the turn down ratio is limited to about 2:l. Modulation is correspondingly restricted and on/off operation is common.

- *Rotary burners:* normally used on larger boilers of the welded shell type, where fuel heavier than Grade D is burned. Atomisation is achieved by centrifugal action as oil is fed to a rotating cup, which throws droplets into an air stream produced by the primary combustion air fan. A secondary combustion air fan enables the burner to operate over a wide turn-down range, which may be up to 5:l. This type of burner can be readily adapted for dual fuel (gas/oil) operation. However, it is relatively noisy in operation and may require sound attenuation measures.

Solid fuel boilers

Solid fuel burners are less flexible in use than those for gaseous or liquid fuels and consideration must be given at an early stage to arrangements for the storage and handling of fuel, the removal of ash and grit, flue gas cleaning and operation and maintenance of the boiler house. Also, it is necessary to design the system to ensure that heat can be safely dissipated when the boiler is shut down or the load sharply reduced.

- *Gravity feed burners:* suitable for use with outputs up to about 500 MW. Their rate of combustion may be controlled by modulating the fan supplying combustion air, giving a good turn-down ratio and a high thermal efficiency.

- *Underfeedstokers:* most commonly used for sectional and fabricated steel boilers operating at outputs up to 1.5 MW. The fuel is supplied through a tube using a screw, regulated to match the requirements of the furnace, and combustion air is controlled by a fan. Fuel types and grades may be restricted.

- *Coking stokers:* used with shell boilers rated at up to 4.5 MW. A ram pushes coal from a hopper into the boiler, where there is partial distillation of the volatile components of the coal. The fuel then travels forward into a moving grate where combustion is completed, relying on induced draught.

- *Chain grate stokers:* used in large shell boilers, with outputs of up to 10 MW. An endless chain grate feeds coal continuously into the boiler furnace, where combustion takes place with either forced or induced air supply.

- *Sprinkler stokers:* an air stream is used to convey coal to a fixed grate in shell boilers with outputs between 600 kW and 8.5 MW.

- *Fluidised bed systems:* these rely on fuel fed into a furnace bed consisting of particles of inert material that are continuously recycled. The mixture is fluidised by a flow of air large enough to hold the fuel in suspension while combustion takes place. This type of combustion is suitable for a wide range of coal types, including poor quality coal. It is well suited to automatic control and may be able to reduce acid gas emissions by the use of additives in the fuel bed.

Boiler selection

The following factors need to be taken into account in selecting a boiler for a particular application:

- output in relation to calculated system requirements, see section 4.7

- efficiency, particularly at part load, see section 4.2.3

- hydraulic pressure at which the boiler must operate

- system operating temperature: it is particularly important that return water be maintained above the minimum recommended by the manufacturer for non-condensing oil-fired boilers to avoid corrosion from acid condensation in the flue system

- flue gas conditions, to comply with emission requirements, see section 5.5

- corrosion and water treatment, taking account of the specific recommendations of the boiler manufacturer

- acoustic considerations, taking account of noise both inside and outside the boiler room

- floor temperature beneath the boiler: the temperature of a concrete floor should not be allowed to exceed 65 °C; this should not occur where the base of the boiler is water cooled, but may otherwise require a refractory hearth under the boiler
- space in the boiler house, especially with regard to access for maintenance
- access for initial installation and subsequent replacement.

5.1.2.2 District or local heat supplies

Where a supply of delivered heat is available, connection to the main may be either direct or indirect, via a heat exchanger. Direct connection is normally used in small heat distribution systems where heat is distributed at temperatures not exceeding 90 °C, e.g. using heat from a CHP unit based on an internal combustion engine.

For indirect connection, the role of the boiler is effectively assumed by a heat exchangery either a non-storage shell and tube calorifier or, more commonly in recent years, a plate heat exchanger. This allows the distribution system within the building to be run at a temperature and pressure suitable for the building rather than for the heating main. The distribution network, controls and heat emitters in the building can effectively be the same as those used with a boiler.

When connecting to a heat distribution system, it is important to design the connection method and the secondary system so that water is returned to the system at as low a temperature as possible. This reduces flow rates and lowers network costs. It is recommended that the heat supply company should be allowed to review and comment on the design of the connection method and the heat distribution. Good Practice Guide 234: *Guide to community heating and CHP*[15], gives detailed guidance.

5.1.2.3 Small-scale combined heat and power (CHP)

Small-scale combined heat and power units may be used to replace part or all of the boiler capacity in buildings with a suitable electricity demand profile. CIBSE Applications Manual AM12 [16] describes the main features of CHP plant and its integration into buildings. The CHP unit is typically used as the lead boiler in a multi-boiler system and sized to minimise life cycle costs, which may involve some dumping of heat. A computer program is available under the government's Energy Efficiency Best practice programme for optimising the capacity of CHP units in certain types of buildings.

CHP systems based on reciprocating engines are available with electrical outputs ranging from 50 kW to 4500 kW. Small installations generally favour systems with spark ignition engines, fuelled by gas, including LPG, biogas and landfill gas, as well as natural gas. Larger installation may use diesel engines, fuelled by either gas or oil, or gas turbines. Gas turbines are favoured particularly when high grade heat is required for steam raising or when it is necessary to produce a high ratio of electricity to heat through operation in combined cycle mode.

Micro-CHP units, based on Stirling engines, are becoming available for installation as replacements for boilers in dwellings. Heat output must be around 10-20 kW to meet the heat load in a typical installation but electrical output is typically restricted to around 1 kW, to maximise the proportion of kW.h generated that can be used within the dwelling.

Heat may be recovered from various sources within CHP units, including the exhaust, the engine and oil cooling circuits and the after cooler. Figure 5.3 shows alternative schemes for heat recovery.

5.1.2.4 Heat pumps

Air source heat pumps may be used to extract heat either from outside air or from ventilation exhaust air. When outside air is used as a heat source, the coefficient of performance tends to decline as the air temperature drops. There can also be problems with icing of the heat exchanger where the outside air is of high humidity, which is frequently the case in the UK. This requires periodic defrosting, which is often achieved by temporary reversals of the heat pump and reduces the coefficient of performance (CoP). Because of these factors, air-to-air heat pumps have a relatively low CoP (in the range of 2.0 to 2.5) when used for heating in a typical UK climate. As CoP declines with outside temperature, it is not economic to size air source heat pumps for the coldest conditions, and they often include electrical resistance coils for supplementary heating.

Ground or water source heat pumps extract heat from the ground or bodies of water, either at ambient temperature or with temperature raised by the outflow of waste heat. They have the advantage over air source heat pumps that their heat source has much greater specific heat capacity and, provided it has sufficient mass, varies much less with outside temperature. Small ground source heat pumps have a seasonal CoP of around 3.5 in a typical UK climate.

The CoP figures given above are for electrically-driven vapour compression cycle heat pumps. Absorption cycle heat pumps have a much lower CoP but have the advantage that they can be powered directly by gas. When used for heating, the CoP obtainable in practice (typically 1.4) still offers a considerable advantage over a boiler. Domestic sized absorption heat pumps are currently being evaluated in field trials in the Netherlands; these are silent in operation and compact enough to be considered as a replacement for a boiler.

Most heat pumps used for heating in commercial buildings in the UK are reversible and can therefore provide cooling in summer at no additional capital cost.

The environmental advantages/disadvantages of heat pumps hinge on their coefficient of performance and the potential CO_2 emission of the fuel used to power them. Gas-fired heat pumps with a relatively low CoP may therefore produce lower CO_2 emissions per unit of useful heat output than electrically driven units. For electricity drawn from the UK grid, a seasonal CoP of around 2 is required to achieve lower emissions than would be obtained from a gas condensing boiler.

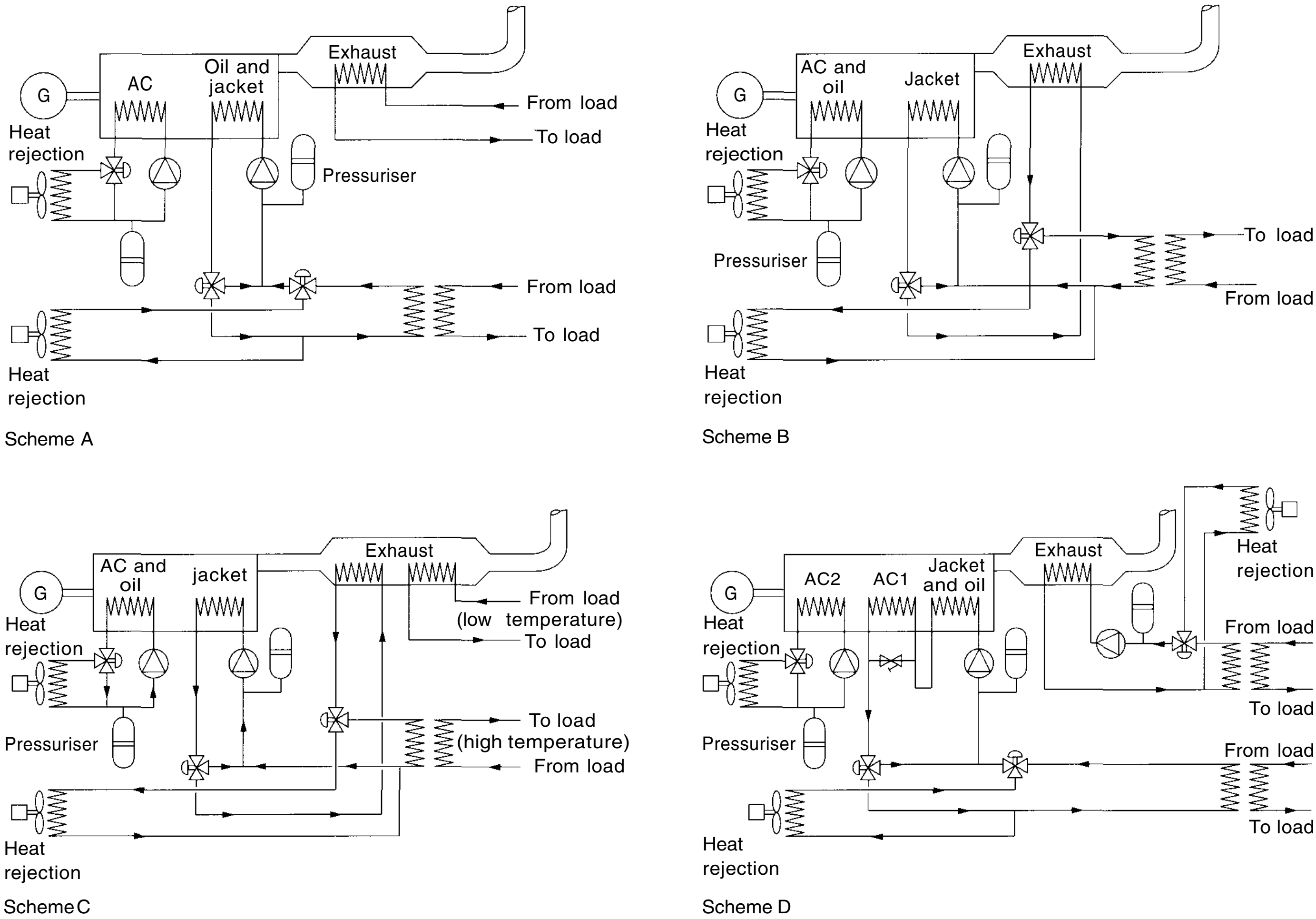


Figure 5.3 Schemes for heat extraction from CHP units[16]

5.1.2.5 Solar water heating panels

Solar water heating panels are widely used around the world to provide domestic hot water, particularly where sunshine is plentiful and fuel is relatively expensive. In the UK, the great majority of installed systems are in dwellings.

The efficiency of solar collector panels depends on a number of factors[17], including the type of collector, the spectral response of the absorbing surface, the extent to which the panel is insulated and the temperature difference between the panel and the ambient air. It is conventional to show collector efficiency against the function:

$$[(t_{f,i} - t_a)/I_f]\ \mathrm{K.m^{2}.W^{-1}}$$

where $t_{f,i}$ and t_a are panel and ambient temperatures, respectively, (°C) and I_f is the intensity of the incident solar radiation (W.m^{-2}). Figure 5.4 shows the efficiency of some types of flat plate collectors in this format. This shows that, in general, the efficiency declines sharply as panel temperature increases above air temperature and that the surface finish of the collector is important. Evacuated tube collectors tend to be no more efficient at low temperature rises but are able to maintain their efficiency at high temperatures.

BS 5918 [18] classifies the performance of solar collectors in terms of the ratio of collector heat loss (W.m^{-2}.K^{-1}) to zero-loss collector efficiency. Typical values of this measure range from greater than 13 for unglazed collectors with no special coating to between 3 and 6 for vacuum insulated panels. The current generation of flat plate collectors with selective coatings generally lie in the range 3 to 5.

A typical solar water heating installation consists of one or more roof mounted panels, a hot water storage cylinder and a means of transferring heat from the panels to the cylinder. Very simple systems, used where sunshine is abundant, rely on gravity circulation but systems designed

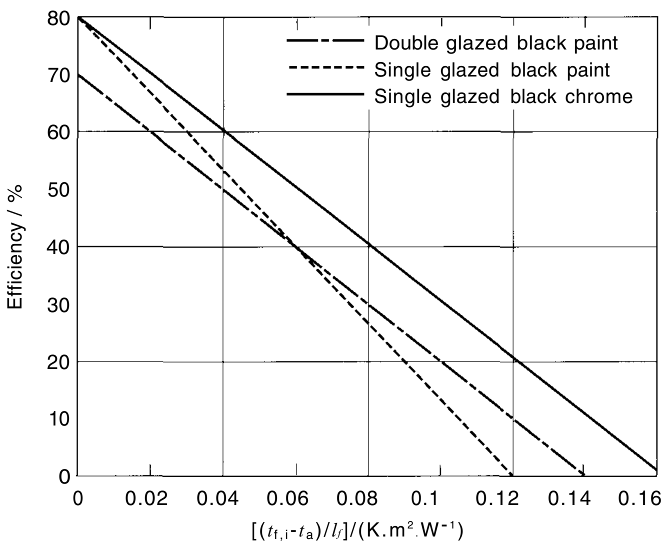


Figure 5.4 Efficiency of typical flat plate solar collectors

for a typical UK climate require a pumped primary circulation. BS 5918 gives guidance for the design and installation of such systems. Some systems used in the UK have separate storage cylinders for solar heated water, which can be kept at an intermediate temperature to maximise the amount of heat collected. Others rely on an additional heating coil in the main hot water cylinder, which is also heated by a central heating system or by an electric immersion heater. The circulation pump is usually controlled by a differential temperature sensor, which causes the pump to operate whenever the temperature of the collector exceeds the temperature of the stored water in the cylinder by a pre-set margin of 2 or 3 °C. Primary circuits often contain a water/glycol solution to avoid freezing.

The energy content of the hot water produced annually per unit area of solar water heating panel depends upon several factors, including the collector efficiency, storage volume and usage patterns. BS 5918 gives a method for sizing solar hot water systems for individual dwellings, taking account of climate, panel orientation and collector performance. It shows that the optimum panel orientation is just west of south but that there is little effect on output within 45° of the optimum. Optimum tilt for the UK is around 33° but there is little difference within ± 15°, which includes most pitched roofs in the UK. Although individual household requirements vary considerably, a rule of thumb is that a house requires 2 to 4 m^2 of panel area, which will yield around a 1000 kW.h per year of heat and meet around half of annual hot water requirements. A set of European Standards dealing with solar heating systems has been developed[19-21] .

Solar panels are also well suited to heating swimming pools. The low temperature required and the very large thermal capacity of the pool water makes it possible to achieve relatively high collector efficiency using simple unglazed panels. Typical installations in the UK (covered by BS 6785[22]) have a panel area of around half of the pool surface area and produce an average temperature rise above ambient air temperature of around 5 K provided the pool is covered at night or indoors.

5.1.3 Pipework

The layout and sizing of pipework for hydronic heating systems is a vital aspect of system design. Once the emitters have been selected and the design flow and return temperatures decided, the circulation requirements in each part of the circuit can be determined. Pipe sizes for individual parts of each circuit may then be selected to give acceptable pressure drops and flow velocities. Consideration should also be given at this stage to the compatibility of emitters connected to particular circuits and to how the system can maintain balance as flow is restricted by control valves.

The designer has considerable flexibility in choosing appropriate pipe sizes. A larger pipe diameter reduces the friction pressure drop and hence the pump power needed to achieve the design circulation. Even a small increase in diameter can have a significant effect, as the pressure drop is approximately proportional to the fifth power of diameter for the same mass flow. An example is given in Appendix A1.3.

Table 5.10 Values of correction factor C for water at different temperatures

Flow velocity /m.s^{-1}	Correction factor for stated water temperature / °C: 40	50	60	70	75
0.2	1.161	1.107	1.060	1.018	1.000
1.0	1.156	1.104	1.058	1.017	1.000
2.0	1.150	1.099	1.055	1.017	1.000
4.0	1.140	1.092	1.051	1.015	1.000

The theoretical basis for calculating pressure drops in pipework is covered in detail in CIBSE Guide C[3] , section 4, which also provides tables giving pressure drop per metre run for a range of pipe sizes and materials. Pipe sizes should ideally be selected to achieve minimum life cycle cost, taking account of both capital cost of pumps and pipework and the running cost to provide the pumping power required. In practice, the starting point for pipe sizing is usually based on flow velocity, ranging from < 1 m.s^{-1} for small bore pipes to 3 m.s^{-1} for pipes with a diameter of greater than 50 mm. The tables in Guide C are banded to show flow velocity. Another approach is to size for a particular pressure drop per unit length, typically between 200 to 300 Pa.m^{-1} .

The tables in Guide C relevant to heating circuits are calculated for temperatures of 75 °C. When using water temperature temperatures lower than 75 °C, the pressure drop will be greater, due mainly to the higher viscosity. Table 5.10 gives the correction factor to be applied to the tabulated data, see equation 5.10. The correction factor does not vary with diameter, though velocity does have a small effect.

$$\Delta p = C \Delta p_{75} \tag{5.11}$$

where Δp is the corrected pressure drop (Pa), C is the correction factor and Δp_{75} is the tabulated pressure drop at 75°C (Pa).

5.1.4 Pumps

5.1.4.1 Pump characteristics

Centrifugal pumps are well suited to providing the necessary circulation in hydronic heating systems. They operate by using the energy imparted by a rotating impeller fitted in a carefully designed casing; liquid enters near the centre of the impeller and leaves at higher velocity at its perimeter. A typical centrifugal pump characteristic is shown in Figure 5.5, in which it may be observed that maximum pressure is produced at zero flow and maximum flow at zero pressure.

Centrifugal pumps have the following characteristics:

- flow varies directly with the speed of rotation of the impeller
- pressure varies as the square of the speed
- power absorbed varies with cube of the speed.

If the diameter of the impeller is changed, but speed of rotation kept constant:

- flow varies as the cube of the impeller diameter

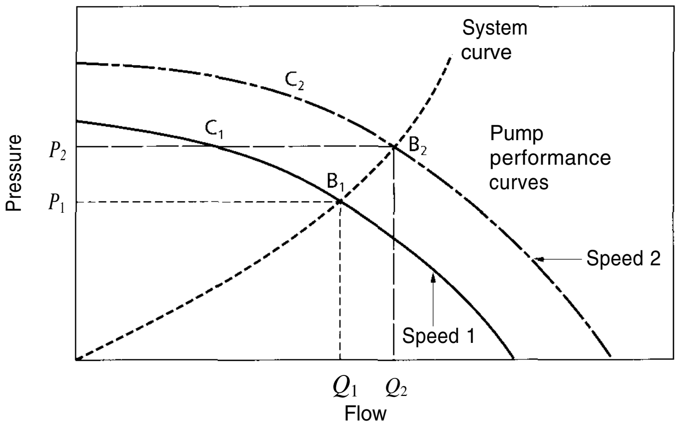


Figure 5.5 Performance curves for a centrifugal pump

- pressure varies as the square of the impeller diameter
- power absorbed varies as the fifth power of the impeller diameter.

The flow available from a centrifugal pump in a circuit depends upon the resistance characteristics of the circuit. Figure 5.5 shows a typical system curve superimposed on the performance curves of the pump. The flow obtained at a given pump speed can be determined from the point at which the pump and system curves intersect. A pump speed is selected which can provide the required flow at the pressure drop around the path of the circuit with the highest pressure drop, otherwise known as the 'index' circuit.

5.1.4.2 Variable speed pumping

Maximum flow and power are only required under design conditions in which all loads are calling for heat. As demand is satisfied, full flow is no longer required in parts of the circuit and pumping power can be reduced to match the system requirement at the time. The most effective method of controlling pump speed is by means of induction motors powered by variable frequency inverters; such a combination can maintain high efficiency over a wide range of speeds. Variable speed motors, which have a built-in inverter drive, are also available. Pump energy savings of 60-70% are possible, with payback times of around 2 years.

The design of variable speed pumping systems needs to allow two-port control valves to close without causing unwanted flow or pressure variations in other parts of the circuit. The most common method of controlling pump speed is to maintain a constant pressure differential between two points in the index circuit. BSRIA Application Guide AG 14/99 (23) describes procedures for the design of systems with variable speed pumping.

5.1.5 Controls

Section 4.3.8 describes the general principles of control and control functions applied to hydronic systems; this section describes the key components required to implement those functions. Control equipment may be broadly subdivided into sensors, actuators and processors, all linked by some form of network.

5.1.5.1 Sensors

Sensors may be broadly grouped into three types:

- *Analogue sensors:* the measured variable to be controlled is converted into a continuously variable signal, usually electrical. The signals produced are often very small and require signal conditioning, amplification or conversion to digital form before they can be fed to a processor or actuator. The signal conditioning may be remote from the sensing element or incorporated within the same unit. Standard signal levels for interfacing with controllers and actuators include 0 to 10 volts and 4 to 20 mA.
- *Status sensors:* an on/off signal is produced, depending on whether the variable is above or below a set point. They are frequently electromechanical devices, where a physical movement causes contacts to open or close. They exhibit hysteresis (a 'dead band') in operation, which must be overcome before they change state.
- *Intelligent sensors:* some element of processing is incorporated, in addition to the basic sensor function.

5.1.5.2 Temperature sensors

Electromechanical thermostats are widely used for room temperature control. Higher accuracy and lower hysteresis may be obtained from resistance thermometers (platinum or nickel) or thermistors. If a fast response is required, thermocouples may be the best choice, particularly for temperature difference measurements. Section 3.1.3 of CIBSE Guide H (24) gives guidance on the selection of temperature sensors.

- *Thermostatic radiator valves:* an effective form of room temperature control when incorporated in a suitable overall control system. They provide autonomous local control by combining a wax-operated thermostat with a directly coupled valve.
- *Programmable room thermostats:* these combine the functions of a programmer (or time switch) and a thermostat and offer the possibility of different set points at different times of day or week; e.g. a lower set-back temperature at night to prevent excessive overnight cooling.

5.1.5.3 Humidity sensors

Humidity sensors are necessary when air conditioning or humidity control is provided in combination with the heating system. Heating in 'heritage' rooms may be controlled to prevent low relative humidity, when preservation takes precedence over visitor comfort. Simple mechanical humidity sensors (or hygrometers) are based on the expansion of a natural material or nylon as water is absorbed from the ambient air but these have poor accuracy and repeatability. More accurate measurements of relative humidity have traditionally been made with the wet and dry bulb hygrometer, but this device does not lend itself well to automatic operation.

For building services applications, the sensors most commonly used for automatic control rely on capacitive polymer film sensors, with the sensing element protected by a membrane or netting filter. The polymer film, which forms the dielectric within the capacitor, responds to relative humidity and affects the measured capacitance. Where accurate measurements are required (e.g., for calibration of other sensors), dewpoint measurements are recommended. Automatic dewpoint sensors are available but are too expensive for routine application.

5.1.5.4 Flow sensors

Flow measurement may be required during the commissioning of a heating system. Turbine flow meters and orifice plates may be used for this purpose provided suitable metering points have been incorporated in the system pipework. Where that is not the case, it may be possible to make non-intrusive measurements using ultrasonic flow meters. Flow switches, which give a status signal to indicate that flow is taking place, may also be used as part of a control system.

Balancing valves (Y-pattern), which incorporate a close-coupled orifice and a pair of pressure tappings, are available ready calibrated for use during commissioning. Unfortunately, the permanent inclusion of an orifice plate incurs an unnecessary permanent pressure drop and increased pumping power over the life of the installation.

5.1.5.5 Pressure sensors

The maintenance of design system pressure is important both for proper function and safe operation in hydronic heating systems, particularly in sealed systems and those operating at high pressure. Simple Bourdon gauges may be used to give a visual display and various pressure transducers are available for supplying signals to automatic control systems. Pressure operated switches are also available.

5.1.5.6 Actuators

Actuators are principally required in hydronic heating systems to operate valves that control the flow to various circuits and emitters. Apart from the built-in thermic actuators used in thermostatic control valves, electric motor and solenoid actuators are widely used with valves. The actuator may be fully modulating, where the position of the actuator is proportional to the control signal, or multi-state, where it can assume two or more fixed positions in response to a signal. Actuators often include a positional feedback signal, which may be analogue or status.

Pneumatic actuators were once very widely used in HVAC systems but have been largely supplanted by electrical types. However, they are often still used to operate large valves and in high pressure applications that require a motor with high torque. Where existing pneumatic systems are being upgraded to electronic control, it may be possible to retain pneumatic operation of actuators by using hybrid electro-pneumatic transducers.

5.1.5.7 Valves

Various valve types are shown in Figure 5.6. Two- and three-port valves are commonly used in heating circuits. Three-port valves may be of the mixing type, which has two inlet ports and one outlet, or the diverting type, which has one inlet port and two outlets. Correct application of three-port valves requires that account be taken of the direction of flow through each port. Small four-port valves are often used on fan coil units. However, they are essentially three-port valves with a built in bypass, which has flow and return connections into the valve and flow and return connections out.

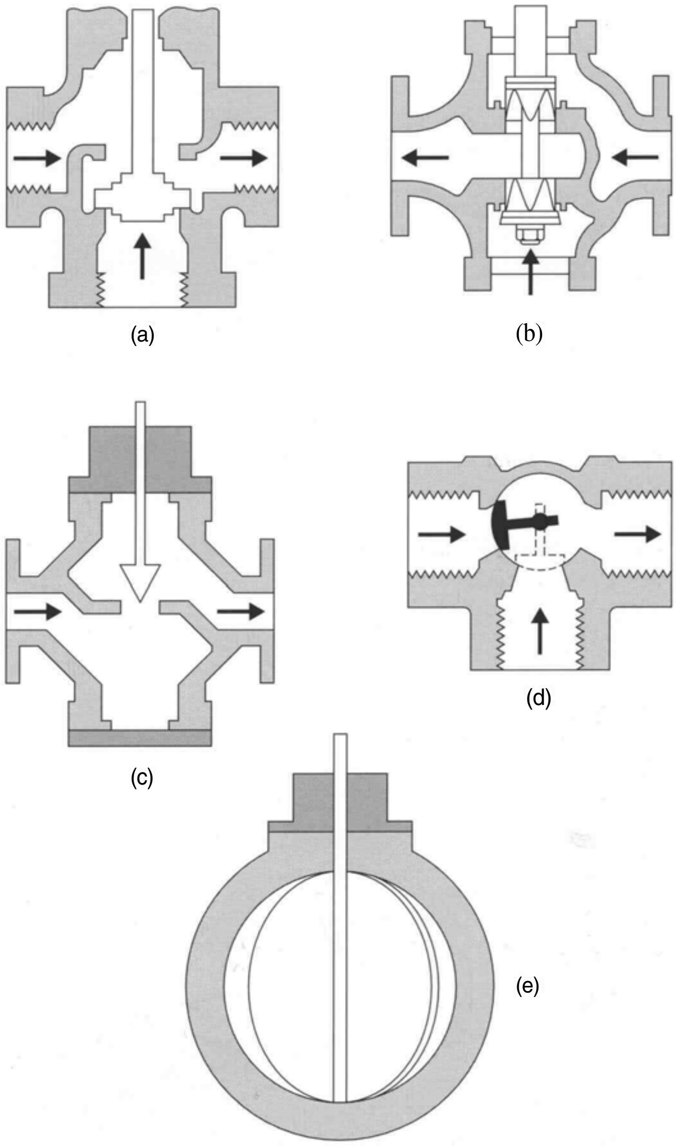


Figure 5.6 Common valve types: (a) plug and seat three-port mixing valve, single seat; (b) plug and seat three-port mixing valve, double seat; (c) two-part plug and seat valve; (d) rotary shoe valve; (e) butterfly valve

Valve characteristic curves, which relate flow to actuator position, are very important to successful control system design. Figure 5.7 shows some typical valve characteristics. The way in which valve stem position can influence flow through the circuit it controls depends not only on the valve characteristic but also on how the pressure drop across the valve compares with that around the rest of the circuit. If the valve is too large, the resistance to flow will be dominated by the rest of the circuit except when the valve is near to its closed position and therefore the valve will have little effect over much of its range. If the valve is too small, it will cause a large pressure drop and require additional pumping pressure under normal operating conditions.

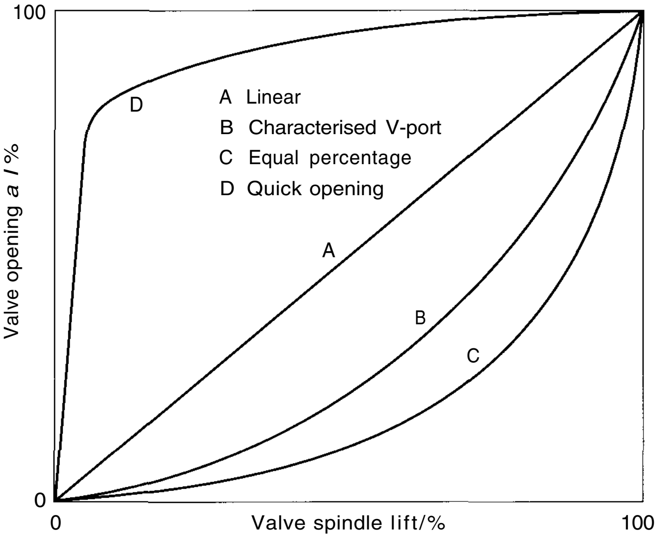


Figure 5.7 Typical valve characteristics

Valve performance is usually defined in terms of flow capacity *K,* defined by the relationship:

$$K = q_v / \sqrt{\Delta p} \qquad (5.12)$$

where K is the flow capacity ($m^3 \cdot h^{-1} \cdot bar^{-0.5}$), q_v is the volumetric flow rate ($m^3 \cdot h^{-1}$) and Δp is the pressure drop (bar).

The action of the valve in intermediate positions is best characterised in terms of relative valve capacity:

$$\alpha = (K_v / K_o) \times 100 \qquad (5.13)$$

where a is the percentage opening of the valve (%), K_v is the flow capacity for a particular valve position and K_o is the flow capacity in the fully open position.

The relationship between valve and circuit pressure drop is expressed in terms of valve authority, N_{des}, which is defined as:

$$N_{des} = \Delta p_{vo} / (\Delta p_{vo} + \Delta p_c) \qquad (5.14)$$

where N_{des} is the valve authority, Δp_{vo} is the pressure drop across the valve in fully open position at design flow (Pa) and Δp_c is the pressure drop across the rest of the circuit (Pa).

Valve selection is covered in detail in section 3.3.4 of CIBSE Guide H [24].

Compound valves, incorporating measurement and/or regulating functions, are available to assist commissioning and control. These include the following:

- *double regulating valves:* incorporate a device that allows a pre-set position to be retained while also providing an isolating function
- *variable orifice double regulating valves*: include tappings that allow the pressure drop across the valve to be measured
- *fixed orifice double regulating valves:* incorporate an orifice plate with tappings to enable flow measurement; this type is also known as a 'commissioning set'
- *constant flow regulators:* automatically control flow rate provided the differential pressure across the valve is maintained within certain limits
- *differential pressure control valves:* maintain a constant pressure across a branch of a circuit.

Both constant flow regulators and differential pressure control valves may be used to assist in the commissioning of circuits.

5.2 Equipment for steam systems

5.2.1 Boilers

A steam boiler differs from a water circuit boiler in that it produces a phase change from water to steam, which introduces additional requirements for the control of both the pressure and the water level within the boiler. Also, as a pressurised vessel containing water and steam at above 100 °C, it requires greater attention to the maintenance of safety in operation.

Steam boilers may be broadly classified into two types:

- *shell (or fire tube) boilers*
- *water tube boilers.*

Shell boilers operate by passing heated gases through tubes in the boiler. Figures 5.8 and 5.9 show typical shell boiler configurations for 'two-pass' operation, in which the heated gases from the furnace are reversed to flow through the boiler for a second pass to extract more heat. In the 'dry back' configuration, the flow is reversed by a refractory lined chamber; in the 'wet back' version the reversal chamber is contained entirely within the boiler, which improves the efficiency of heat transfer. Modern packaged boilers commonly use three passes to achieve high efficiency and compact dimensions. Shell boilers are covered by BS 2790[25].

Water tube boilers differ from shell boilers in that the heat source surrounds tubes circulating the boiler water, see Figure 5.10. They are able to operate at higher pressures than shell boilers because the tube diameters are much

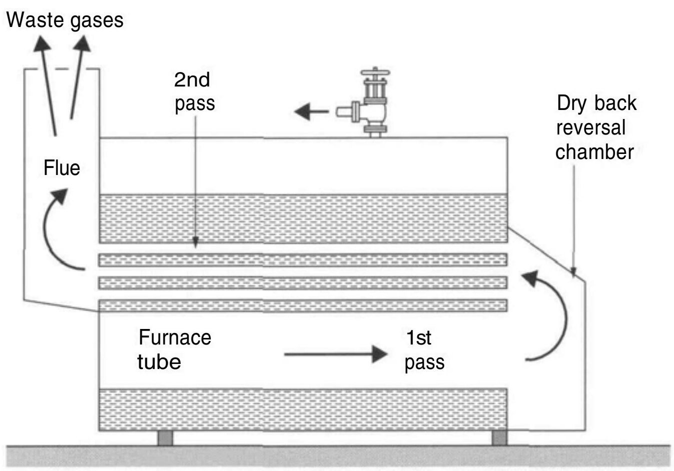


Figure 5.8 Two-pass dry back shell boiler (courtesy of Spirax-Sarco Ltd)

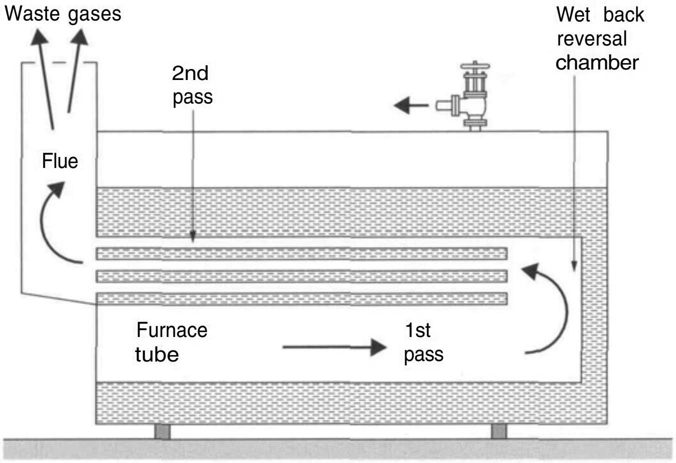


Figure 5.9 Two-pass wet back shell boiler (courtesy of Spirax-Sarco Ltd)

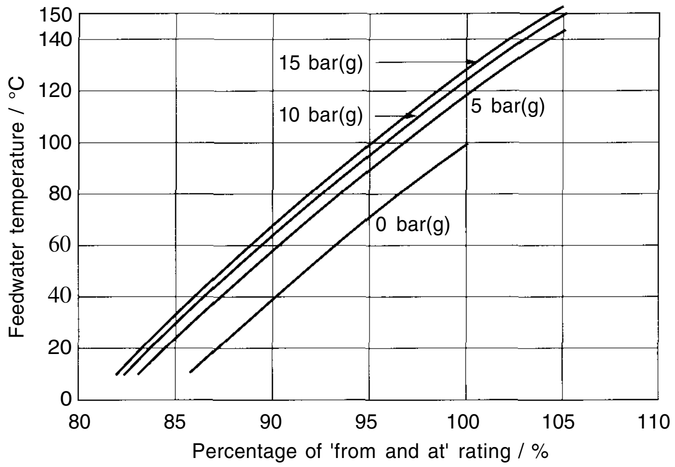


Figure 5.11 'From and at' variation with temperature (courtesy of Spirax-Sarco Ltd)

lower than those of the shell of a shell boiler, with corresponding reductions in the hoop stress. As shell boilers are limited in practice to pressures below 27 bar (gauge), or a steam temperature of 340 °C, water tube types tend to be used for applications requiring high pressure, high temperature or very large steam output. Water tube boilers are available in smaller sizes but offer no advantage over shell boilers for most commercial and industrial applications involving heating. Water tube boilers are covered by BS 1113 [26].

Steam boiler output depends on operating conditions and is rated in three ways:

- 'from and at' rating
- kW rating
- boiler horse power (BoHP).

The 'from and at' rating is based on the amount of steam (in kg) at 100 °C and atmospheric pressure that the boiler

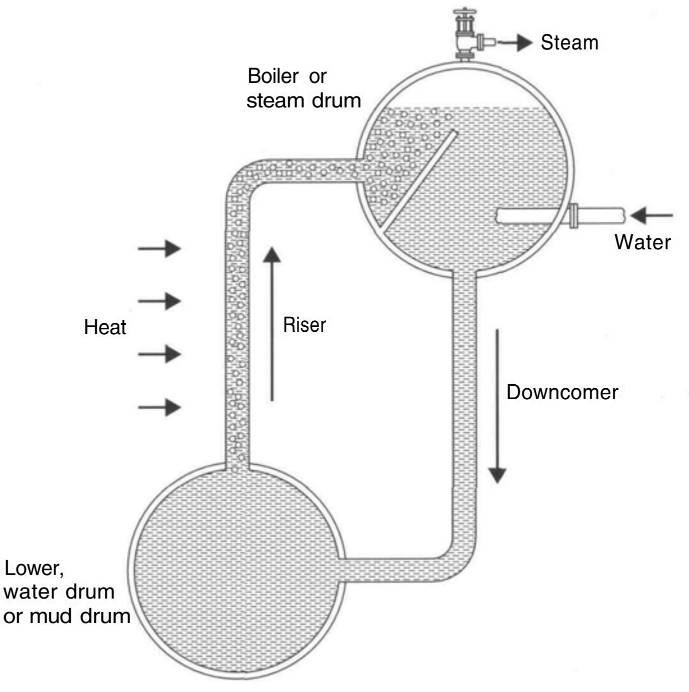


Figure 5.10 Riser-downcomer configuration of water tube steam boiler (courtesy of Spirax-Sarco Ltd)

can generate in 1 hour from a feed water temperature of 100 °C. Under these conditions, each kilogram of water requires 2258 kJ of heat per hour to convert it to steam, which is equivalent to 627 W. In practice, boilers are operated under a range of conditions and the steam output under different conditions may be calculated using steam tables or estimated from the graph shown in Figure 5.11. 'From and at' ratings are widely used by manufacturers of shell boilers.

Some manufacturers give boiler ratings in kW. Steam output may be calculated from the difference between the specific enthalpy of the feed-water and the steam at the required pressure.

'Boiler horse power' tends to be used only in the USA, Australia and New Zealand and should not be confused with the imperial unit of power, which is approximately 746 W. In Australia and the USA, 1 BOHP is defined as the power required to evaporate 34.5 pounds of water per hour at 212 °F at atmospheric pressure. This is essentially the same form of definition as the 'from and at' rating, except that it is based on 34.5 lb instead on 1 kg; 1 BOHP is equivalent to 15.51 kg/hour. In New Zealand, BOHP is defined in terms of the heat transfer area of the boiler.

Boiler efficiency depends upon the design of the boiler and the conditions under which it is operated. Some boilers incorporate an 'economiser', which consists of an additional heat exchanger using exhaust gases to preheat the feed-water before it is returned to the boiler. However, economisers may not be used on boilers with on-off level controls. Efficiency in steam systems also depends on minimising heat losses from the boiler feed-tank, which should be well insulated to prevent heat losses.

Steam boilers must be fitted with appropriate safety devices. In the UK, these are currently covered by BS 759: Part 1 [27]. Each boiler must have a name plate, with a serial number and model number which uniquely identifies it and its manufacturer and gives details of various tests to which it has been subjected. It must also be fitted with a safety valve to protect it from overpressure and the risk of explosion; in the UK, BS 6759 [28] covers safety valves for steam boilers. Safety valves are also covered by section 8 of BS 2790 [25], which relates to the design and manufacture of shell boilers of welded construction.

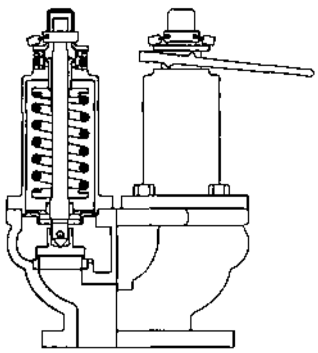


Figure 5.12 Typical steam boiler safety valve (courtesy of Spirax-Sarco Ltd)

A typical safety valve is shown in Figure 5.12. Safety valves must be capable of discharging the full 'from and at 100 °C' capacity of the boiler within 110% of the design boiler pressure and be set at no higher than the design pressure. At least one safety valve is required for all boilers; boilers rated at more than 3700 $kg.h^{-1}$ are required to have two single safety valves or one double safety valve. The discharge pipe from the safety valve must have no obstructions and be drained at the base to ensure that condensate cannot accumulate. Each boiler must also be fitted with a stop valve (or crown valve) to isolate it from the plant it serves. This should always be fully open or fully closed, and should not be used as a throttling valve.

Other safety equipment required by steam boilers includes:

- a feed check valve to prevent return flow from the boiler when the feed pump is not operating and flooding from the static head in the feed tank
- a bottom blow-down valve, which may be manual or automatic in operation
- a pressure indicator, which may be a simple Bourdon gauge with a dial of least 150 mm in diameter
- a gauge glass to show the level of water in the boiler (see Figure 5.13). In the UK, gauge glasses should comply with BS 3463 [29].

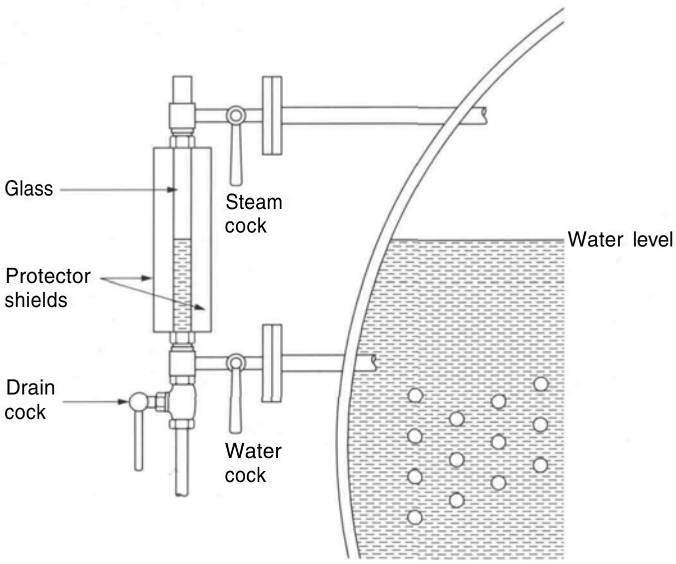


Figure 5.13 Gauge glass and fittings (courtesy of Spirax-Sarco Ltd)

5.2.2 Steam traps

Steam traps are used to drain condensate automatically from the system while preventing the escape of steam. They operate according to three main principles, as follows:

- *thermostatic steam traps:* operate in response to change in temperature and open when condensate temperature falls below a pre-set threshold; they are available in various types suited to particular applications
- *mechanical or balanced pressure steam traps:* operate by sensing the difference in density between steam and condensate; they include 'ball float' and 'inverted bucket' types, which both operate by simple mechanical means
- *thermodynamic steam traps:* these are operated in part by the formation of flash steam from condensate; hot condensate released under pressure closes the trap when it evaporates.

The choice of steam traps for particular applications involves a number of considerations, including air venting, condensate removal (either continuous or intermittent), capacity, thermal efficiency and reliability. The avoidance of water hammer may also depend upon the selection and positioning of traps, as the presence of water hammer may cause traps to fail. Dirt is another factor to be considered in trap selection; traps that operate intermittently with a blast action are less susceptible to dirt than those that depend on small orifices for their operation. Table 5.11 shows a range of steam traps, together with typical applications.

5.2.3 Air vents

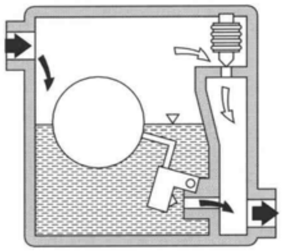
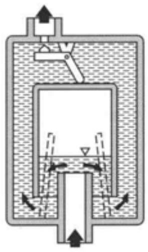
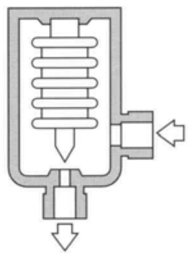
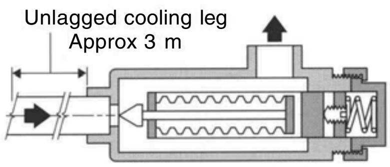
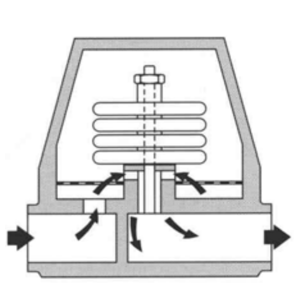
Steam traps are capable of venting air from steam systems but separate air vents are fitted in certain situations, particularly at the end of a steam main. An automatic air vent typically consists of a thermostatically operated valve, see Table 5.11, items 7 and 8. It is best installed at a location where the temperature is low enough for steam to have condensed before reaching it, but where condensation does not collect. In practice this is typically at the top end of a 300 mm length of pipe arranged as a 'collecting bottle', which is left unlagged.

5.2.4 Feedwater equipment

A typical feedwater system is shown in Figure 5.14. The feed tank receives condensate returned from the system and treated water as required to make up losses from the system. The feed pump takes water from the feed tank and supplies it into the boiler at the rate required to maintain the water level in the boiler.

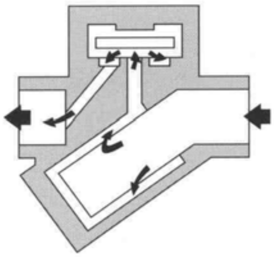
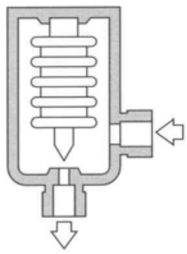
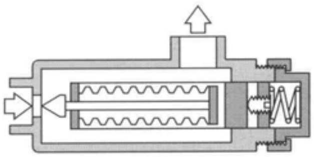
The treatment of make-up water is vital to the longevity, safe operation and efficiency of the system. In particular, it seeks to avoid scaling, corrosion and caustic embrittlement in boilers by removing dissolved and suspended solids and dissolved gases thereby keeping the pH value of the water within defined limits. Water softening, to remove scale-producing ions, may be carried out using (in

Table 5.11 Characteristics of steam traps and air vents

Type	Schematic	Notes
Float trap		*Advantages:* suitable for widely fluctuating loads and pressures; easy to install and maintain; removes condensate continuously as it forms; types with balanced pressure air vents automatically discharge air. *Disadvantages:* can be damaged by water-hammer and corrosive condensate; normally three or four differently sized valves and seats are required to cover the normal working range.
Inverted bucket trap		*Advantages:* can be made for high pressure and superheated steam; will withstand water-hammer; can be made of corrosion resisting materials; a check valve should be fitted at the inlet where used with superheated steam; working parts are simple. *Disadvantages:* wasteful of steam if oversized; does not respond well to severe fluctuations of pressure and discharges air slowly; a thermostatic air vent fitted in a by-pass is recommended; should be lagged when used outdoors. *Notes:* no longer manufactured but some may still be found in service; open top bucket traps have similar advantages and disadvantages.
Thermostatic steam trap		*Advantages:* compact; automatically discharges air; valve is wide open on start-up, so cool condensate and air discharge quickly; capacity is high; unlikely to freeze if condensate can run from trap outlet; maintenance is easy;. traditional elements have corrugated brass or phosphor bronze bellows, newer designs have a stainless steel bellows **or** diaphragm-type element. *Disadvantages:* older type elements liable to damage by water hammer, corrosive condensate or superheated steam (stainless steel elements are more robust and some designs are suitable for use with superheated steam).
Liquid expansion steam trap		*Advantages:* can be used with superheated steam and at higher pressures than balanced pressure traps; valve is wide open on start-up, ***so*** cool condensate and air discharge quickly; capacity is high; operates by continuous discharge, ***so*** quiet in operation and unaffected by vibration, steam pulsation and water-hammer; automatically discharges air. *Disadvantages:* does not respond quickly to change in load or steam pressure; element can be damaged by corrosive condensate. *Note:* because element is on discharge side of valve orifice, trap will hold back condensate. This permits use of some sensible heat from condensate provided that water-logging of steam space is acceptable; if this is not the case, a cooling leg must be fitted before the trap.
Bi-metallic steam trap		*Advantages:* usually small and robust; when cold valve is wide open and air is freely discharged; capacity is greatest when condensate is coolest; some types are not damaged by freezing; withstands water hammer and some are unaffected by corrosive condensate; suitable for use on high pressure and superheated steam; will work over wide range of pressures without need to change size of valve orifice, although position of orifice may need to be adjusted; holds back condensate until cooling occurs thus using some of the sensible heat. *Disadvantages:* will not discharge condensate until it has cooled below saturation temperature, so unsuitable for use where condensate must be cleared as soon as it forms unless a cooling leg is provided; responds slowly to changes in steam pressure and condensate load.

Continued on next page

Table 5.11 Characteristics of steam traps and air vents - *continued*

Type	Schematic	Notes
Thermodynamic steam trap		*Advantages:* very compact but has large discharge capacity; will work over full range of pressures without adjustment; can be used with superheated steam and can withstand vibration or severe water-hammer; normally made of stainless steel and therefore can withstand corrosive condensate and is not damaged by being frozen. *Disadvantages:* normally requires a minimum pressure differential in order to function; on starting up, if pressure at trap builds up slowly it can discharge a lot of air, but if pressure builds up quickly the resulting high velocity air can shut the trap in the same way as steam and it will air bind; operation of trap can be noisy; due to blast, discharge operation sight glasses and check valves should be fitted about 1 metre from the trap.
Balanced pressure air vent		Similar to balanced pressure steam trap. Valve is wide open when plant is cold; as temperature surrounding the element approaches steam temperature the internal liquid expands thereby generating a pressure within the element which closes the valve seat.
Liquid expansion air vent		Similar to liquid expansion steam trap. Changes in temperature cause the oil filled element to expand or contract causing the valve to move towards or away from its seat.

Note: where water-hammer is present or the steam is superheated, the liquid expansion air vent is the better choice since either of these conditions may damage balanced pressure units. Both the liquid expansion and balanced pressure vents are suitable for any pressure within their range without changing the valve seat, but if conditions vary greatly liquid expansion units may require re-setting

ascending order of effectiveness) base-exchange methods, de-alkalisation, or de-mineralisation.

Feed tanks are made from various materials, including cast iron, carbon steel and austenitic stainless steel.

5.2.5 Space heating equipment

Space heating by steam often uses a heat exchanger to transfer heat from the steam to a secondary hot water circuit, which uses standard hydronic heating equipment.

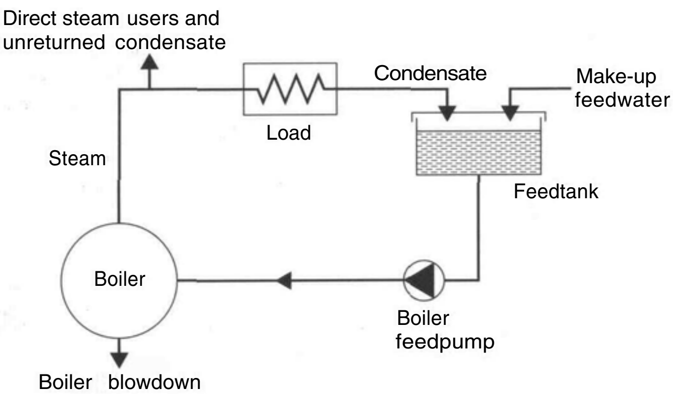


Figure 5.14 Feed-water loop in a steam circuit (courtesy of Spirax-Sarco Ltd)

Figure 5.15 shows a heat exchanger, controlled to maintain a constant secondary flow temperature. Alternatively, it is possible to use a range of emitters powered directly by steam including radiators, natural convectors, fan coil units and radiant panels. Steam radiators usually operate from steam at 0.33 bar with vacuum condensate removal. Vacuum condensate pumping can be problematical and maintenance costs high.

5.3 Equipment for warm air systems

5.3.1 Heat sources

5.3.1.1 Suspended unit heaters

These are small independent gas-fired heaters, with outputs up to 100 kW, typically comprising a burner and heat exchanger inside a painted steel casing, see Figure 5.16. A low powered axial fan blows recirculated air horizontally across the heat exchanger and directly into the heated space. The basic form uses an atmospheric gas burner, usually of the ladder type, firing into a simple pressed steel heat exchanger, which is aluminised or similarly treated to provide corrosion protection. The

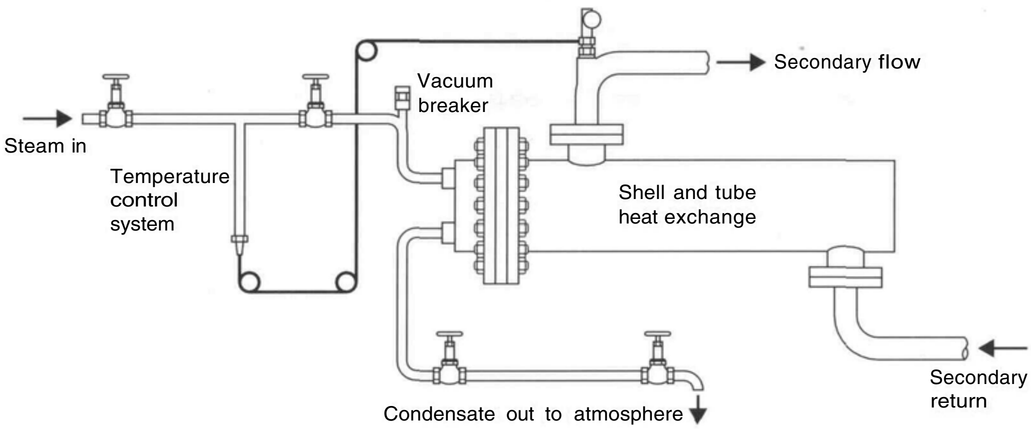


Figure 5.15 Steam-to-water heat exchanger for a hydronic space heating system (courtesy of Spirax-Sarco Ltd)

degree of modulation possible is limited by the need to avoid condensation in the heat exchanger and flue; 60% of full output is the normal minimum output. Flues are usually single skin stainless steel terminating with a cowl at least 1 m above the roof. A draught diverter is usually built into the heater itself.

Variations on this basic design include:

- stainless steel heat exchangers for use in aggressive environments or with fresh air inlet
- room sealed units with induced draft fan and ducted combustion air inlet
- condensing burners
- on/off, two stage or modulating control
- centrifugal fans, for use with air distribution ducting.

Heaters are normally mounted at heights between 2.5 m and 3.5 m above floor level, but higher mounting is possible. Only limited distribution ducting is possible due to the low available fan discharge pressure.

5.3.1.2 Cabinet heaters

These are larger (up to 300-400 kW output) individual heaters, normally gas- or oil-fired, and used in industrial premises where quiet operation and close environmental control are not essential. They are usually floor mounted, but some versions are suitable for high level mounting.

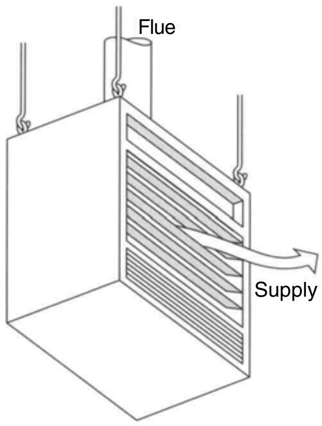


Figure 5.16 Suspended warm air heater (reproduced from EEBPP Good Practice Guide GPG303 by permission of the Energy Efficiency Best Practice Programme)

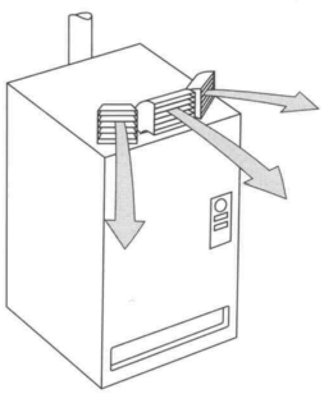


Figure 5.17 Floor-standing cabinet heater (reproduced from EEBPP Good Practice Guide GPG303 by permission of the Energy Efficiency Best Practice Programme)

A typical unit comprises an externally mounted forced-draught burner firing into a steel combustion chamber, with flue gases passing through a tubular heat exchanger before exiting through the flue, see Figure 5.17. Some low cost designs use atmospheric burners. Stainless steel or a protective coating may be used to increase longevity.

A centrifugal fan in the base of the heater blows air across the heat exchanger and the heated air is discharged horizontally through discharge louvres on the top. Alternatively, air may be discharged through distribution ductwork, although the limited fan pressure available on some heaters can mean that extensive ducting is impractical. Inlet air is usually recirculated room air but some heaters can have a ducted inlet for combustion air and/or ventilation air. Flues are usually single-skin stainless steel terminating with a cowl at least 1 m above the roof.

Condensing gas-fired cabinet heaters include an additional stainless steel heat exchanger to cool the flue gases to condensing point. They are relatively uncommon as the high efficiency of non-condensing heaters makes it difficult to justify the extra cost of the additional heat exchanger.

5.3.1.3 Direct fired heaters

These flue-less gas fired heaters are usually of the cabinet type. The gas is burnt directly in the main ventilation airstream (with no heat exchanger) and the products of combustion are therefore distributed into the heated space. A 'cheese grater' burner configuration is usual, with a perforated stainless steel V-shaped shroud around the burner tube. Modulating control of heat output is usually provided. Control of combustion and ventilation is critical to ensure that sufficient dilution of the combustion gases is achieved. Guidance on combustion and ventilation air is given in BS 6230[30]. Direct fired heaters are always used with a fresh air inlet, but some recirculation may be permissible.

5.3.1.4 Air handling units

Although air handling units can incorporate any of the above heater types, heating is more commonly provided by a hot water heater battery supplied from a boiler system or occasionally by direct electrical heating elements. In industrial premises, small individual fresh air supply units with recirculation facilities and water or steam heater

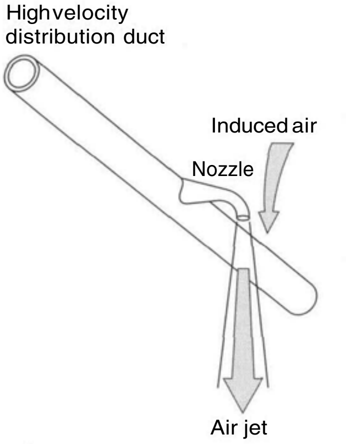


Figure 5.18 Induced jet warm air heating (reproduced from EEBPP Good Practice Guide GPG303 by permission of the Energy Efficiency Best Practice Programme)

batteries may be used. These are most commonly used when hot water or steam is readily available due to process usage.

5.3.1.5 Induced jet systems

Induced jet systems distribute warm air via high velocity ducts, see Figure 5.18. The heat source for these systems is generally a large central gas- or oil-fired unit, built in the style of a specialised air handling unit or cabinet heater. Direct fired heaters must always have the burner situated in the fresh air supply, but some re-circulation is permissible providing it is introduced downstream of the burner, see BS 6230[30].

5.3.2 Ductwork and diffusers

Heating systems involving comprehensive ducting are usually combined with ventilation systems and are therefore also covered in CIBSE Guide B2[31].

Where individual heaters as described in section 5.3.1 are used, duct systems should be limited to provide air distribution through a single space. It is difficult to duct heat to a production area and its associated offices successfully from a single heater.

Duct systems for induced jet heating are usually circular in cross-section and installed at high level in the roof space. Purpose designed nozzles and induction hoods are used to provide the necessary induction and throw, normally producing high duct velocity.

Diffusers are considered in Guide B2[31], section 5.13. The characteristics of various types of air terminal devices are described, including information on typical face velocities and noise levels. Diffusers may be radial, part radial or linear and normally utilise the Coanda effect and/or swirl to avoid excessive room air movement.

5.3.3 Heat recovery

Mechanical ventilation systems, including those that incorporate heating, offer the opportunity to recover heat from air returned by the ventilation system. In energy terms alone, recirculation of air is the most efficient form of heat recovery since it involves little or no energy penalty, but must be limited by the need to maintain an adequate supply of fresh air. Various types of equipment are available for heat recovery, which can both reduce heat requirements in winter and cooling requirements in summer. Heat recovery devices are described in detail in CIBSE Guide B2[31], section 5.6.

5.3.4 Heat distribution combined with fresh air provision

Systems that combine ventilation, heating and cooling are considered in CIBSE Guide B2[31].

It is important to ensure that as much care is given to the successful distribution of heated air as is given to the distribution of ventilation air, since airflow characteristics and circulation patterns will differ between modes.

5.3.5 Heating combined with air conditioning

Buildings with central air conditioning systems normally include provisions for heating, cooling and ventilation. There are a number of different types of systems, including the following:

- *Dual duct systems:* two separate ducts are employed to circulate cooled and heated air to zonal mixing boxes. Thermostatic controls in each zone ensure that air from the hot and cold ducts are mixed in appropriate proportions to achieve the required conditions in the zone. Mixing two air streams to produce an intermediate comfort temperature wastes heating and cooling energy, particularly in constant volume systems.
- *Variable air volume (VAV) systems:* these offer significantly improved energy efficiency compared with constant volume systems, although both systems represent a significant energy cost.
- *Fan coil systems:* these systems heat or cool air in the heated space using coils fed by heated or chilled water, which is distributed by conventional hydronic circuits. A fan coil is a packaged assembly comprising coils(s), condensate tray, circulating fan and filter. 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.

  (a) *Two-pipe non-changeover systems:* 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.

  (b) *Four pipe systems:* separate heating and cooling coils are incorporated, fed by heating and chilled water circuits respectively.

These systems are covered in CIBSE Guide B2[31].

5.3.6 Controls for warm air systems

Control strategies for warm air systems can be kept reasonably simple. Where individual heaters are used it is usually sufficient to provide time control by time-switch or optimiser with on/off temperature control using an air temperature sensor or thermostat. Manufacturers generally offer these simple controls as part of their equipment. BMS or other centralised control can be used but is often not considered necessary.

It is normal practice for each heater to have a dedicated room thermostat to provide individual control. When small output heaters are used it is sometimes possible to control more than one heater from a thermostat but four is considered to be the practical maximum. Averaging from several sensors is not normally used except for central systems since control zones are rarely large enough to justify averaging.

Most individual heaters incorporate a fan run-on circuit, so that the main fan continues to run in order to cool the heat exchanger after' the burner has been switched off (for energy efficiency and to reduce heat exchanger stress) until a pre-set low-limit leaving air temperature is reached.

Two-stage burner control may be used to provide finer control of temperature. The heater fan may continue to be run at full speed or run at slow speed at the low burner output condition. For gas- or oil-fired heaters the low fire output is typically 60% of full output.

The best control of room temperature is obtained using modulating control of the heater output. This can be provided on most forms of warm air heater, but the turn-down ratio is severely limited on indirect gas- or oil-fired heaters. Modulation can be used to maintain a constant room temperature or a constant leaving air temperature. The latter is usually used when the warm air is providing a tempered make-up air supply rather than full space heating. A low-limit control is usually required to prevent the modulating control from reducing the leaving air temperature to such a level as to cause discomfort.

De-stratification systems should be controlled to prevent build up of unacceptable temperature gradients. For low velocity systems the fans should be controlled to run during the full heating period (often from the heater time control). For high velocity systems thermostatic control is preferable to avoid cool drafts.

CIBSE Guide H [24] provides more detailed information on control systems.

5.3.7 Other standards and guidance

There are a number of other standards relevant to warm air heating; these include:

- BS5864[32] andBSEN1319(33)fordomesticsystems
- BS 5991[34] for indirect gas-fired industrial systems
- BS 5990[35] for direct gas-fired industrial installations.

5.4 Radiant heaters

This section deals with equipment used to provide heating with a high proportion of radiant output and good directional properties, characteristics that make radiant heaters suitable for heating areas within larger open spaces.

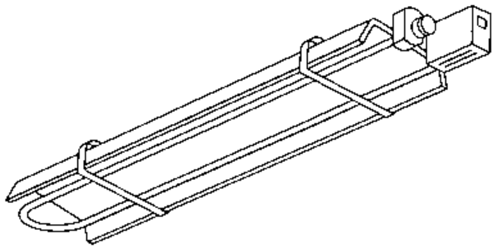


Figure 5.19 Gas-fired overhead radiant tube heater (reproduced from EEBPP Good Practice Guide GPG303 by permission of the Energy Efficiency Best Practice Programme)

In practice, this restricts it mostly to gas and electric radiant heaters that operate at relatively high temperatures. Steam and high-pressure hot water tubes and radiators may also be used, as can air-heated radiant tubes. All are capable of operating temperatures up to 150 °C but their outputs are less readily directed and have a lower radiant percentage than the gas and electric types described below.

5.4.1 Gas-fired radiant heaters

Gas-fired radiant heaters are typically of two types: radiant overhead tube heaters and radiant plaque heaters. Radiant tube heaters may be either flued or un-flued. Radiant plaques are un-flued and offer very high efficiencies and are well-suited to spot heating. The relevant British Standard is BS 6896[36] .

5.4.1.1 Radiant tube

Figure 5.19 shows a typical overhead radiant tube heater. Radiant tube heaters are available in several configurations: U-tube (as shown), linear and continuous (multi-burner). Outputs from individual units are typically in the range of 10 to 40 kW and up to 180 kW can be obtained from multi-tube or continuous tube assemblies, operating at a temperature of around 500 °C. They may be mounted at heights between 3.5 and 20 metres and are mostly used for general area heating, rather than local spot heating. Low-level mounting is avoided to ensure even distribution of heat and to minimise the effects of noise. Reflectors are usually made of polished stainless steel or rigid aluminium, shaped for optimum heat distribution. Tubes are usually steel, often blackened for maximum efficiency. Stainless steel may be used for the first section of tube from the burner, particularly with high output burners. Minimum ventilation requirements for un-flued heaters are given in BS 6896 [36].

5.4.1.2 Radiant plaque

A typical radiant plaque heater is shown in Figure 5.20. Heaters of this type offer outputs typically in the range of 5 to 40 kW. They operate at around 900 °C and are often used for local spot heating. Due to the high operating temperatures, the ceramic burners glow red/orange in use. Like un-flued radiant tube types, they must be located where ventilation rates are high to avoid condensation and to dilute flue gases. A cone configuration is available to provide 360° coverage of a particular location; domestic patio heaters are small-scale portable versions of this type of heater.

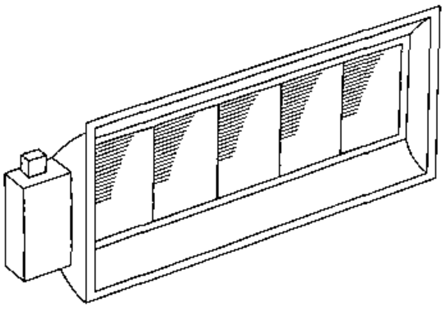


Figure 5.20 Gas-fired radiant plaque heater (reproduced from EEBPP Good Practice Guide GPG303 by permission of the Energy Efficiency Best Practice Programme)

5.4.2 Electric radiant heaters

Electric radiant heaters typically use quartz-enclosed radiant elements operating at up to 2000 °C and parabolic reflectors. They have good directional properties and 100% efficiency in converting from electricity to heat; however, energy costs are high and upstream carbon dioxide emissions are high when electricity is generated from fossil fuels. They are mostly used for local spot heating, mounted at levels between 2 and 4 metres. A typical unit is shown in Figure 5.21.

5.4.3 Controls for radiant heating

As noted in section 4.6.3, control of radiant heating should ideally rely on the sensing of dry resultant temperature, which requires the use of a black-bulb thermometer. Hemispherical black-bulb sensors are available for wall mounting, but suffer from the disadvantage that they are not located at the point where control is required. Also, they tend to be slow to respond. Control based on sensing air temperature is also used, particularly when the whole space is being heated (as opposed to spot heating). Stand-alone controllers may be used or the control function may be integrated into a building management system (BMS).

5.5 Chimneys and flues

5.5.1 Environmental legislation affecting chimneys and flues

Several different strands of legislation are relevant to the design of flues and chimneys, depending on the power of the plant they serve, the fuels used and where they are located.

The Environmental Protection Act 1990 [37] gives powers to local authorities to control pollution from industrial and other processes, which includes the generation of heat and power. Large scale ('Part A') processes, with an output exceeding 50 MW, are subject to control by the Environment Agency. Local authorities control smaller scale ('Part B') processes, which may include large boilers and CHP units. One of the many requirements is for the use of 'best available techniques not entailing excessive cost' ('BATNEEC') to meet limits on levels of contaminants in flue in flue discharges.

The Environment Act 1995 [38] includes provisions for 'local air quality management' and sets air quality standards for seven key urban pollutants: nitrogen dioxide, carbon monoxide, sulphur dioxide, PM10 particles, benzene, 1,3 butadene and lead. An area where any of the standards is likely to be exceeded must be designated as an 'air quality management area' and action taken to reduce levels. This can lead to additional restrictions on development in those areas.

Part 1 of the Clean Air Act 1993 [39] prohibits the emission of 'dark smoke', including emission from a chimney of any building. Part 2 empowers the Secretary of State to prescribe limits on the rates of emission of grit and dust from the chimneys of furnaces, including boilers and other heating appliances. Section 14 of the Act requires that chimney heights must be approved by local authorities for furnaces burning liquid or gaseous fuel at a rate equivalent to 366.4 kW or more, solid matter at a rate of 45.4 $kg.h^{-1}$ or more, or pulverised fuel.

The legislation has an important impact on the design of chimneys and flues, particularly on the height at which combustion products are discharged to the atmosphere. Chimneys contribute to the control of local pollution levels by dispersion and consequent reduction of concentrations at ground level. Dispersion is effective over a range of around 50 to 100 times the chimney height, beyond which it has little effect. For large plant or plant with special characteristics or restrictions, it is likely that individual dispersion modelling will be required. However, plant used for heating can for the most part be dealt with using published guidance. *The Clean Air Act Memorandum: Chimney Heights* (3rd edition) [40] has long been recommended as a source of this guidance and remains valid. However, some types of plant require additional considerations to meet the requirements of the Environmental Protection Act; reference should be made to HMIP Guidance Note D1: *Guidelines for Discharge Stack Heights for Polluting Emissions* [41]. CIBSE TM21 [42] provides guidance on minimising pollution at air intakes, including the contribution made by chimneys and flues. For natural gas and other very low sulphur fuels, guidance may also be obtained from British Gas publication IM/11 [43] .

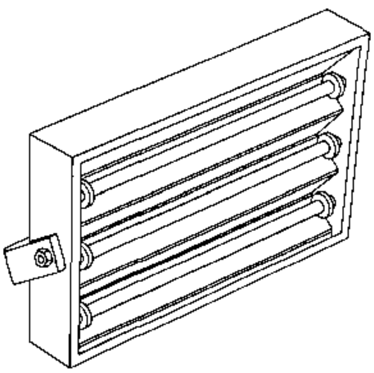


Figure 5.21 Electric radiant heater with quartz enclosed elements and parabolic reflectors (reproduced from EEBPP Good Practice Guide GPG303 by permission of the Energy Efficiency Best Practice Programme)

5.5.2 The Building Regulations

Part J of the Building Regulations [44] applies to all chimneys and flues, irrespective of the type of building, or the capacity of the appliance they serve. It includes the following requirements:

- that sufficient combustion air is supplied for proper operation of flues
- that combustion products are not hazardous to health

- that no damage is caused by heat or fire to the fabric of the building.

Similar requirements are contained in Part F of the Building Standards (Scotland) Regulations [45] and the Building Regulations (Northern Ireland) [46] .

Approved Document J[47] gives guidance on how to satisfy the requirements of Part J. It also makes clear that although Part J applies to all heat producing appliances, the guidance in the Approved Document itself deals mainly with domestic installations. Accordingly, the specific guidance it contains is limited to solid fuel installations of up to 50 kW rated output, gas installations of up to 70 kW net (77.7 kW gross) rated input and oil installations of up to 45 kW rated heat output. The guidance includes:

- the positioning of flues in relation to boundaries and openings
- protection from heat for persons likely to come into contact with flues
- the diameter of flues required for different types of appliances
- materials from which flues and chimneys may be constructed
- how chimneys may be lined to serve gas fired appliances.

For installations with ratings higher than those mentioned above, the guidance referred to in section 5.5.1 applies. Specialist assistance is likely to be required for large installations (above 366 kW), which are also subject to the Clean Air Act. However, some larger installations may be shown to comply by adopting the relevant recommendations contained in this Guide, and codes of practice and standards produced by BSI (particularly BS 6644 [48] and BS 5854 [49]) and the Institution of Gas Engineers.

5.5.3 Principles of flue and chimney design

A chimney or flue must produce sufficient suction to enable the installed plant to operate as intended and to disperse flue gases effectively. A natural draught chimney produces suction at its base by virtue of the difference in the density between the column of hot gas within the chimney and the outside air. This can be expressed by the formula:

$$\Delta p_d / H = (p_a - p_g) g \qquad (5.15)$$

where Δp_d is the pressure difference between top and bottom of chimney (Pa), H is the height of the chimney (m), p_a is the density of ambient air (kg.m^{-3}), p_g , is the mean density of flue gases (kg.m^{-3}) and g is the acceleration due to gravity (m.s^{-2}).

The draught produced by a chimney is proportional to its height and the temperature of the gas within it. Figure 5.22 shows the draught available for typical winter and summer ambient conditions at various chimney temperatures. This gross draught is available to provide the energy required to move the flue gases through the particular boiler, flue and chimney system.

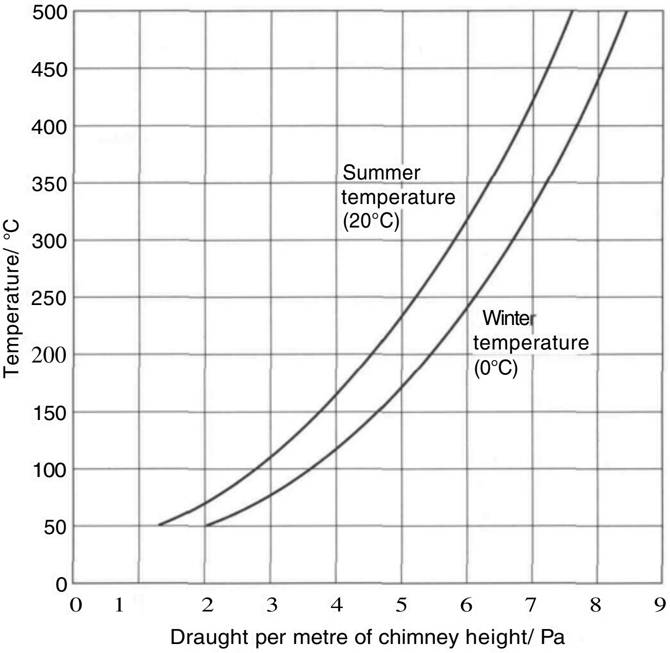


Figure 5.22 Chimney draught at summer and winter temperatures

5.5.3.1 System resistance

The chimney/flue cross-sectional areas must be selected taking account of system resistance to gas flow and the required efflux velocity from the chimney terminal. It is important that the flue layout is carefully considered and designed to limit shock losses at bends etc. In general the following aspects should be observed in flue design:

(a) Position the boilers as close as possible to the chimney to limit friction and heat losses in the connecting flue system.

(b) Avoid all short radius 90° bends in flue systems.

(c) Avoid abrupt section changes and use transformation sections with 15° included angles.

(d) Arrange the entry section to slope at 45° or more to the horizontal

(e) Avoid protrusion of the flues beyond the inner face of the chimney or main flue connection.

(f) Make flues circular or square and avoid aspect (width to depth) ratios greater than 1.5 to 1.

(g) Slope flues upwards towards the chimney where possible.

(h) Provide clean-out doors at each bend in the flues, at the chimney base, and adjacent to fans and dampers to aid maintenance.

(i) Avoid long 'dead' chimney pockets under the flue entry points, which are corrosion zones, and can cause harmonic pulsation problems.

5.5.3.2 Chimney efflux velocity

Chimney gas efflux velocities need to be high enough to avoid 'down-washing' of flue gases on the leeward side of the chimney. Guidance on chimney design is usually based on minimum full-load efflux velocities of 6 m.s^{-1} for natural draught and 7.5 m·s^{-1} for fan forced or induced draught installations. Low efflux velocities may also cause inversion, whereby cold air enters the top of the chimney and flows downward, reducing chimney internal skin temperatures below the acid dew-point and causing acid smut emission. The maintenance of an adequate efflux velocity at all loads is difficult where one chimney serves more than one boiler, particularly if each boiler has high/low or modulating firing.

It may not always be possible to achieve efflux velocities of 6 m·s-' on natural draught plant, particularly if the whole flue and chimney system is designed on this velocity basis, due to the excessive system resistance involved. In such cases, the system can be designed for a lower velocity and a nozzle fitted at the chimney outlet to increase efflux velocity to the extent that the excess available draught allows.

5.5.3.3 Flue corrosion and acid smut formation

Flue gases have a dew-Point below which water vapour condenses. With sulphur bearing fuels, a second acid dew-point occurs at a higher temperature that depends on the type of fuel, amount of excess air, sulphur content and combustion intensity. The sulphur in the fuel is oxidised to SO_2 during the combustion process and a proportion of this is oxidised further to SO_3, with subsequent formation of sulphuric acid.

The peak rate of corrosion tends to occur some 30–40 "C below the acid dew-point and a dramatic increase in corrosion rate occurs below the water dew-point. Acid dew-points generally lie in the range 115–140 "C for the type of boiler plant used for heating but depend upon excess air used, flame temperature, sulphur content etc. A significant depression in acid dew-point temperature occurs where fuels have less than 0.5% sulphur content. It can also be reduced or eliminated by stoichiometric combustion conditions that can only be approached on very large plants.

A smut is an agglomeration of carbon particles resulting from a combination of stack solids and low temperature corrosion products. If the inner surface of any fluehhimney falls below the acid dew-point temperature of the waste gases, an acidic film forms on the surface. Stack solids adhere to this film and build up into loose layers, which are dislodged and ejected from the chimney as the firing rates change.

5.5.3.4 Flue/chimney area and siting

Where chimneys are oversized, or where more than one boiler is used with one flue/chimney, the inner chimney surface temperatures may fall below acid dew-point conditions, even with insulation applied. To avoid these problems, it is strongly recommended to install one flue/chimney per boiler, correctly sized for maximum practicable full load flue gas.

Chimney outlets should not be positioned such that air inlets into the building are on the leeward side of the chimney for the prevailing wind direction. Generally internal chimneys have less heat dissipation than free-standing units but where external chimneys are used they should, where possible, be positioned on the leeward side of the building or site, considering the prevailing wind direction.

When the flue/chimney area is reduced to give high flue gas velocities and a pressurised flue system, the construction of flues and chimneys must be carefully considered. With a mild steel flue/chimney system all joints should be welded or otherwise permanently sealed. Expansion should be accommodated by means of bellows type expansion joints and all explosion relief doors, clean out doors etc. should be fitted with the requisite joints to withstand pressurised flue conditions. Where concrete or brick chimneys with lining bricks are used, they should generally be sized to be under suction conditions unless the construction is specifically designed for operation under pressurised flue conditions. By combining several flues into one insulated envelope the cooling losses are reduced and the effective chimney plume height increased. The chimney outlet should be at a minimum height of 3 m above the highest point of the adjacent building roof level in order to limit wind pressure variations on the flue outlet and present the minimum face area to the prevailing wind. Chimney heights must comply with environmental legislation and the Building Regulations, see section 5.5.4. The sizing and height of chimneys and flues is considered in detail in Appendix A2.

5.5.3.5 Cold air admission

The admission of cold air into the flue/chimney system reduces the flue gas temperatures and hence the available natural draught. Draught stabilisers deliberately introduce cold air to regulate the draught by this means. The use of draught stabilisers is not recommended when high sulphur fuels are used, as reduced flue gas temperature also produces corrosion and acid smut emissions.

Dampers for draught regulation should be fitted with safety interlocks to prevent firing against a closed damper. With high chimneys the damper should be arranged to close when the firing equipment is off-load, to isolate the boiler and limit cold air ingress to the system. This limits the cooling effect on the internal flue and chimney system, and the corrosion mechanism within the boiler gas-side heating surfaces.

5.5.3.6 Heat loss

To enable the correct chimney construction to be selected it is necessary to predict the minimum internal surface temperature likely to be obtained at the chimney terminal under all loads. An approximate value may be obtained using the following method. It should be noted that average values are used for some parameters and that radiation from the gases to the chimney is ignored in order to simplify calculations.

The rate of heat loss from the chimney or duct is given by:

$$\phi_c = U A \, (t_g - t_{ao}) \tag{5.16}$$

where ϕ_c is the heat loss rate (W), U is the overall thermal transmittance (W.m^{-2}.K^{-1}), A is the surface area (m^2), t_g is the mean waste gas temperature ("C) and t_{ao} is the outside air temperature ("C).

The overall thermal transmittance is given by:

$$\frac{1}{U} = \frac{1}{h_o} + \frac{l_1}{\lambda_1} + \frac{l_2}{\lambda_2} + \cdots + \frac{1}{h_1} \qquad 5.17$$

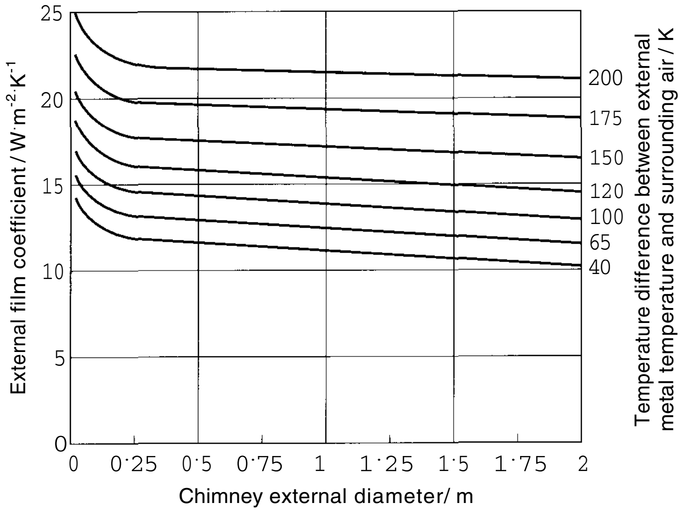


Figure 5.23 Values of external film coefficient

where h_o is the external film coefficient (W.m^{-2}.K^{-1}), l_1 etc. is the thickness of chimney layer 1etc. (m), λ_1 is the thermal conductivity of chimney layer 1 and h_i is the internal film coefficient (W.m^2.K^{-1}).

Values of film Coefficients h_o and h_i are given in Figures 5.23 and 5.24.

The heat loss may also be deduced from:

$$\emptyset_c = q_m c_p \ (t_{g1} - t_{g2}) \tag{5.18}$$

where qm, is the mass flow rate of gases (kg.s^{-1}), c_p is the specific heat capacity at constant pressure of waste gases (J.kg^{-1}.K^{-1}), t_{g1} is the temperature of gases entering the bottom of the chimney (°C) and t_{g2} is the temperature of gases leaving the top of the chimney (°C).

Alternatively, the volume flow rate of waste gases (m^3.s-') may be used in conjunction with the specific heat capacity (J.m^{-3}.K^{-1}). The specific heat is usually taken to be 1.22 kJ.m^{-3}.K^{-1} at 200 °C.

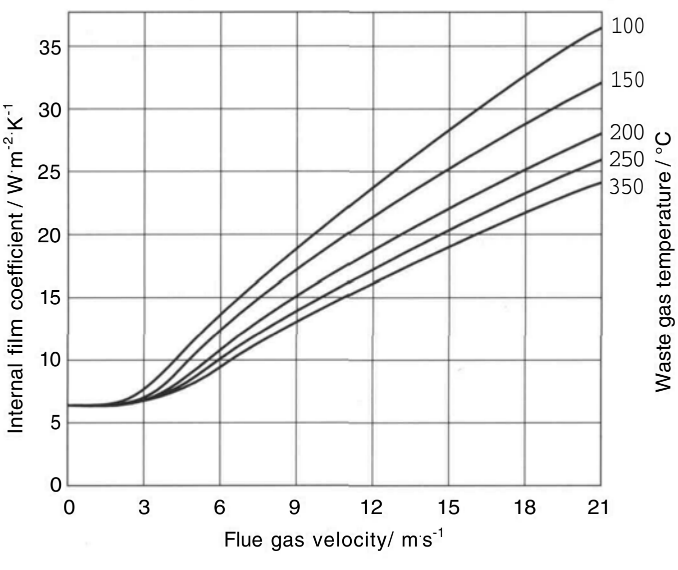


Figure 5.24 Values of internal film coefficient

For thermal equilibrium, equations 5.16 and 5.18 must give the same heat loss, so they may be equated, i.e:

$$U A \ (t_g - t_{ao}) = q_m c_p \ (t_{g1} - t_{g2}) \tag{5.19}$$

where t_g is given by:

$$t_g = \tfrac{1}{2} \ (t_{g1} + t_{g2}) \tag{5.20}$$

If the temperature of the waste gases entering the chimney or duct is known or estimated, the temperature of the gases leaving the chimney may be determined from equation 5.19. The minimum surface temperature may then be established from:

$$h_i \ (t_{g2} - t_{si}) = U(t_{g2} - t_{ao}) \tag{5.21}$$

where t_{si} is the temperature of the inside surface of chimney (°C).

5.5.4 Compliance with Building Regulations and environmental legislation

Detailed guidance for the design of chimneys and flues for small appliances is given in Approved Document J([47]) and BS 5440([50]).For higher rated outputs, the methods outlined in Appendix A2 may be followed. Larger plant, which falls within the scope of the environmental legislation, should be assessed in accordance with HMIP Technical Guidance Note Dl([41]), see section 5.5.1. There also a general requirement to follow the manufacturer's instructions for installation and maintenance.

5.5.5 Draught production equipment

The draught necessary to move flue gases through the flue/chimney system and discharge them at a suitable velocity under specified firing rates can be produced in several ways, as described below.

5.5.5.1 Natural draught systems

Natural draught chimneys are generally favoured for the smaller range of open bottom cast-iron sectional boilers fitted with oil or gas burners and for suspended warm-air heaters. The natural draught in the stack has to overcome the boiler resistance to gas flow. A draught diverter is usually fitted in the flue next to the boiler outlet to maintain correct combustion conditions under all firing conditions. Flue gas velocities must be relatively low in order to reduce system resistances to a practical level, especially where chimneys are not of excessive height. As a result, chimney cross-sectional area is generally greater than for a forced draught system of similar boiler capacity.

5.5.5.2 Forced draught systems

In forced draught systems, the firing equipment is fitted with a fan to provide the necessary combustion air and to overcome the burner resistance and the boiler resistance to gas flow. The chimney draught required in these cases has to overcome less overall resistance than in the natural draught case and flue gas velocities can often be increased for a given chimney height. Forced draught is typically

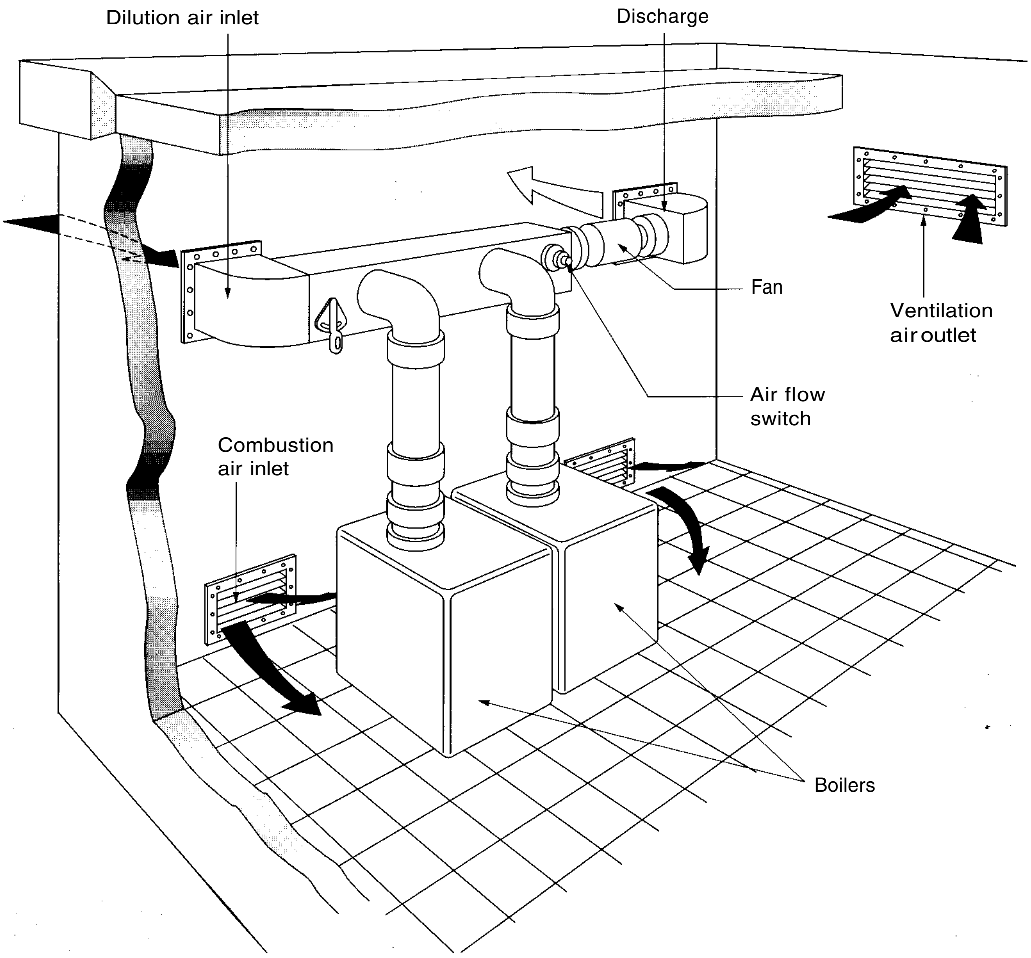


Figure 5.25 Typical fan dilution flue system

used with oil- or gas-fired packaged steel shell or cast iron sectional boilers or cabinet warm-air heaters.

5.5.5.3 Induced draught systems

A fan may be fitted at the boiler outlet to cater not only for the resistance of the firing equipment and the boiler but also, in certain instances, of the flue and chimney when burning at maximum rating. Examples of this type are found in coal-fired shell boilers and certain water tube boilers. Due to the fan power employed, draught is not dependent upon chimney buoyancy conditions and gas velocities can be increased depending upon the fan power requirement.

5.5.5.4 Balanced draught systems

A forced draught fan is fitted to provide all combustion air and overcome the resistance of air registers, or fuel bed. An induced draught fan is fitted at the boiler outlet to take the hot gases and overcome resistance of the boiler and the flues and chimney system. It is usual to fit a draught controller which, by damper control on the fans, maintains the balanced 'zero' condition in the combustion chamber. Examples of this type are found in most coal-fired boilers fitted with chain grate stokers and oil- and gas-fired water tube boilers. Due to the fan power employed high velocities can be used in the flue system, which again is not dependent upon chimney height. Generally such an arrangement is only applicable to larger installations.

5.5.5.5 Fan dilution systems

Fan dilution was developed for gas appliances in ground floor shops in mixed developments of offices, shops and flats. Fresh air is drawn in through a duct by a fan, mixed with the products of combustion, and finally discharged to the atmosphere with a carbon dioxide content of not more than 1%. In practice, fan dilution is only suitable for gas, which has very low sulphur content. A typical system is shown in Figure 5.25.

Fan dilution systems have been used extensively for launderettes, shops, restaurants, public houses etc. Many local authorities allow the discharge to be made at low level, above a shop doorway for instance, or into well ventilated areas with living or office accommodation above.

To comply with legislation:

— diluted exhaust must be discharged not less than 2 m (preferably 3 m) above ground level

— diluted exhaust must not be discharged into an enclosed courtyard

— the terminating louvre must be at least 3.54 m from the nearest building.

Fan dilution is normally used where natural draught flues are not practical. Ideally, the air inlet and discharge louvres should be positioned on the same wall or face of the building. Shielding is recommended if the louvres are likely to be subjected to strong wind forces. A damper or butterfly valve is fitted near the dilution air inlet to balance the installation. Protected metal sheet can be used for ducting as flue temperatures with this system are low, typically 65 °C.

5.5.5.6 Balanced flues

Balanced flues are used mainly for domestic gas-fired appliances and suspended warm-air heaters, but may also be used for low sulphur content fuels, e.g. kerosene. The appliance is of a room-sealed construction and is sited adjacent to an outside wall. The air for combustion is drawn from outside and the products of combustion are discharged using a common balanced flue terminal. The close proximity of air inlet and combustion products outlet makes the balanced flue terminal relatively insensitive to wind conditions and location.

At present, balanced flue boilers and heaters are available only in the lower output range but special designs are possible for larger outputs. Fan assistance can be used to reduce the size of the flue assembly and allow the appliance to be sited away from an external wall. Balanced flue terminals must be sited in accordance with Part J[44] of the Building Regulations.

5.5.5.7 SE-ducts and U-ducts

SE-ducts and U-ducts increase the scope for applying room-sealed gas appliances to multi-storey dwellings by making possible the connection of many appliances to a single flue system. An SE-duct is a single rising duct, with an opening at the bottom to provide fresh air for the appliance and an opening at the top to provide an outlet for combustion products. A U-duct is a pair of rising ducts joined at their base and open at the top, one to provide fresh air and the other to provide an outlet for combustion products. Only room-sealed appliances may be connected to SE-ducts and U-ducts and connection should be made in compliance with guidance obtained from the manufacturers of the appliances.

5.5.5.8 Branched flue systems

The branched flue system for gas appliances, sometimes called the shunt system, is designed for venting appliances of the conventional flue type. It represents considerable space-saving Over venting each appliance with an individual flue. For further information on conditions and sizing see BS 5440(50).

5.5.6 Chimney linings

Chimneys should have internal surfaces that:

— have sufficient thermal insulation to maintain inner skin temperatures above the acid dew-point during normal running operations

— are chemically resistant to acids and flue gas deposits generally

— resist absorption of moisture and its re-evaporation

— can withstand fairly rapid internal gas temperature changes

— have low thermal capacity to limit heat up time

— can be installed, inspected and replaced economically.

Flexible steel liners may be used to line existing chimneys but Building Regulations do not permit their use in new masonry chimneys.

5.5.7 Chimney construction

5.5.7.1 Stainless steel

Stainless steel chimneys are available with either single-skin construction or with a twin wall in diameters up to 600 mm. Twin wall types may have either an air gap or insulation.

5.5.7.2 Steel

Steel chimneys are either of single or multi-flue construction, the outer windshield being designed to cater for the required wind pressures under either guyed or self-supporting design conditions. The structural requirements are covered by BS 4076(51).

With single flue construction a simple method of insulation consists of applying externally a cladding of 1.6 mm polished aluminium sheet located 6 mm from the Outer mild steel chimney surface by means of heat resisting spacers at 1.2 m intervals. This provides a 6 mm stagnant air space for insulation, assisted by the reflectivity of the polished aluminium.

With high sulphur fuels and chimneys having a gas volume turndown of more than 2.5 times with modulating or two-position firing equipment, this insulation is insufficient for chimney heights above 10-12 m. A mineral wool insulation at least 50 mm thick should be substituted for the 6 mm air space. with multi-flue construction the inner flues are placed within a windshield structurally calculated for wind pressures etc. as before. The internal flues are either insulated with mineral wool, or the whole space around the flues filled with a loose insulation that can be pumped into place. Thermal expansion problems must be considered in the design and provision made for replacing any one flue at a future date.

A similar mild steel multiple flue system can be installed within a concrete structural outer shell, again providing facilities for subsequent replacement.

5.5.7.3 Brick

Brick flues/chimneys should always be lined internally. For solid or liquid fuels the lining may be gunned solid insulation refractory or diatomaceous earth type insulation. The insulation standard should not be less than the equivalent of 115 mm thickness of diatomaceous earth for flue gas temperatures up to 315 °C.

Where flue gas conditions dictate (e.g. low temperature, high sulphur and moisture) an acid resisting brick inner lining, backed by a lining of insulation material, can be used. Careful attention must be paid to the lining construction and the type of jointing mortar used to prevent flue gases leaking through behind the lining and setting up corrosive conditions.

The effect of pressurised operation on these linings is questioned and for general operation such chimneys should be operated under suction or balanced draught conditions. They must be carefully designed by a competent structural engineer who is aware of the combined physicalchemical effects involved.

5.5.7.4 Concrete construction

Similar comments to those on brick construction apply, but the insulation thickness should generally not be less than the equivalent of 150 mm diatomaceous earth in order to limit the interface concrete temperature to a maximum of 50 °C under normal boiler plant operating conditions.

5.5.7.5 Ventilated chimneys

Here a ventilated air space is situated between the inner lining and outer chimney shell. The construction should not be used in general with high sulphur fuels due to the cooling effect created and the consequent danger of acid dew-point and acid smut emission.

5.6 Corrosion in boilers, flues and chimneys

5.6.1 Mechanisms of corrosion

The most common cause of corrosion in boilers and chimneys is the presence of water vapour and oxides of sulphur following combustion of fuels containing sulphur.

When any fuel containing hydrogen and sulphur is burned, Water vapour and sulphur dioxide (SO_2) are produced. A small proportion of the SO_2 is further oxidised to sulphur trioxide (SO_3), which immediately combines with water vapour to produce sulphuric acid. This will condense. on any surface below the acid dew-point temperature, giving rise to corrosion. The acid dew-point is the temperature at which the combustion gases become saturated with acid vapour and, when cooled without change in pressure, condense as a mist.

The acid dew-point varies with the type of acid and its concentration. Further cooling of the gases to the water dew-point may produce corrosive effects even more serious than those produced at higher (i.e. more concentrated) acid dew-points. During normal operation it is unlikely that the water dew-point (about 38 "C) will be reached but this may occur for intermittently operated plant. When the system operation is such that the water circulating temperatures can fall to 38 °C, condensation is inevitable.

In addition to sulphur, other constituents such as chlorine and nitrogen react to give acidic gases which can combine with water vapour and thereby cause corrosion if allowed to condense on cooler metal surfaces.

In boilers that are shut-down, flue deposits become damp because of their hygroscopic nature and produce acid sulphates which are likely to cause corrosion. Acid corrosion is less likely to occur with coal rather than residual fuel oils, for the following reasons:

- the average sulphur content of coal is generally lower than that of residual fuel oils and about one tenth is retained in the ash
- the hydrogen content of coal is lower than that of other fuels, therefore the amount of water vapour produced during combustion is also lower
- the small amounts of fly ash in the flue gases tend to absorb free SO_2 and thus reduce the production of corrosive acid.

The combination of less water vapour and lower levels of SO_2 means that lower gas temperatures may be used, resulting in a corresponding gain in plant efficiency. On large, well operated and maintained plant, the production of SO_2 may be minimised by controlling the excess oxygen in the combustion zone. However, precise control is necessary and this is unlikely to be achieved on small plants.

5.6.2 Prevention of corrosion in boilers

During boiler shut-down in the summer months, all surfaces should be cleaned of all partially burned fuel and ash, and dampers should be left open to ensure that air is drawn through the boiler. Lime washing of all accessible surfaces may be beneficial and, where good air circulation can be obtained, trays of a moisture absorbing material, such as quicklime, should be provided.

For plant in operation, the system should be designed so that the average boiler water temperature does not fall below about 50 oc. This helps to enSure that the water dew-point is not exceeded.

For details of control of boiler systems, see section 5.1.2 and CIBSE Guide H (24). Under no circumstances should the boiler thermostat be used as a control thermostat to reduce the flow temperature in a heating system.

Low temperature hot water boiler corrosion usually occurs at the smoke box prior to the flue connection, and is often referred to as 'back-end corrosion'. Maintaining the return water temperature above 50 "C can provide protection from this type of corrosion. At start-up, a thermostatically controlled bypass between the flow and return connections can be used to blend a small proportion of hot flow water with cooler return water. Circulation is achieved either by a small shunt pump or by connecting the flow end of the bypass pipe to the primary pump discharge and controlling the flow/return blend through a three-port valve. In each case, the bypass is isolated automatically when the system return temperature reaches the pre-set minimum.

5.6.3 Prevention of flue corrosion

To minimise the risk of corrosion, the following points should be noted.

- Sufficient insulation should be provided to maintain inner skin temperature above the acid dew-point during normal operation.
- The flue or chimney lining should be chemically resistant to acids and flue gas deposits.
- The flue gas velocity must be sufficiently high to prevent precipitation of acids and deposits on internal flue linings.
- Avoid abrupt changes of direction in the flue and stack.

The flue connection from the boiler should rise to the stack and be kept as short as possible.

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6 Fuels

6.1 Classification and properties of fuels

6.1 .I Gaseous fuels

The main gaseous fuels are broadly classified as natural gas and liquefied petroleum gases *(LPG).* Natural gas consists predominantly of methane and is delivered by pipeline. *LPG* includes propane and butane, and is delivered as a liquid contained in a pressurised vessel. The key properties of the main gaseous fuels are shown in Table 6.1.

6.1.1.1 Wobbe number

The Wobbe number (w> is designed to indicate the heat produced at a burner when fuelled by a particular gas, and is defined as:

$$W = h_g/d^{0.5} \qquad (6.1)$$

where W is the Wobbe number (MJ.m-3), h_g is the gross calorific value (MJ.m-^{3}) and d is the relative density of the gas (relative to air at standard temperature and pressure).

6.1 .1 .2 Supply/working pressure

Natural gas supplies are regulated at the metering point to an outlet pressure of 2100 Pa (21 mbar). This pressure may be reduced further at the appliance to provide the required pressure at the burner. LPG is supplied via tanks or cylinders, regulated to a pressure of 3700 Pa (37 mbar) for propane and 2800 Pa (28 mbar) for butane. This pressure is not normally reduced at the appliance.

6.1 .I .3 Landfill and sewage gas

Landfill gas is collected from wells inserted in land-fill sites, often complementing measures to prevent hazards arising from the escape of gas. It typically consists of between 40 and 60% methane by volume with the remainder mostly carbon dioxide and traces of many other gases. The calorific value of landfill gas is in the range 15 to 25 MJ.m-^{3}, depending on its methane content.

Landfill gas is mostly used without processing, other than the removal of moisture and dust. Because of its low calorific value it is relatively expensive to transport and is most suitable for heat generation when it can be produced close to a heat load, which favoured its early exploitation for brick kilns adjacent to clay pits used for land fill. In other cases, it is used to generate electricity from gas turbines or reciprocating engines. The life expectancy of gas production from landfill sites is typically 15 to 30 years.

Sewage gas is produced from digestion of sewage sludge. Some of the gas produced is used to maintain optimum temperature for the digestion process. It is economic in many cases to use combined heat and power generation in this situation, and to export the surplus power to the grid.

6.1.2 Liquid fuels

6.1.2.1 Oil fuels

BS 2869[1] contains specifications for various classes of liquid fuels designated by the letters A to G. The fuels commonly used for heating are Class C2 (kerosene or burning oil), Class D (gas oil), Class E (light fuel oil), Class F (medium fuel oil) and Class G (heavy fuel oil). The key properties of these fuels are shown in Table 6.2.

Table 6.1 Properties of commercial gas supplies at standard temperature and pressure

Property	Natural gas	Commercial propane	Commercial butane
Density relative to air	0.60	1.45 to 1.55	1.9 to 2.10
Gross calorific value (MJ.m^3)	38.7	93	122
Wobbe number (MJ.m-^{3})	45 to 55	73.5 to 87.5	73.5 to 87.5
Supply/working pressure (Pa)	1750 to 2750	3700	2800
Stoichiometric air to gas volume ratio	9.73	24	30
Flame speed (m.s-^{1})	0.43	0.47	0.38
Flammability limits (%gas in air)	5–15	2–10	2–9
Boiling point (°C)	—	−45	0
Latent heat of vaporisation (kJ.kg-^{1})	—	357	370
Flame temperature ("C)	1930	1950	—
Ignition temperature ("C)	704	530	470

Table 6.2 Key properties of typical petroleum fuels

Property	Class C2	Class D	Class E	Class F	Class G
Density at 15 "C (kg,m-3)	803	850	940	970	980
Minimum closed flash point (°C)	38	60	66	66	66
Kinematic viscosity (mm2.s-') at 40 °C	1.0 to 2.0	1.5 to 5.5	—	—	—
Kinematic viscosity (mm2d) at 100 °C	-	-	≤ 8.2	≤ 20.0	≤40.0
Maximum pour point (°C)	—	-	-6	24	30
Gross calorific value (MJ.m-3)	46.4	45.5	42.5	41.8	42.7
Net calorific value (MJ.m-3)	43.6	42.7	40.1	39.5	40.3
Maximum sulphur content by mass (%)	0.2	0.2	3.2	3.5	3.5
Mean specific heat 0-100 °C (MJ.kg-I)	2.1	2.06	1.93	1.89	1.89

Further information can be found in CIBSE Guide C[2], section 5.5.2, including graphs showing the kinematic viscosity of fuel oils at different temperatures.

6.1.2.2 Liquid bio-fuels

Fuels may be produced from crops grown specifically for the purpose. Historically, the principal fuel crop has been coppice wood for charcoal production. In recent decades, interest has been focussed on the production of liquid and gaseous fuels suitable for use in transport. The process and the crops used determine the type of fuel produced. Thermal processing (by combustion, gasification or pyrolysis) is best suited to dry materials. Anaerobic fermentation is better suited to wet bio-mass materials, which can yield both methane rich bio-gas and liquid fuels, according to the type of fermentation used. Ethanol has been produced commercially from sugar cane, notably in Brazil. In Europe, rape seed oil is used to produce bio-diesel, which has very similar properties to petroleum-derived diesel and is capable of being used in existing engines without significant modification.

6.1.3 Solid fuels

Coal is classified according to its chemical composition and graded according to size. CIBSE Guide C[2], section 5.5 gives the properties of numerous varieties of coal, including moisture, ash and sulphur content. Gross calorific value ranges from 24 to 34 MJ.kg^{-1}.

Municipal waste may be burnt unprocessed, with heat extracted or electricity generated as part of the incineration process. Alternatively it may be used to produce refuse-derived fuel pellets, which may be used to fire some types of boiler plant. It has a calorific value about two thirds of that of coal and produces around 50% more ash.

Wood fuels are of interest because their use can result in a net decrease in greenhouse gas emissions. Forestry waste results from the normal processes of forestry management, which is has the principal objective of maximising the value of the timber crop. Thinning and harvesting leave residues, consisting of branches and tree tops which have no value as timber and, if not used for fuel, would be discarded. Waste wood is also available from industrial sources, particularly from saw-milling and furniture making. Its use as a fuel has a net benefit in greenhouse gas emissions, both by avoiding the need to burn a fossil fuel and by avoiding the production of methane that would result from decomposition on the forest floor or in landfill.

Wood fuel may be produced by growing arable coppice specifically for fuel production. The carbon dioxide released on combustion will have been sequestered during growth and there is no net contribution to CO_2 emissions. Notwithstanding its environmental advantages, wood is a low quality fuel, with a calorific value of around 19 MJ.kg^{-1} when dry and only around 10 MJ·kg^{-1} at the typical moisture content (55%) when harvested.

Straw is also used as a fuel, particularly since the phasing out of straw-burning on fields in the early 1990s. It is burnt in high temperature boilers and used to supply heat and hot water, usually on a fairly small scale.

6.1.4 Electricity

Electricity is the most versatile form in which energy is delivered and may serve almost any end-use of energy, including those for which fuels are consumed directly. However, the high quality and versatility of electricity must be seen in the context of its high cost, which reflects the high primary energy input to electricity generation.

The *Digest of UK Energy Statistics*[3] shows that the generation mix for electricity in the UK has changed radically since 1992, when gas-fired power stations began to come on stream, displacing coal-fired plant. In 1991, 65% of all fuel used for generating electricity connected to the public electricity supply system was coal and less than 1% gas; in 2000, these proportions had changed to 33% and 35% respectively. When account is taken of the higher efficiency of gas generation, the proportion of electricity supplied from gas generation is even higher, at 39% (c.f. 31% from coal generation). Nuclear power accounted for 21% of electricity supplied in 2000, a proportion that has not changed substantially over recent years. Hydro-electricity contributed only 1.3%, although pumped-storage hydro-electricity stations perform an important role in balancing system loads.

The shift towards gas generation has several important implications for UK electricity, apart from fuel supply considerations. Gas produces negligible emissions of sulphur dioxide to the atmosphere, and reduced concentrations of other atmospheric pollutants. As a result, UK sulphur dioxide emissions from power stations have declined by around two-thirds since 1990, contributing to a greater than 50% reduction in UK emissions from all sources. The amount of carbon dioxide released per unit of heat energy obtained from gas is also lower than for coal. The current generation of gas-fired power stations using combined cycle technology are more efficient than coal-fired stations. The overall effect is that the gas generated electricity is less than half as carbon intensive as coal generated electricity. Coefficients for carbon dioxide emissions of fuels are given in section 6.2 below.

6.1.5 Renewable electricity generation

In addition to the use of bio-fuels described above, there are many possibilities for generating electricity directly from renewable sources of energy based on solar radiation, wind, tides, waves hydropower and geothermal heat.The UK lacks the terrain to permit further exploitation of large-scale hydroelectric power, but there are numerous opportunities for small-scale exploitation. The UK also has very limited opportunities for exploiting geothermal power but has considerable resources for wind, wave and tidal power. A wide ranging assessment of the opportunities for renewable energy in the UK was undertaken by the Energy Technology Support Unit[4] in the early 1990s.

The Digest of UK Energy Statistics(3) shows that the UK produced 2.8% of its electricity from renewable sources in 2000, of which just under half was by large-scale hydro-electricity stations. The remainder came largely from combustion of bio-fuels, led by landfill gas and refuse combustion. Wind power was the largest contributor other than large-scale hydro-electricity and bio-fuels but amounted to only 0.26% of total electricity produced.

Although the contribution to electricity generation by renewable sources is small at present, the government has ambitious targets for expanding it to reach 5% in 2003 and 10% in 2010. In this context it should be noted that renewable generating capacity doubled in the four year period between 1996 and 2000. In the short term much of this expansion will come from wind and land-fill gas.

6.2 Factors affecting fuel choice

6.2.1 Fuel prices

The price of fuel remains a very important factor affecting fuel choice and a strong determinant of life-cycle cost. Current energy prices and recent price trends may be obtained from the Department of Trade and Industry (DTI)[5]. Separate tables are given for domestic and industrial prices.

Figure 6.1 shows how industrial fuel prices changed during the decade to 2000. Most prices were relatively stable and, when allowance is made for inflation during

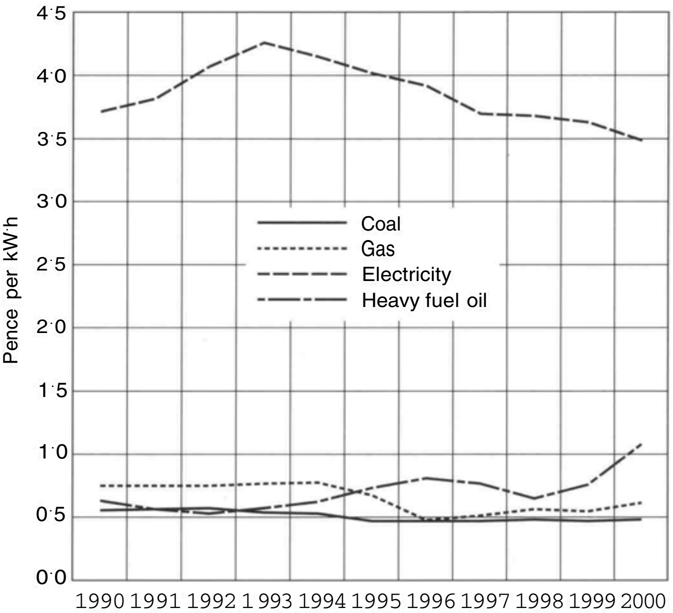

Figure 6.1 Industrial fuel prices during the 1990s in cash terms

the decade, declined in real terms. The most obvious feature of the graph is the high price of electricity compared to fuels consumed at the point of use, which serves to illustrate why electricity should be reserved for purposes in which its special advantages are needed. This normally precludes its use as a principal source of space heating, although it can be economical for localised and occasional use, particularly as radiant spot heating. Coal and gas remained broadly similar in price throughout the period shown, while heavy fuel oil increased significantly at the end of the decade as the price of crude oil rose.

6.2-2 Environmental impact

The use of energy affects the environment both at the Point of use and indirectly, through the upstream activities associated with production, conversion and delivery. It can have detrimental impacts locally on air quality and acid deposition and, on a global scale, on stratospheric Ozone depletion and greenhouse gas concentrations in the atmosphere, which is widely recognised as a likely cause of climate change. As heating accounts for around three-quarters of all energy used in buildings and more than a third of all final energy use in the UK, it is a very significant contributor to the total environmental impact from energy use.

The *Digest of UK Energy Statistics*[3] *identifies* the main sources of CO_2 emissions arising from fuel combustion: 28% from power stations; 24% from industry; 22% from transport and 15% from the domestic sector. This reveals the high CO_2 emissions associated with electricity, which should be taken into account when considering its final use. There are also some additional emissions of CO_2 during the production of gas, oil and solid fuels, which should be similarly taken account of although they are much less significant.

Table 6.3 shows average CO_2 emissions attributable to each unit of energy used in the UK, taking account of upstream and overhead effects. This may be used to compare alternative options for fuel and shows the advantage of natural

Table 6.3 co2 emission factors for UK in 2000-2005(6)

Fuel	CO_2 emission per GJ of delivered energy/ kgC.(kW.h)-l
Natural gas	0.053
LPG	0.068
Gas oil/burning oil	0.074
Coal	0.086
Electricity (average of public supply)	0.113

gas over other fuels. Electricity obtained from the public supply has an emission factor of about two and a half times that of gas.

6.2.3 Other factors affecting fuel choice

Availability of a mains supply of natural gas is a key factor in the choice of fuel, given its advantages of clean combustion and low price. In remote areas, the absence of mains gas normally leaves a choice between oil, LPG and solid fuel, all of which require significant space for storage and access for delivery. Solid fuel is often the lowest in price but has greater maintenance costs than oil or LPG. LPG may be cleanest and most convenient but is generally significantly more expensive than heating oil. Although expensive, electricity may be the best choice where heating requirements are very small, especially if it can be used with a ground source heat pump.

6.3 Handling and storage of fuels

6.3.1 Natural gas

6.3.1.1 Pressures

Natural gas is normally supplied to consumers at gauge pressures up to 5 kPa and regulated to a nominal 2.1 kPa at the gas meter before feeding to appliances. Pressure losses in pipes should not be more than 100 Pa under maximum flow conditions. Tables giving pressure losses for natural gas in steel and copper pipes are given in section 4.7 of CIBSE Guide C(2).

Higher operating pressures are needed for large commercial and industrial plants, where burners are fan assisted or pressurised. This may be obtained from a pressure booster, which can also allow the use of smaller pipework to appliances. The gas supplier must be consulted before fitting a pressure booster, which must include protection against disturbance to the gas supply or damage to the meter by excessive suction or pressure. This is normally achieved using a low pressure cut-off switch and a non-return valve on the gas supply side of the booster and a pressure relief by-pass around the compressor.

6.3.1.2 Pipework

Gas distribution pipework in domestic premises should comply with BS 6891(7) and relevant parts of the Gas Safety (Installation and Use) Regulations(*). In particular, pipes must:

(a) be protected from failure caused by movement when installed in walls and floors

(b) not be installed within the cavity of a cavity wall

(c) not be installed under the foundations of a building or a wall

(d) not be installed in an unventilated shaft, duct or void

(e) take the shortest practicable route through a solid structure and be enclosed in a gas-tight sleeve

(f) be electrically bonded, including temporary bonding during modification.

Steel pipes should comply with BS 1387(9) and copper pipes with BS EN 1057(10).

CIBSE Guide C(2), section 4.7, includes tables giving pressure drop per unit length for natural gas in pipes and pressure loss factors for components such as tees, elbows and valves.

6.3.1.3 Safety

All combustion installations using gas must comply with the Gas Safety (Installation and Use) Regulations(@,which cover the safe installation of gas fittings, appliances and flues. They also require that installation work be undertaken by a member of a class of persons approved by the Health and Safety Executive (HSE); in practice, that means they must be registered with CORGI, the Council for Registered Gas Installers. The main requirements of the Gas Safety (Installation and Use) Regulations are outlined below but for more detailed information reference should be made to the Health and Safety Commission's Approved Code of Practice. For public, commercial and industrial buildings, workplace legislation[11] is also relevant.

The Gas Safety (Installation and Use) Regulations control all aspects of the installation, maintenance and use of systems burning gas (including natural gas and LPG). The text of the Regulations and guidance on how to comply with them are contained in Health and Safety Executive (HSE) Approved Code of Practice L56: *Safety in the installation and use of gas systems and appliances*[12]. The detailed guidance applies principally to small appliances but the similar requirements apply generally.

6.3.2 Liquid petroleum gas (LPG)

6.3.2.1 Storage

LPG installations are subject to legislation enforced by the Health and Safety Executive. For small storage installations, in which the tank stand in the open air, it is possible to show compliance by following the guidance given in Approved Document J[13] of the Building Regulations and

Table 6.4 Installation distances for LPG storage tanks

Tank capacity (m^3)	Minimum distance / m	
	From buildings	Between vessels
<0.45	0*	0.6
0.45 to 2.25	3	0.9
2.25	7.5	0.9
>9	15	1.5

* A tank of less than 0.45 m^3 capacity may be sited close to a buliding, allowing space for maintenance, with a minimum distance of 2.5 m from the tank filing point to any opening of the building

Part 1 of L P Gas Association Code of Practice: *Bulk LPG storage at fixed installations* [14].

LPG (propane) is stored at a pressure of 690 kPa at 15 °C, and within the range 200 to 900 kPa as conditions vary. LPG storage tanks should be installed in the open, not enclosed by a pit or bund, and adequately separated from buildings, boundaries and fixed sources of ignition. Drains, gullies and cellar hatches close to tanks should be protected from gas entry. Reduced separation is permitted when a fire-wall is built between a tank and a building, boundary or source of ignition. Fire walls should contain no openings, have fire resistance of least 60 minutes and be at least as high as the pressure relief valve on the storage vessel.

Where LPG is stored in cylinders, provision should be made to enable cylinders to stand upright, secured by straps or chains against a wall outside the building in a well ventilated position at ground level. LPG storage vessels and LPG fired appliances fitted with automatic ignition devices or pilot lights must not be installed in cellars or basements.

An industrial LPG storage installation usually consists of one or more tanks mounted horizontally on concrete foundations. The general requirements are as for domestic storage with requirements for tanks to be sited clear of buildings to avoid the risk of overheating should a building catch fire. Table 6.4 gives installation distances for tanks of various sizes. Where supplies are to be delivered by road, consideration must be given to access for vehicles.

6.3.2.2 Pipework

pipework for LPG is similar to that for natural gas but allowance must be made for the higher density of LPG. Section 4.3.3 of CIBSE Guide c(2) gives the Colbrook-White equation, from which pressure drops for LPG in flowing in pipes may be calculated.

6.3.3 Oil

6.3.3.1 Storage

The storage of oil in tanks with a capacity of up to 3500 litres is covered by Part J of the Buildings regulation[15].

Requirement J5 seeks to minimise the risk of fire from fuel igniting in the event of fire in adjacent buildings or premises by controlling their construction and separation from buildings and the boundary of the premises on which they stand. It applies to all fixed oil storage tanks with a capacity greater than 90 litres. Requirement 56 seeks to reduce the risk of pollution arising from the escape of oil and applies to tanks serving private dwellings up to a capacity of 3500 litres.

Guidance on compliance with requirements J5 and J6 for Class C2 and Class D is given in Approved Document J[13] of the Building Regulations. It cites BS 5410-1(16) as a source of guidance, supplemented by specific guidance on fire protection and prevention of fuel spillage. In particular, it recommends fire resistant fuel pipework protected by a fire valve system complying with the recommendations given in BS 5410-1, sections 8.2 and 8.3.

Larger installations must also comply with Requirement J5, for which Approved Document J states that advice should be sought from the relevant Fire Authority. Although not covered by Requirement J6, larger tanks serving buildings other than private dwellings are likely to be subject to the Control of Pollution (Oil Storage) Regulations 2001(17).The specification of oil storage tanks is covered by BS 799-5(18).

Where supplies areto be delivered by road, consideration must be given to access for vehicles.

6.3.3.2 Temperatures for storage of liquid fuels

Fuel oils of classes E to H require heating to provide the recommended storage temperatures, which are shown in Table 6.5. Heating may be provided by steam or hot water coils, or by electric immersion heaters. It is usual to maintain tanks at the temperatures given in column 2 of Table 6.5 and raise the temperature further by a separate outflow heater to the level shown in column 3.

Class C and D fuels do not generally require heating, but some class D fuels may require heat to ensure an adequate flow of oil; both tank heating and trace heating on lagged pipes may be used to maintain temperatures in the range 0 tO 5 oc in that instance.

6.3.3.3 Pipework

Single pipe delivery is suitable for class C and D fuels, normally with a positive head at the suction side of the boiler fuel pump. Class E fuel oil should be supplied from a heated storage tank via a circulating ring main, with further preheating of the fuel within the burner before feeding to the atomiser. Class F and G oils require an outflow heater to raise the oil to pumping temperature and trace heating applied to the ring main pipework and other components.

Table 6.5 Storage temperatures for fuel oils

Class	Minimum temperature (°C)	
	Storage	outflow
E	10	10
F	25	30
G	40	50
H	45	55

Table 6.6 Bulk density and specific volume of various coal types

Coal type	Size / mm	Bulk density / tonne·m^{-3}	Specific volume / m^3·tonne^{-1}
Graded	> 12.5	0.80	1.25
Dry smalls	< 12.5	0.83	1.20
Wet smalls	C12.5	0.88	1.14

6.3.4 Solid fuels

6.3.4.1 Storage

Solid fuel is normally delivered by road vehicle and unloaded by tipping or by conveyer. Access for delivery should be designed to suit the type of delivery vehicle expected, taking account of turning circle and space for tipper operation.

Table 6.6 gives the bulk density and specific volume of various types of coal, which may be used to design storage capacity. A minimum capacity equivalent to at least 100 hours operation at full output is recommended. The usable capacity of a bunker depends upon the methods by which fuel is delivered and extracted from the bunker and may be less than the nominal volume. Rectangular bunkers with flat bases are difficult to empty completely without manual trimming. Bunkers with hopper bottoms empty completely but require vehicle access at a high level if they are to be filled by tipper. Bunker bases should be designed to suit the method of coal extraction, avoiding dead volumes that fail to leave the bunker. Low friction linings for outlet chutes may assist free flow and aid extraction.

Bunkers should be covered by grid screens, which are sized to prevent the entry of large objects that could damage the coal extraction equipment. A 100 mm grid is usual, strong enough to support the weight of operators or, if necessary, vehicles.

6.3.4.2 Safety

Hazards can arise from spontaneous combustion and explosions caused by dust or methane. Monitoring of carbon monoxide levels and minimising storage volume during the summer shutdown period can help to avoid spontaneous combustion. Dust and gas explosions arise within certain concentrations, which may be monitored and controlled. Specialist advice should be sought on the prevention of explosions in solid fuel storage.

6.3.4.3 Fuel handling

Screw conveyors and elevators are used to raise coal to mechanical stoker hoppers for small boilers; overhead monorail, skip hoist and pneumatic handlers are also used. For large boiler plant, chain-and-bucket and belt-and-bucket elevators are used, as are belt, drag-link and screw conveyors.

6.3.4.4 Ash extraction and disposal

Fully automatic ash removal is available on some boilers but on others ash and clinker must be removed by hand. Various methods are available for ash handling, including screw and vibratory conveyors and vacuum systems.

For large plant, the ash may be sold directly for use as a construction material. If it is to be used for block making, it should meet the requirements described in BS 3797(1[9]). For smaller plant, ash is likely to removed as part of the general waste removal service, after which it may be disposed of in land fill or supplied to the construction industry.

References

1 BS 2869: 1998 *Specification for fuel oils for agricultural, domestic and industrial engines and boilers* (London: British Standards Institution) (1998)

2 *Reference data* CIBSE Guide *C* (London: Chartered Institution of Building Services Engineers) (2001)

3 *Digest of UK Energy Statistics* (London: The Stationery Office) (published annually)

4 *An Assessment of Renewable Energy for the UK* (London: Her Majesty's Stationery Office) (1994)

5 *Quarterly energy prices* (London: Department of Trade and Industry) (published quarterly)

6 The Building Regulations 2000 Approved Document L *Conservation of fuel and paver* (London: The Stationery Office) (2001)

7 BS 6891: 1998 *Specification for installation of low pressure gas pipework of up to* 28 *mm (R1) in domestic premises (2nd family gas)* (London: British Standards Institution) (1998)

8 Gas Safety (Installation and Use) Regulations 1998 Statutory Instrument 1998 No. 2451 (London: The Stationery Office) (2001)

9 BS 1387: 1985 *Specification for screwed and socketed steel tubes and tubulars and for plain end steel tubes suitable for welding or for screwing to BS 21 pipe threads* (London: British Standards Institution) (1985)

10 BS EN 1057: 1996 *Copper and copper alloys. Seamless, round copper tubes for water and gas in sanitary and heating applications* (London: British Standards Institution) (1996)

11 *Workplace (Health, Safety and Welfare) Regulations* 1992 *Approved Code of Practice and Guidance* Health and Safety Executive L24 (London: HSE Books) (1992)

12 *Safety in the installation and use of gas systems and appliances — Approved Code of Practice and Guidance* Health and Safety Executive L56 (London: HSE Books) (1998)

13 The Building Regulations 2000 Approved Document J *Combustion appliances and fuel storage* (London: The Stationery Office) (2001)

14 *Bulk LPG storage at fixed installations — Part 1: Design, installation and operation of vessels located above ground* LPGA Code of Practice 1 (Ringwood: LP Gas Association) (1998)

15 Building Regulations 2000 Statutory Instrument 2000 No. 2531 (London: The Stationery Office) (2000)

16 BS 5410-1: 1997 *Code of practice for oil firing. Installations up to 45 kW output capacity for space heating and hot water supply purposes* (London: British Standards Institution) (1997)

17 Control of Pollution (Oil Storage) (England) Regulations 2001 Statutory Instrument 2001 No. 2954 (London: The Stationery Office)

18 BS 3797: 1990 *Specification for lightweight aggregates for masonry units and structural concrete* (London: British Standards Institution) (1990)

19 BS 799-5: 1987 *Oil burning equipment. Specification for oil storage tanks* (London: British Standards Institution) (1987)

Appendix AI : Example calculations

AI .I Sizing of water expansion vessel

It is required to determine the size of the sealed expansion vessel required if the pressure in the system is not to exceed 3.5 bar gauge (i.e. 350 kPa gauge).

Initial data:

- water volume of system = 450 litres
- design flow temperature = 60 °C
- design return temperature = 50 ° C
- height of system = 7.5 m
- design pump pressure rise = 42 kPa
- temperature of plant room during plant operation = 25 °C (i.e. 298 K)

Consider first the water. It may be assumed that, at its coolest, the temperature of the water in the system will be 4 °C (i.e. 277 K). Similarly, depending on the control system, the greatest expansion could occur at part load, when the entire water system is at the design flow temperature. Table 4.3 gives the expansion between 4 °C and 60 °C as 1.71%.

$$\Delta V \; 0.0171 \times 450 = 7.7 \text{ litres}$$

Ideally the pressure vessel should be connected to a position of low water pressure. This would reduce the required volume of a sealed vessel. However it is more convenient for items of plant to be located in close proximity, and in this example the expansion vessel is being connected to pipework 7.5 m below the highest position of the circuit. It must, however, be positioned on the return side of the pump, not the outlet.

The 'cold fill' pressure at the pump, due to the head of water, will partially compress the air within the expansion vessel, thus necessitating a larger expansion vessel. Thus it is advisable to pre-pressurise the air within the vessel to this pressure so that, once connected, it will still be full of air. Thus the initial air volume will be the same as the vessel volume. No further head of water should be applied as it would serve no useful purpose and would increase the operating pressure of the system, which is undesirable.

Pre-pressurisation required for a head of 7.5 m is given by:

$$P_1 = \rho g z \tag{A1.1}$$

where p_1 is the initial pressure (kPa), p is the density (kg.m-^{3}), g is the acceleration due to gravity (m.s-,) and z is the head (m).

Hence,

$$\begin{aligned} p_1 &= 1000 \times 9.81 \times 7.5 \\ &= 73.58 \text{ kPa gauge} \approx 174 \text{ kPa absolute} \end{aligned}$$

Maximum permissible pressure, p_2, at inlet to the pump:

$$\begin{aligned} p_2 &= (350 - 42) \text{ kPa} \\ &= 308 \text{ kPa gauge} \approx 408 \text{ kPa absolute} \end{aligned}$$

Initial volume of air in vessel = V_1; final volume of air = $V_2 = (V_1 - 7.7)$ litres.

For the air, the ideal gas equation will apply, using absolute values of temperature and pressure. Note that since no hot water flows through the vessel, there should be no effect upon the temperature of the air cushion within the vessel. However, it could be affected by the plant room temperature.

The ideal gas equation is:

$$\frac{p_2 V_2}{T_2} = \frac{p_1 V_1}{T_1} \tag{A1.2}$$

Therefore:

$$\frac{408\,(V_1 - 7.7)}{298} = \frac{174\, V_1}{277}$$

Hence, the minimum volume of expansion vessel required, V_1 = 14.23 litres.

As the calculation was carried out based on the maximum permissible pressure, the next size up must be selected. Since sealed pressure vessels constitute such a small portion of the equipment cost, consideration should always be given to selecting one which is larger than necessary, the advantage being a reduced operating pressure for the system.

AI.2 Effect of flow rate on radiator output

A radiator has a nominal output of 1.23 kW for water flow and return temperatures, t_1 and t_2, of 75 ° C and 65 ° C respectively in surroundings at 20 °C. It is required to determine the output when the flow rate is reduced to 40% of the design flow rate, q_1, the flow temperature remaining constant at 75 "C. The heat transfer index n has been found to be 1.25.

$$\begin{aligned} \Delta T &= \tfrac{1}{2}(t_1 + t_2) - t_{ai} \\ &= \tfrac{1}{2}(75 + 65) - 20 = 50 \text{ K} \end{aligned} \tag{A1.3}$$

Using Table 5.1, for mean radiator water temperature of 70 °C, c_p = 4.191 kJ.kg^{-1}.K^{-1}.

Rearranging equation 5.3 to give Km:

$$K_m = \phi \,/\, \Delta T^n \tag{A1.4}$$

where ø is the heat emission (kW).

Thus:

$$K_m = 1.23 / 501.25 = 9.251 \times 10^{-3} \text{kW.K}^{-1.25}$$

The design, or nominal flow rate, q_{mn} is obtained by rearranging equation 5.2:

$$q_{mn} = \frac{\phi_n}{c_p (t_1 - t_2)} \quad \text{(A1.5)}$$

Hence:

$$q_{mn} = \frac{1.23}{4.191\,(75 - 65)} = 0.02935 \text{ kg·s}^{-1}$$

For the situation with 40% of the nominal design flow:

$$q_m = 0.4 \times 0.02935 = 0.01174 \text{ kg.s}^{-1}$$

In order to use equation 5.8, it is necessary to calculate $(K_m / q_m c_p)$, i.e:

$$(K_m / q_m c_p) = (9.251 \times 10^{-3})/(0.01174 \times 4.191)$$
$$= 0.1880 \text{ K}^{-0.25}$$

Before using equation 5.8, a starting value for the outlet temperature, t_2, is required, for which an intelligent estimate is helpful. Clearly it is likely to be lower than the previous outlet temperature of 65 °C, but must be higher than the room temperature 20 °C. Therefore 55 °C is taken as an initial estimate for t_2 and inserted in the right hand side of the equation.

Therefore, using equation 5.8 and inserting the first estimate in the right hand side:

$$t_2 = t_1 - \frac{K_m}{q_m c_p}\left[\tfrac{1}{2}\left(t_1 + t_2\right) - t_{ai}\right]^n \quad \text{(A1.6)}$$

$$t_2 = 75 - 0.1880\,[{}^1/_2\,(75 + 55) - 20]^{1.25} = 53.09 \text{ °c}$$

The updated value is substituted successively into the right hand side until the required accuracy is obtained, i.e:

$$t_2 = 75 - 0.1880\,[{}^1/_2\,(75 + 53.09) - 20]^{1.25} = 53.67 \text{ °c}$$
$$t_2 = 75 - 0.1880\,[{}^1/_2\,(75 + 53.67) - 20]^{1.25} = 53.49 \text{ °c}$$
$$t_2 = 75 - 0.1880\,[{}^1/_2\,(75 + 53.49) - 20]^{1.25} = 53.55 \text{ °c}$$
$$t_2 = 75 - 0.1880\,[{}^1/_2\,(75 + 53.55) - 20]^{1.25} = 53.53 \text{ °c}$$
$$t_2 = 75 - 0.1880\,[{}^1/_2\,(75 + 53.53) - 20]^{1.25} = 53.53 \text{ °c}$$

Hence:

$$t_2 = 53.5 \text{°C}$$

Substituting in equation 5.2 gives:

$$\phi = q_m\, c_p\, (t_1 - t_2) \quad \text{(A1.7)}$$

Thus:

$$\phi = 0.01174 \times 4.191\,(75 - 53.5) = 1.058 \text{ kW}$$

Note that in this case, a reduction in the flow rate of 60% has only resulted in the heat emission reducing from the original 1.23 by 15%.

AI.3 Pipesizing

Figure A1.1 is a simplified schematic of a 2-pipe reverse-return system which, for simplicity, serves 10 emitters each required to have an output of 3.0 kW. The flow and return temperatures are to be 60 and 50 °C respectively. The design internal temperature is 20 °C, and the index of heat emission for the emitters, n, is 1.28.

Approximate distances for pipe runs:

- boiler to A = 10 m
- A to B = B to C etc. = 10 m
- E to F = 25 m
- J to boiler = 10 m

Additional components near the boiler but, for simplicity, not shown in the figure are:

- isolating valves: 4
- Y-pattern angle balancing valve: 1
- other elbows: 4
- tees, to expansion vessel and feed: 2

All elbows are assumed to be 90° with smooth radiussed inner surface.

The manual method for calculating the pressure drop around the circuit is tedious, so it is reasonable to consider the entire system to be at the mean water temperature of 55 °C. However in the following, the flow and return have been considered separately at their respective temperatures. The additional accuracy can be seen to be trivial. The pipework is all copper (BS 2871, Table X) for which pre-calculated pressure drops at 75 "C are given in Table 4.13 of Guide C (A1.1). Corrections for temperature are given herein, see Table 5.10.

Before commencing the pipe selection and pressure drop calculation, it is necessary to be sure of the emitter selection and design flow rates. Under the design criteria specified above, the excess temperature, ΔT, above the surrounding air temperature will be 55 – 20 = 35 K, rather than the nominal value of 50 K used in emitter manufacturers' catalogues. Therefore, it is necessary to calculate the nominal catalogue value required.

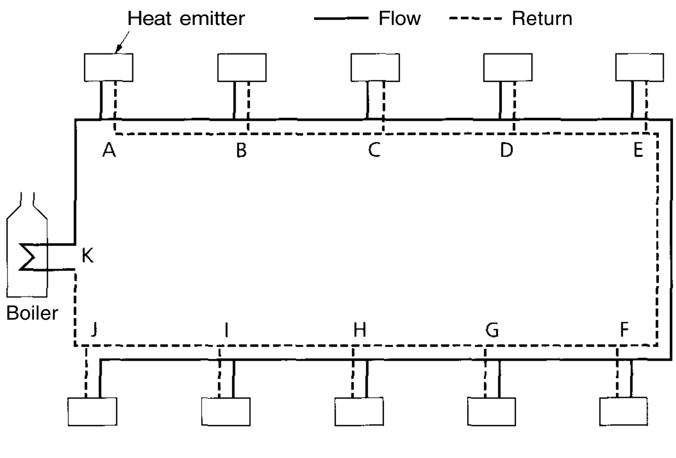

Figure A1.1 Simplified layout of the pipework of a heating system, pump and other ancillaries not shown (not to scale)

Re-arranging equation 5.5:

$$\phi_{50} = \phi_{35}\ (50 / 35)^n \qquad \text{(A1.8)}$$

Hence:

$$\phi_{50} = 3\ (50 / 35)^{1.28} = 4.736\ \text{kW}$$

Therefore, to obtain an output of 3.0 kW at $\Delta T = 35$ K, it is necessary to select a heat emitter giving a nominal output of 4.736 kW (at $\Delta T = 50$ K).

From Guide C, Table 4.A3.1, the properties of water are as follows:

— at 60 °C: $\rho = 983.2$ kg.m^{-3}

— at 55 °C: $\rho = 985.6$ kg.m^{-3}; $c_p = 4.184$ kJ.kg^{-1}·K^{-1}

— at 50 °C: $\rho = 988.0$ kg.m^{-3}.

The mass flow for each emitter is given by re-arranging equation 5.2:

$$q_m = \phi / c_p\ (t_1 - t_2) \qquad \text{(A1.9)}$$

Hence:

$$q_m = 3 / 4.184\ (60 - 50) = 0.0717\ \text{kg·s}^{-1}$$

For 10 emitters, the total flow Will be ($10\ q_m$) = 0.7170 kg·s^{-1}.

If some of the emitters are appreciably remote from one another such that heat losses from the flow pipe result in significantly different inlet water temperatures for the different emitters, the design flow rate for such emitters would need to be re-calculated.

To illustrate the sizing procedure, the following calculations show the flow along the flow pipe to the tee at D, and the return pipe from the tee at D to the boiler.

The designer has a free choice for the values of water velocity (c) and pipe diameter *(d)*. The smaller the pipe, the greater the pressure drop, and the greater the pumping power and energy consumption. Since the pressure drop is approximately inversely proportional to d^5, an increase in diameter from one size to the next size up can greatly reduce the friction pressure drop. Section 5.1.3 suggests a choice of velocity of about 1.0 mes-1. Traditionally, designers have constrained themselves by further limiting the pressure drop per unit length to a 'rule-of-thumb' figure of 300 Pa.m^{-1}. In reality, the choice of water velocity should depend on the length of pipework. In the calculations which follow, the starting point has been to choose a water velocity of 1.0 m.s-'. Where this results in pressure drops per unit length greater than 300 Pa·m^{-1}, the next pipe size up has been selected in order to reduce energy costs.

AI .3.1 Pipe sizing and pressure drops along pipes

The tables of pressure drops in Guide **C** (e.g. Guide C, Table 4.13) give a rough indication of velocity which aids the choice of pipe diameter. However, the tabulated velocities are not sufficiently accurate for subsequent calculations. The values of velocity (c) in Table A1.1 below have been calculated, as follows.

Typically, for pipe run A–B (row 2 of Table A1.1):

$$q_m = 0.717 - 0.0717 = 0.6453\ \text{kg·s}^{-1}$$

Guide C4, Table 4.13, shows that for a velocity less than 1.0 m.s–1, a minimum pipe diameter of 35 mm is required.

Guide C4, Table 4.2, gives the mean internal diameter (d_i) as 32.63 mm. Therefore the cross-sectional area of the pipe *(A)* is:

$$A = \pi d_i^2 / 4 = 8.362 \times 10^{-4}\ \text{m}^2$$

The water velocity is given by:

$$c = q_m / (\rho A) \qquad \text{(A1.10)}$$

Table Al.1 Calculation of pressure drops for straight pipework (flow from boiler to 'D' and return from 'D' to boiler)

Pipe run	q_m/kg·s^{-1}	l/m	d/mm	d_i/mm	c/m·s^{-1}	$(\Delta p/l)$ / Pa.m^{-1}	Δp/Pa
Flow (60°C):							
K-A	0.7170	10	35	32.63	0.8721	224	2240
A-B	0.6453	10	35	32.63	0.7849	185	1850
B-C	0.5736	10	35	32.63	0.6978	151	1510
C-D	0.5024	10	35*	32.63	0.6111	118	1180
							Total pressure drop: 6780
Temperature correction factor (Table 5.10): C = 1.058							Corrected total pressure drop: 7173
Return (50 °C):							
D-E	0.2868	10	28†	26.72	0.5540	125	1250
E-F	0.3586	25	28	26.72	0.6504	185	1850
F-G	0.4303	10	28	26.72	0.7767	257	2570
G-H	0.5024	10	35*	32.63	0.6081	118	1180
H-I	0.5736	10	35	32.63	0.6948	151	1510
I-J	0.6450	10	35	32.63	0.7807	185	1850
J-K	0.7170	10	35	32.63	0.8678	224	2240
							Total pressure drop: 12450
Temperature correction factor (Table 5.10): C = 1.104							Corrected total pressure drop: 13745

*Initial choice of 28mm (c = 0.94 m.s^{-1}) gives $(\Delta p/l)$ = 340 Pa.m^{-1}

†Initial choice of 22mm (c = 0.90 m.s^{-1}) gives $(\Delta p/l)$ = 434 Pa.m^{-1}

Hence:

$$c = 0.6453/(983.2 \times 8.362 \times 10^{-4}) = 0.7849\ \mathrm{m \cdot s^{-1}}$$

From Guide C, Table 4.13, for a water temperature of 75 °C, the pressure drop per unit length ($\Delta p/l$) is 185 Pa · m^{-1}. Hence, for a length of pipe of 10 m:

$$\Delta p 185 \times 10 = 1850\ \mathrm{Pa}$$

However, since the flow temperature is 60 °C, rather than 75 °C, a correction factor must be applied, see Table 5.10 of this Guide.

Table A1.1 shows the pipe sizing and pressure drop for each run of straight pipe for the flow and return taken by water supplying the heat emitter at 'D'.

The flow to each branch is 0.0717 kg . s^{-1}, requiring a pipe diameter of only 15 mm.

AI.3.2 Pressure drops due to fittings

Guide C, section 4.9, gives extensive data on values of pressure loss factors (ξ) for pipe fittings. These should be used in conjunction with equation 4.8 from Guide C, section 4, reprinted here as equation A1.11.

$$\Delta p = \xi \tfrac{1}{2} pc^2 \qquad \text{(A1.11)}$$

It should be noted that the pressure loss factors for tees, whether for the straight flow or the branch flow, are all to be used with the velocity pressure of the combined flow ($\xi \tfrac{1}{2} p_c^2$).

Since the value of ξ for tees depends upon both the relative branch flow (i.e. branch to combined) and the relative branch diameter (i.e. branch to combined), it is convenient to determine these ratios first.

Taking the supply tee at 'D' as an example (see Figure A1.2), subscript 'b' denotes flow in branch, subscript 'c' denotes combined flow upstream of branch and subscript 's' denotes 'straight' flow in pipe immediately downstream of the tee.

Hence:

$$q_{mb}/q_{mc} = 0.0717\ 10.5024 = 0.1427$$

$$d_b/d_c = 15/35 = 0.429$$

From Guide C, Table 4.47, for diverging flow:

$$\xi_{c\text{-}b} = 1.66$$

$$\xi_{c\text{-}s} = 0.67$$

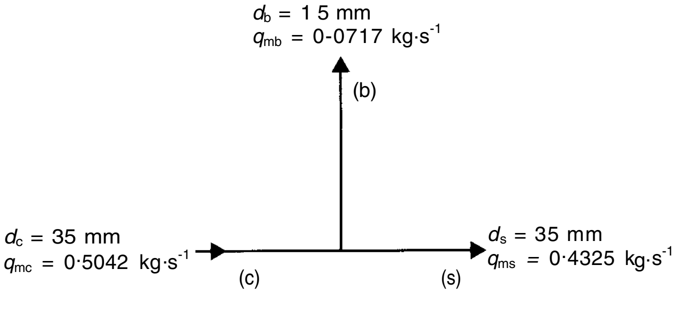

Figure A1.2 Schematic of supply tee at 'D'

From Table Al.1, above, for combined flow at inlet to the tee (i.e. pipe C–D):

$$c_c = 0.6111\ \mathrm{m.s^{-1}}$$

Therefore:

$$\tfrac{1}{2}\rho c_c^2 = 0.5 \times 988.0 \times 0.61112 = 184\ \mathrm{Pa}$$

Hence, pressure drop for the diverging tee at 'D' is given by:

$$\Delta\rho_{c\text{-}b} = \xi_{c\text{-}b}\ \tfrac{1}{2}\rho c_c^2$$

Hence:

$$\Delta\rho_{c\text{-}b} = 1.66 \times 184 = 305\ \mathrm{Pa}$$

The calculated pressure drops for all the fittings are given in Table A1.2.

Before the pressure drop for the entire circuit can be calculated, the pressure drop along each flow route would need to be determined. Some pipe sizes could then be modified to obtain a better balance.

Assuming for the moment that the flow route as used above gave the greatest pressure drop, balancing valves would be required on each of the other branches to equalise the pressure drops. Since balancing is an iterative process, the more inherently in-balance the system is, the better. Thus modifications in pipe sizes at the design stage can reduce the need for balancing.

In a two-pipe reverse return system of the type used for this example, it is not possible to foresee which flow route will produce the greatest pressure drop, i.e. which is the 'index' circuit, since this will depend on the pipe sizes chosen. The pump must be selected to provide a pressure rise equal to the pressure drop of the index circuit.

In the case above, supplying the heat emitter at 'D', the indication is that the total circuit pressure drop is the sum of the pressure drops itemised in Table A1.3.

For circuits requiring variable control via control valves, reasonable control is obtained only if the control valve has a reasonable value of authority, the typical value for which is 0.5. The implication of this is that as much pressure drop occurs across this open control valve as in the rest of the circuit. It is then worthwhile considering increasing all the pipe sizes, thereby reducing both the pressure drop around the circuit and that across the control valve.

AI .3.3 Alternative method for calculating pressure drop due to fittings

The method for calculating the pressure drop due to the fittings given in section A1.3.1 is the conventional method and is identical to that used for calculating pressure drops in ductwork due to ductwork fittings. However, the pre-calculated tables of pressure drops given in section 4 of Guide C offer an alternative method using the concept of the' equivalent length' (l_e) of a component having $\xi = 1$.

For example, for the pressure drop due to the diverting tee at 'D', the pressure loss factor has been determined as $\xi = 1.66$,

Table A1.2 Calculation of pressure drops for the pipework fittings (flow from boiler to 'D' and return from 'D' to boiler)

Item	Number	Guide C table no.	d/mm	d_b/d_c	q_b/q_c	ξ	c/m.s^{-1}	½ρ c_c^2/Pa	Δp/Pa
Gate valve	4	4.52	35	—	—	4 x 0.3	0.8723	374	449
Y-balancing	1	4.52	35	—	—	1 x 3.0	0.8723	374	1122
Elbow	4	4.48(a)	35	—	—	4 x 0.74	0.8723	374	1107
Tee, straight	2	4.47(c)	35	—	—	2 x 0.82	0.8723	374	613
						Total: 8.8	0.8723	374	3291
K–Dflow (60 °C):									
Elbow	2	4.48(a)	35	—	—	2 x 0.74	0.8723	374	326
Straight tee (A)	1	4.47(e)	35	0.417	0.1	0.7	0.8723	374	262
Straight tee (B)	1	4.47(e)	35	0.417	0.111	0.69	0.7849	303	209
Straight tee (C)	1	4.47(e)	35	0.417	0.125	0.68	0.6978	239	163
Diverging tee (D)	1	4.47(d)	28	0.429	0.143	1.66	0.6111	184	305
									Total: 4784
D–K return (50 "C):									
Converging tee (D)	1	4.47(a)	22	0.68	0.25	−0.09	0.9040	404	−36
Straight tee (E)	1	4.47(b)	28	0.519	0.20	0.64	0.6504	208	133
Elbow	2	4.48(a)	28	—	—	2 x 0.78	0.6504	208	324
Straight tee (F)	1	4.47(b)	28	0.519	0.20	0.64	0.7767	298	191
Straight tee (G)	1	4.47(b)	28	0.519	0.143	0.67	0.9417	438	293
Straight tee (H)	1	4.47(b)	35	0.417	0.125	0.68	0.6948	238	162
Straight tee (I)	1	4.47(b)	35	0.417	0.111	0.69	0.7807	301	208
Straight tee (J)	1	4.47(b)	35	0.417	0.10	0.70	0.8678	372	260
									Total: 1535

Note: there are no correction factors for temperature for the pressure drop due to fittings, though the density does affect the value ofipc²

Table A1.3 Pressure drops for flow serving heat emitter at 'D' for pipe sizes chosen above

Source of pressure drop	Pressure drop / Pa
Flow pipework	6780
Flow fittings	4784
Branch pipework	*
Return pipework	13 745
Flow pipework	1535
Boiler	*

* Obtained from emitter and boiler manufacturers

see Table A1.2. As an alternative to calculating the velocity pressure, consider instead the combined flow at entry to the tee, q_c = 0.5024 kgs^{-1}. The chosen pipe diameter is 35 mm. From Guide C, Table 4.13, the equivalent length, l_e, is either 1.5 or 1.6. (Note that this value is given to two significant figures only and is therefore somewhat crude.)

In terms of the equivalent length, the pressure drop is given by:

$$\Delta p = \xi\, l_e\, (\Delta p/l) \qquad \text{(A1.12)}$$

From Table A1.3 above, for the combined flow C–D, $(\Delta p/l)$ = 118 Pa·m^{-1}. Thus, taking l_e = 1.6:

$$\Delta p = 1.66 \times 1.6 \times 118 = 313 \text{ Pa·m}^{-1}$$

It has been noted that the pre-calculated tables of pressure drops given in Guide C are for a water temperature of 75 °C. The pressure drop in the pipework for water at 60 °C is 6% greater than that at 75 °C and this was taken into account in Table A1.3 and there is a temptation to apply a similar correction to the pressure drop for the fittings. However, the evaluation of l, was based on a value of ξ which does not vary with temperature and therefore no further correction needs to be made. The difference between the above value of 313 Pa.m–' and the value determined in Table A1.2 of 305 Pam^{-1} is due entirely to the tabulated values of l_e being quoted to only two significant figures.

It is likely that engineers using this method may add the hypothetical 'equivalent length' to a real length before calculating the pressure drop, and then apply a temperature correction to the combined result. Such an approach would be wrong.

Thus, the equivalent length method contains both a greater chance of error and an inherent inaccuracy. Bearing in mind the tendency to operate heating systems at temperatures lower than 75 ° C, it may be wise to use the conventional 'velocity pressure' method rather than the equivalent length method. Furthermore, the equivalent length method is of no use for fluids other than water or for pipes other than those for which values of equivalent length have been determined.

Reference

A1.1 *Reference data* CIBSE Guide C (London: Chartered Institution of Building Services Engineers) (2001)

Appendix A2: Sizing and height of chimneys and flues

A2.1 General considerations

The following information is needed for chimney and flue sizing:

(a) type of fuel used > its calorific value and the percentage sulphur content

(b) type and rated output of the boiler

(c) overall thermal efficiency of the boiler based on gross calorific value

(d) boiler flue gas outlet conditions at high and low fire, i.e. gas outlet temperatures and percentage carbon dioxide

(e) draught requirements at the boiler outlet at high and low fire

(f) height of the installation above sea level (gas volumes are increased by approximately 4% for every 300 m above sea level and allowance must be made in specifying volumes of forced and induced fans, etc., for installations at more than 600 m above sea level)

(g) location of plant and the character of surroundings, viz. topography, height of buildings surrounding plant, prevailing wind direction and velocities and the position of the boiler (i.e. basement or roof-top)

(h) winter and summer extremes of ambient temperature

(i) proposed general chimney construction to assess the cooling effect on gases.

Procedures for calculating chimney height are described below. For plants burning sulphur-bearing fuels where the full load SO_2 emission exceeds 0.38 $g.s^{-1}$, the chimney height is determined by the requirements of the 1993 Clean Air Act(A2.1)as interpreted by the third edition of the *Clean Air Act Memorandum: Chimney Heights*(A2.2). For smaller plants burning sulphur-bearing fuels, chimney heights are determined by combustion draught requirements with the proviso that such chimneys should terminate at least 3 m above the surrounding roof level or higher should the public health authority so require.

For gaseous fuels with negligible sulphur content, the method given seeks to limit the concentration of other combustion products (such as NO_2 and aldehydes) at ground level.

A2.2 Chimney heights for sulphur-bearing fuels

The maximum fuel burning rate at full plant loading is given by:

$$q_m = 100\phi / (\eta h_g) \qquad (A2.1)$$

where q_m is the maximum fuel burning rate ($kg.s^{-1}$), ϕ is the rated boiler output (kW), η is the thermal efficiency of the boiler (%) and h_g, is the calorific value of the fuel ($kJ.kg^{-1}$)

The maximum sulphur dioxide emission for fired equipment is:

$$E_m = K_1 q_m S \qquad (A2.2)$$

where E_m is the maximum sulphur dioxide emission ($g.s^{-1}$), S is the sulphur content of the fuel (%), K_1 is a constant (20 for oil firing, 18 for coal firing).

Equations A2.1 and A2.2 may be combined to give:

$$E_m = K_2 \phi / \eta \qquad (A2.3)$$

where K_2 is a factor representing the type of fuel, its calorific value and sulphur content. Values for K_2 are given in Table A2.1.

Where the sulphur dioxide emission does not exceed 0.38 $g.s^{-1}$ select the area and category from the following alternatives:

— A: undeveloped area where development is unlikely, where background pollution is low, and where there is no development within 800 m of the new chimney

— B: partially developed area with scattered houses, low background pollution and no other comparable industrial emissions within 400 m of the new chimney

— C: built-up residential area with only moderate background pollution and without other comparable industrial emissions

— D: urban area of mixed industrial and residential development, with considerable background pollution and with other comparable industrial emissions within 400 m of the new chimney

— E: large city, or an urban area of mixed heavy industrial and dense residential development, with severe background pollution.

Refer to Figure A2.1 to obtain the corrected chimney height, using the line matching the category selected above. For fuels with more than 2% sulphur content, add 10% to this height. If the height obtained is more than 2.5 times the height of the building or any building in the immediate vicinity, no further correction is required. Where this is not so, the final chimney height is obtained by substitution in the following formula:

$$H = (0.56\, h_a + 0.375\, h_b) + 0.625\, h_c \qquad (A2.4)$$

where H is the final chimney height (m), h_a is the building height or greatest length whichever is the lesser (m), h_b is the building height (m) and h_c is the uncorrected chimney height (m).

Table A2.1 Properties of fuels and values of K_2 for equation A2.3

Type of fuel	Properties: Calorific value / MJ·kg⁻¹	Properties: Sulphur content / %	K_2
Liquid fuels:			
— gas oil (class D)	45.5	1.0	43.9
— light fuel oil (class E)	43.4	3.2	147.5
— medium fuel oil (class F)	42.9	3.5	163.2
— heavy fuel oil (class C)	42.5	3.8	178.8
Solid fuels:			
— anthracite 101 and 102	30.0	1.1	66.0
— dry steam coal 201	30.5	1.1	64.9
— coking steam coal 202 and 204	30.7	1.1	64.5
— medium volatile coking coal 301a and 301b	30.5	1.3	76.7
— low volatile coal 200H	30.0	1.3	78.0
— very strongly caking coal 401	29.5	1.9	115.5
— strongly caking coal 501 and 502	29.4	1.9	116.3
— medium caking coal 601 and 602	27.6	1.9	123.9
— weakly caking coal 701 and 702	26.7	1.8	121.4
— very weakly caking coal 802	25.2	1.9	135.7
— non-caking coal 902	23.8	1.8	136.1

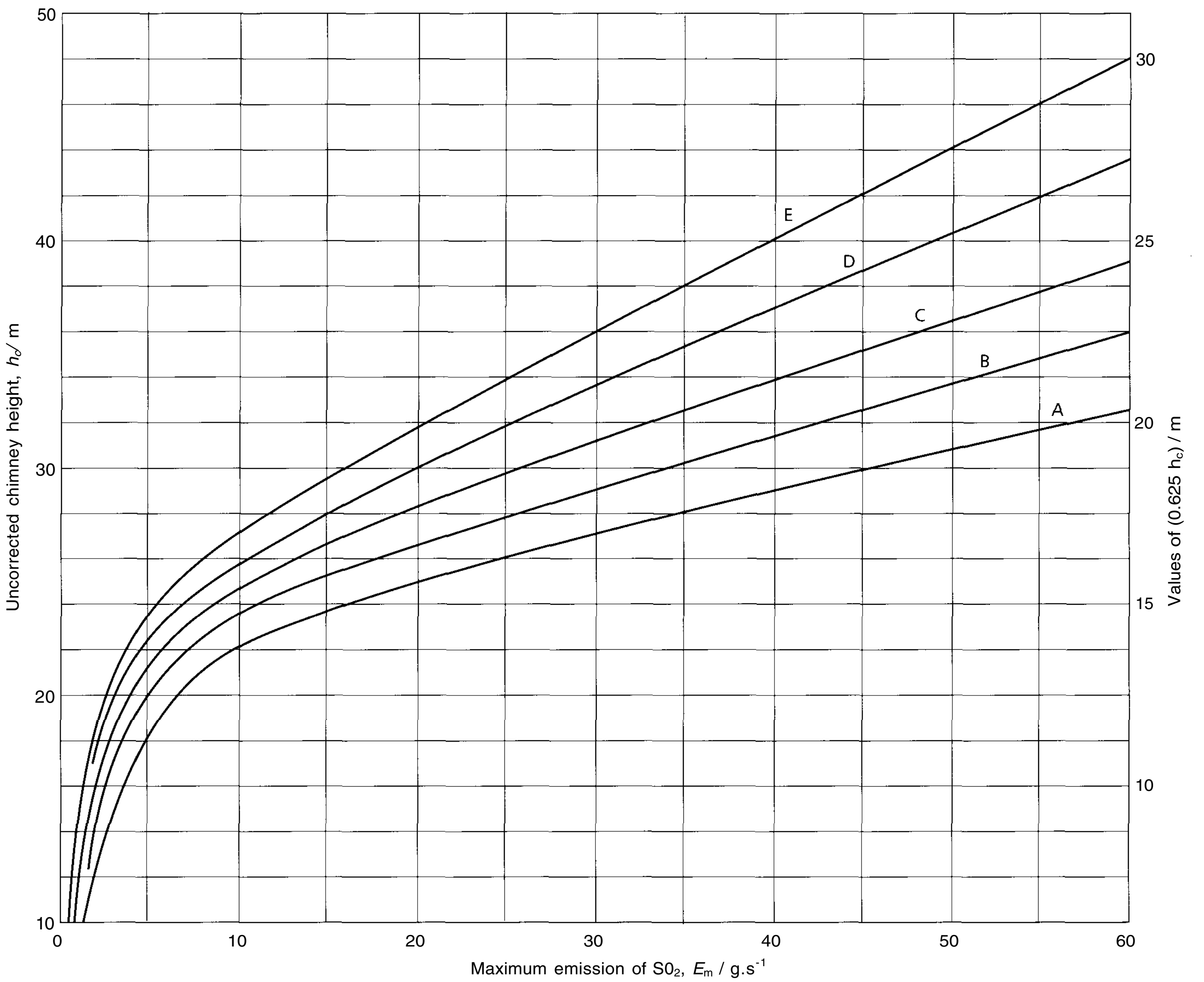

Figure A2.1 Uncorrected chimney heights

Table A2.2 Value of bracketed term in equation A2.4

Building height h_b / m	Value of $(0.56\,h_a + 0.375\,h_{,,})$ for stated building height or greatest length (h_a) / m																	
	9	12	15	18	21	24	27	30	33	36	39	42	45	48	51	54	57	60
9	8.4	10.1	11.8	13.5	15.1	16.8	18.5	20.2	21.9	23.5	25.2	26.9	28.6	30.3	31.9	33.6	35.3	37.0
12	9.5	11.2	12.9	14.6	16.3	17.9	19.6	21.3	23.0	24.7	26.3	28.0	29.7	31.4	33.1	34.7	36.4	38.1
15	10.7	12.3	14.0	15.7	17.4	19.1	20.7	22.4	24.1	25.8	27.5	29.1	30.8	32.5	34.2	35.9	37.5	39.2
18	11.8	13.5	15.2	16.8	18.5	20.2	21.9	23.6	25.2	26.9	28.6	30.3	32.0	33.6	35.3	37.0	38.7	40.4
21	12.9	14.6	16.3	18.0	19.6	21.3	23.0	24.7	26.4	28.0	29.7	31.4	33.1	34.8	36.4	38.1	39.8	41.5
24	14.0	15.7	17.4	19.1	20.8	22.4	24.1	25.8	27.5	29.2	30.8	32.5	34.2	35.9	37.6	39.2	40.9	42.6
27	15.2	16.8	18.5	20.2	21.9	23.6	25.2	26.9	28.6	30.3	32.0	33.6	35.3	37.0	38.7	40.4	42.0	43.7
30	16.3	18.0	19.7	21.3	23.0	24.7	26.4	28.1	29.7	31.4	33.1	34.8	36.5	38.1	39.8	41.5	43.2	44.9
33	17.4	19.1	20.8	22.5	24.1	25.8	27.5	29.2	30.9	32.5	34.2	35.9	37.6	39.3	40.9	42.6	44.3	46.0
36	18.5	20.2	21.9	23.6	25.3	26.9	28.6	30.3	32.0	33.7	35.3	37.0	38.7	40.4	42.1	43.7	45.4	47.1
39	19.7	21.3	23.0	24.7	26.4	28.1	29.7	31.4	33.1	34.8	36.5	38.1	39.8	41.5	43.2	44.9	46.5	48.2
42	20.8	22.5	24.2	25.8	27.5	29.2	30.9	32.6	34.2	35.9	37.6	39.3	41.0	42.6	44.3	46.0	47.7	49.4
45	21.9	23.6	25.3	27.0	28.6	30.3	32.0	33.7	35.4	37.0	38.7	40.4	42.1	43.8	45.4	47.1	48.8	50.5
48	23.0	24.7	26.4	28.1	29.8	31.4	33.1	34.8	36.5	38.2	39.8	41.5	43.2	44.9	46.6	48.2	49.9	51.6
51	24.2	25.8	27.5	29.2	30.9	32.6	34.2	35.9	37.6	39.3	41.0	42.6	44.3	46.0	47.7	49.4	51.0	52.7
54	25.3	27.0	28.7	30.3	32.0	33.7	35.4	37.1	38.7	40.4	42.1	43.8	45.5	47.1	48.8	50.5	52.2	53.9
57	26.4	28.1	29.8	31.5	33.1	34.8	36.5	38.2	39.9	41.5	43.2	44.9	46.6	48.3	49.9	51.6	53.3	55.0
60	27.5	29.2	30.9	32.6	34.3	35.9	37.6	39.3	41.0	42.7	44.3	46.0	47.7	49.4	51.1	52.7	54.4	56.1

Table A2.3 Trial flue gas velocities

Chimney height / m	Trial flue gas velocity / ms-'	
	Natural draught boilers	Boilers with pressurised combustion chambers
< 12	3.6	6.0
12 to 20	4.5	—
12 to 24	—	7.5
20 to 30	6.0	—
24 to 30	—	9.0
> 30	7.5	12.0

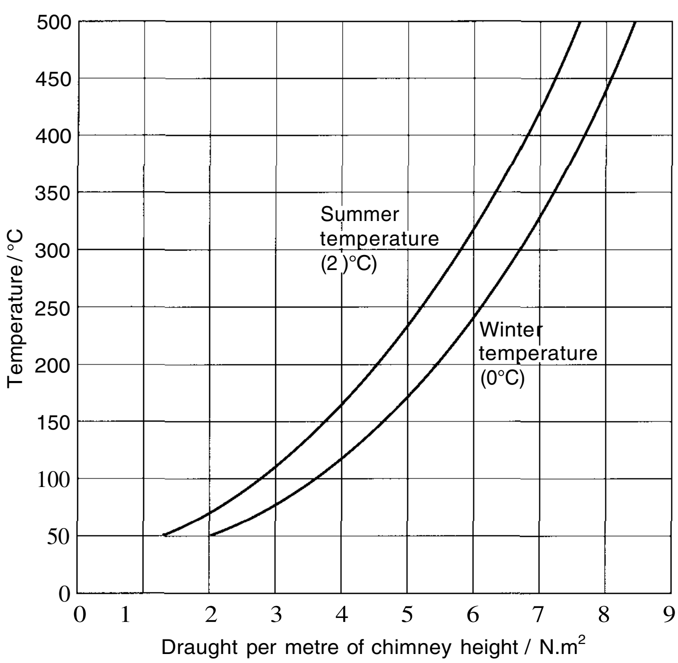

Figure A2.2 Chimney draught at 0 ° C and 20 "C ambient temperatures

Table A2.2 provides solutions to the term in brackets against known values of h_a and $h_{,}$. The final chimney height may then be obtained by adding this result to the appropriate value read from the scale on the right hand side of Figure A2.1. Note that 10 % must be added to this latter value for fuels with more than 2 % sulphur content.

Where the sulphur dioxide emission is less than 0.38 g.s -', the procedure is as follows:

(a) assess the height of buildings through which the chimney passes or to which it is attached

(b) add 3 m to this height to obtain the preliminary chimney height

(c) where the particular building is surrounded by higher buildings, the height of the latter must be taken into consideration as above

(d) select a trial flue gas velocity (from Table A2.3) and calculate the flue and chimney resistance

(e) compare this with the available chimney draught (Figure A2.2) and adjust the chimney height to suit, recalculating where necessary to give the highest possible efflux velocity from the chimney.

A2.3 Chimney heights for non-sulphur-bearing fuels

The following procedure should be followed:

(a) assess the boiler plant heat input rate

(b) for single free-standing chimneys, read the corres-'ponding chimney height from Figure A2.3

(c) for single chimneys passing through, or adjacent to buildings (the more usual case), the additional height may be read off from Figure A2.3 and added to the building height to give the final chimney height.

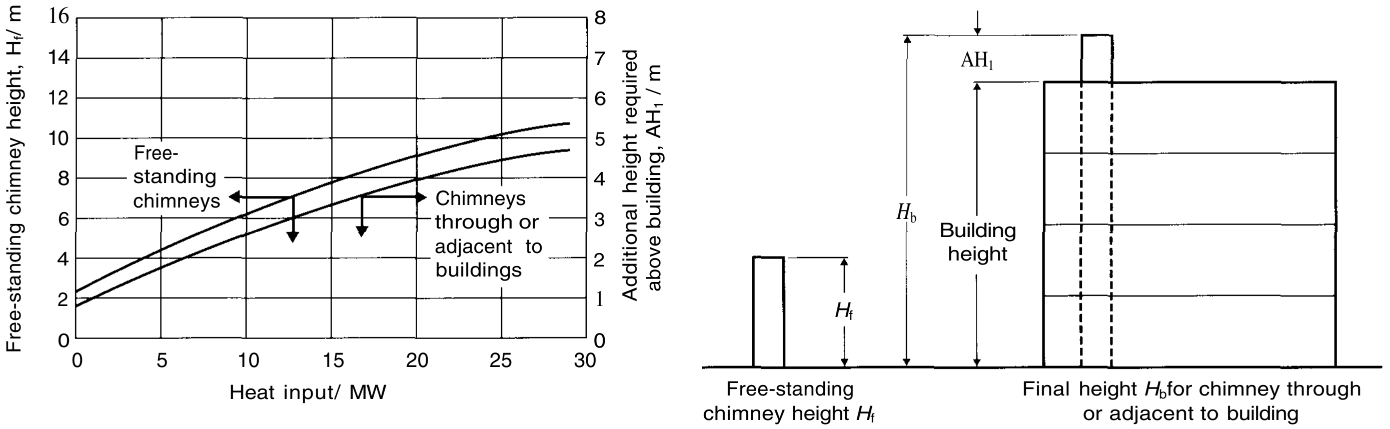

Figure A2.3 Heights for single chimneys

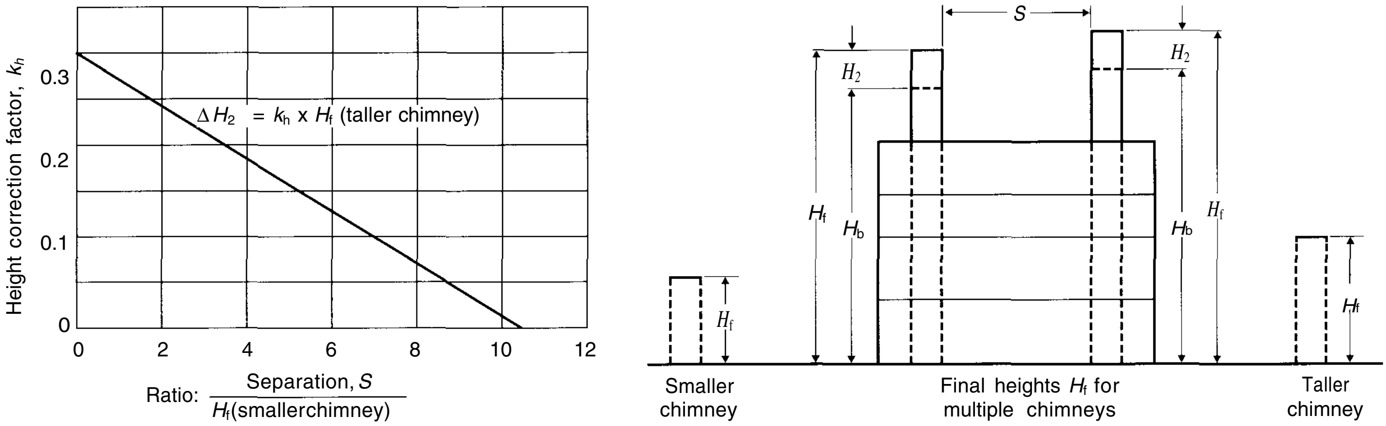

Figure A2.4 Heights for adjacent chimneys

Where two or more chimneys are in close proximity, the height of each should be increased slightly as follows:

(a) find the individual final chimney heights as before

(b) express the separation between a pair of chimneys as a multiple of the free-standing height of the smaller chimney

(c) read off the height correction factor from Figure A2.4; the required increase in height is then given by:

$$\Delta H_1 = k_h \times H_f \qquad (A2.5)$$

where ΔH_1 is the increase in height (m), k_h is the height correction factor and H_f is the free standing height of the taller chimney(m)

(d) repeat these steps for each pair of chimneys

(e) add the largest increase in height found to the final height of each chimney found as before.

Check that the height of each chimney provides the required combustion draught.

Example A2.1

Figure A2.5 shows a building 12 m high with 3 chimneys passing through it. If the heat inputs are 6 MW to A, 15 MW to B and **3** MW to ***C***, determine the chimney heights.

From Figure A2.3 (left axis), the free-standing heights are:

— chimney A: H_f = 4.6 m
— chimney B: H_f = 7.8 m
— chimney C: H_f = 3.4 m

From Figure A2.3 (right axis), the heights to be added to that of the building are:

— chimney A: ΔH_1 = 1.9 m
— chimney B: ΔH_1 = 3.3 m
— chimney C: ΔH_1 = 1.4 m

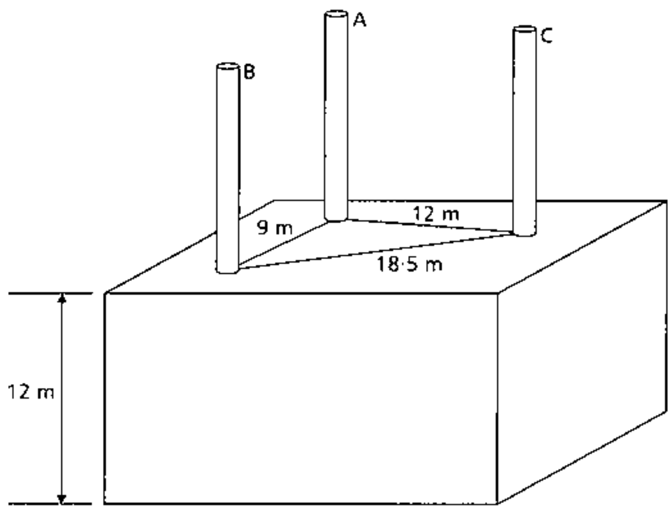

Figure A2.5 Diagram for example A2.1

Using the values obtained for free-standing heights and the separations obtained from Figure A2.5, the separations of pairs of chimneys expressed as a multiple of the free-standing height of the smaller chimney of each pair are AB = 2, BC = 5.4 and CA = 3.5.

From Figure A2.4, the height corrections for chimney proximity are:

— chimney A: $k_h = 0.24$

— chimney B: $k_h = 0.15$

— chimney C: $k_h = 0.21$

From equation A2.5, the required increases in height are 2.1, 1.2 and **1**, respectively, the largest of which must be added to the height of all three chimneys.

The final chimney heights are obtained by adding the height of the building, the additional heights above the building, ΔH_1 obtained from right-hand axis of Figure A2.3 and the height corrections for chimney proximity, k_h.

Hence, final chimney heights are:

— chimney A: $H = 12 + 1.9 + 2.1 = 16$ m

— chimney B: $H = 12 + 3.3 + 2.1 = 17.4$ m

— chimney C: $H = 12 + 1.4 + 2.1 = 15.5$ m

The calculated chimney height may be then be used to check the available chimney draught at various flue gas temperatures by reference to Figure A2.2.

A2.4 Determination of flue/chimney area

The area must be selected to provide the highest possible flue gas velocity and the smallest cooling area, bearing in mind the available draught and frictional resistance of the flue and chimney considered. In order to avoid down-wash, a chimney efflux velocity of approximately 7.5 to 9 m·s-' is required, but this cannot always be achieved on natural draught plant. The procedure is as follows:

(a) Calculate the flue gas volume flow rates to be handled at full- and low-fire conditions at the temperatures involved at the particular boiler outlet.

(b) Select a flue gas velocity which appears reasonable for the plant considered (see Table A2.3) and obtain the area equivalent from:

$$A = q_v / v \qquad \text{(A2.6)}$$

where A is the area equivalent (Table A2.4) (m2), q_v is the flue gas volume flow rate at full fire ($m^3 \cdot s^{-1}$) and v is the flue gas velocity (m-1).

(C) Calculate the resistance to flow of a flue chimney, as follows. The pressure drop is calculated in the same manner as for ductwork. The total pressure drop is the sum of the pressure drop of the fittings and for the straight lengths of ductwork. For each fitting of the flue an additional pressure drop is given by:

$$\Delta p_1 = \xi \tfrac{1}{2} \rho v^2 \qquad \text{(A2.7)}$$

where Δp_1 is the pressure drop due to each ductwork fitting (Pa), ξ is the pressure loss factor for the fitting, ρ is the density of the flue gases ($kg.m^{-1}$). Velocity pressure for the flue gases at different temperatures and velocities are given in Table A2.5.

For the straight lengths of ductwork there are no pre-calculated values. Thus equation 4.7 from section 4 of Guide C(A2.3) is used, as follows:

$$\Delta p_2 = \lambda (l/d_k) \tfrac{1}{2} \rho v^2 \qquad \text{(A2.8)}$$

where Δp_2 is the pressure drop due to a straight length of ductwork (Pa), A is the friction factor, l is the length of the straight ductwork (m), d_h is the hydraulic diameter (m) and v is the mean gas velocity ($m.s^{-1}$).

Table A2.4 Areas of various chimney sections

Circular and square sections			Rectangular and elliptical sections								
a/m	Area/m²		(a x b)	Area/m²		(a x b)	Area/m²		(a x b)	Area/m²	
	Circle	Square		Ellipse	Rectangle		Ellipse	Rectangle		Ellipse	Rectangle
0.3	0.07	0.09	0.3 x 0.4	0.09	0.12	0.9 x 1.0	0.71	0.90	1.5 x 1.6	1.88	2.40
0.4	0.13	0.16	0.3 x 0.5	0.12	0.15	0.9 x 1.1	0.78	0.99	1.5 x 1.7	2.00	2.55
0.5	0.20	0.25	0.3 x 0.6	0.14	0.18	0.9 x 1.2	0.85	1.08	1.5 x 1.8	2.12	2.70
0.6	0.28	0.36	0.4 x 0.5	0.16	0.20	1.0 x 1.1	0.86	1.10	1.6 x 1.7	2.13	2.72
0.7	0.39	0.49	0.4 x 0.6	0.19	0.24	1.0 x 1.2	0.94	1.20	1.6 x 1.8	2.26	2.88
0.8	0.50	0.64	0.4 x 0.7	0.22	0.28	1.0 x 1.3	1.02	1.30	1.6 x 1.9	2.39	3.04
0.9	0.64	0.81	0.5 x 0.6	0.24	0.30	1.1 x 1.2	1.04	1.32	1.7 x 1.8	2.40	3.06
1.0	0.79	1.00	0.5 x 0.7	0.27	0.35	1.1 x 1.3	1.12	1.43	1.7 x 1.9	2.53	3.23
1.1	0.95	1.21	0.5 x 0.8	0.31	0.40	1.1 x 1.4	1.21	1.54	1.7 x 2.0	2.67	3.40
1.2	1.12	1.44	0.6 x 0.7	0.33	0.42	1.2 x 1.3	1.22	1.56	1.8 x 1.9	2.68	3.42
1.3	1.33	1.69.	0.6 x 0.8	0.38	0.48	1.2 x 1.4	1.32	1.68	1.8 x 2.0	2.83	3.60
1.4	1.54	1.96	0.6 x 0.9	0.42	0.54	1.2 x 1.5	1.41	1.80	1.8 x 2.1	2.97	3.78
1.5	1.77	2.25	0.7 x 0.8	0.44	0.56	1.3 x 1.4	1.43	1.82	1.9 x 2.0	2.98	3.80
1.6	2.01	2.56	0.7 x 0.9	0.49	0.63	1.3 x 1.5	1.53	1.95	1.9 x 2.1	3.13	3.99
1.7	2.27	2.89	0.7 x 1.0	0.55	0.70	1.3 x 1.6	1.63	2.08	1.9 x 2.2	3.28	4.18
1.8	2.54	3.24	0.8 x 0.9	0.57	0.72	1.4 x 1.5	1.65	2.10	2.0 x 2.1	3.30	4.20
1.9	2.83	3.61	0.8 x 1.0	0.63	0.80	1.4 x 1.6	1.76	2.24	2.0 x 2.2	3.45	4.40
2.0	3.14	4.00	0.9 x 1.1	0.69	0.88	1.4 x 1.7	1.87	2.38	2.0 x 2.3	3.61	4.60

Table A2.5 Velocity pressure ($1/2\ \rho v^2$) of flue gases at different temperatures

Velocity (m/s)	Velocity pressure ($\frac{1}{2} \rho v^2$) / Pa at stated temperature / °C									
	50	100	150	200	250	300	350	400	450	500
3	4.9	4.2	3.7	3.4	3.0	2.8	2.5	2.4	2.2	2.0
4	8.7	7.5	6.7	6.0	5.4	4.9	4.5	4.2	3.9	3.6
5	13.7	11.8	10.4	9.3	8.4	7.7	7.1	6.5	6.1	5.7
6	19.7	17.0	15.0	13.4	12.1	11.1	10.2	9.4	8.7	8.3
7	26.8	23.1	20.4	18.2	16.5	15.1	13.8	12.8	11.9	11.1
8	35.0	30.2	26.6	23.8	21.5	19.6	18.1	16.7	15.6	14.6
9	44.3	38.2	33.8	30.2	27.3	24.9	22.9	21.2	19.7	18.4
10	54.5	47.1	41.7	37.2	33.6	30.8	28.2	26.1	24.3	22.8
11	66.0	56.9	50.4	45.0	40.6	37.2	34.1	31.6	29.4	27.5
12	78.5	67.8	60.0	53.5	48.4	44.1	40.5	37.6	35.0	32.8
13	92.2	79.6	70.4	62.9	56.9	51.9	47.6	44.1	41.1	38.4
14	107	92.4	81.7	73.0	65.9	60.3	55.4	51.2	47.6	44.6
15	123	106	93.5	83.8	75.6	69.2	63.5	58.8	54.6	51.2
16	139	121	107	95.2	86.0	78.8	72.3	66.9	62.1	58.1
17	158	136	121	107	97.1	88.7	81.4	75.4	70.3	65.6
18	177	153	135	121	109	99.5	91.5	84.4	78.8	73.8

Table A2.6 Values of duct friction factor λ for flue gases (valid for flue gas temperatures of 180–340 °C)

Flue hydraulic diameter / mm	Mean gas velocity/m.s^{-1}	Smooth concrete or welded steel	Riveted steel or smooth cement pargeting	Brick or rough cement pargeting
150	1.5	0.0216	0.0276	0.0760
	3.0	0.0192	0.0264	0.0748
	4.5	0.0180	0.0256	0.0740
	≥ 6.0	0.0176	0.0252	0.0720
230	1.5	0.0188	0.0232	0.0720
	3.0	0.0172	0.0226	0.0704
	4.5	0.0160	0.0224	0.0684
	≥ 6.0	0.0156	0.0222	0.0672
305	1.5	0.0168	0.0216	0.0584
	3.0	0.0156	0.0204	0.0580
	4.5	0.0148	0.0200	0.0576
	≥ 6.0	0.0144	0.0196	0.0572
355	1.5	0.0160	0.0200	0.0552
	3.0	0.0146	0.0196	0.0540
	4.5	0.0140	0.0192	0.0520
	≥ 6.0	0.0136	0.0184	0.0520
460	1.5	0.0146	0.0188	0.0520
	3.0	0.0140	0.0180	0.0512
	4.5	0.0132	0.0176	0.0506
	≥ 6.0	0.0126	0.0172	0.0506
610	1.5	0.0144	0.0172	0.0500
	3.0	0.0132	0.0160	0.0492
	4.5	0.0124	0.0156	0.0480
	≥ 6.0	0.0120	0.0152	0.0476
1220	1.5	0.0120	0.0140	0.0444
	3.0	0.0108	0.0136	0.0440
	4.5	0.0104	0.0132	0.0436
	≥ 6.0	0.0100	0.0128	0.0432
1830	1.5	0.0108	0.0124	0.0404
	3.0	0.0100	0.0116	0.0392
	4.5	0.0096	0.0112	0.0388
	≥ 6.0	0.0092	0.0108	0.0380
1830	1.5	0.0108	0.0124	0.0404
	3.0	0.0100	0.0116	0.0392
	4.5	0.0096	0.0112	0.0388
	≥ 6.0	0.0092	0.0108	0.0380

The total pressure drop is then given by:

$$\Delta p_t = \Sigma \Delta p_1 + \Sigma \Delta p_2 + \Delta p_d \quad \text{(A2.9)}$$

where Δp_t is the total pressure drop (Pa) and Δp_d is the draught required at the boiler (Pa).

Some values of duct friction factors λ, are given in Table A2.6 for a limited range of temperatures. Should there be a need to calculate the value of λ from first principles using Guide C[A2.3], knowledge of density p and viscosity η would be required in order to establish a value of Reynolds number, R_e. These should not differ significantly for the different flue gases resulting from combustion of different fuels as the dominant constituent is nitrogen. Table A2.7 gives values of flue gas density. In the absence of other information, the values of viscosity could be taken as those for nitrogen, to be found in Guide C[A2.3], Appendix 4.A1, Table 4.A1.5.

Table A2.7 Flue gas density at various temperatures

Temperature /°C	Density, ρ / kg m^{-3}
50	1.09
100	0.942
150	0.834
200	0.744
250	0.672
300	0.616
350	0.564
400	0.521
450	0.486
500	0.456

The total resistance obtained from equation A2.9 must be compared with the available chimney draught. If the residual chimney draught is excessive, the flue areas can be recalculated using a higher flue gas velocity or a nozzle can be fitted to the chimney to take up the excessive draught by providing for increased efflux velocity. Downwash of gases and inversion can occur at low velocities and a minimum efflux velocity of 7.5 m.s^{-1} will obviate these problems in general. Such a velocity may not be possible on small natural draught plants with non-pressurised combustion chambers as the above calculations will demonstrate. In such cases the maximum practicable efflux velocity should be sought.

Where high velocities are required, or where the flue run gives high resistance, the use of increased forced draught fan power and/or induced draught fans must be considered. The calculations are performed in a similar manner to ascertain the fan duties required to overcome the flue system resistances involved at the selected velocity.

The resistance to gas flow on low-fire should be assessed and compared with the available chimney draught. There may be excessive suction at the boiler outlet on low fire resulting in the need for control dampers/draught controllers where the values fall outside the manufacturers' stated limits.

Under high velocity flue conditions, the flues and chimneys will probably be under pressurised conditions. Extra care must be taken in flue/chimney construction where such running conditions are required and it is not good practice to pressurise flues/chimneys of brick construction.

References

A2.1 Clean Air Act (1993) (c. 11) (London: Her Majesty's Stationery Office) (1993)

A2.2 Chimney Heights - Third edition of the 1956 Clean Air Act Memorandum (London: Her Majesty's Stationery Office) (1981)

A2.3 Reference data CIBSE Guide C (London: Chartered Institution of Building Services Engineers) (2001)

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