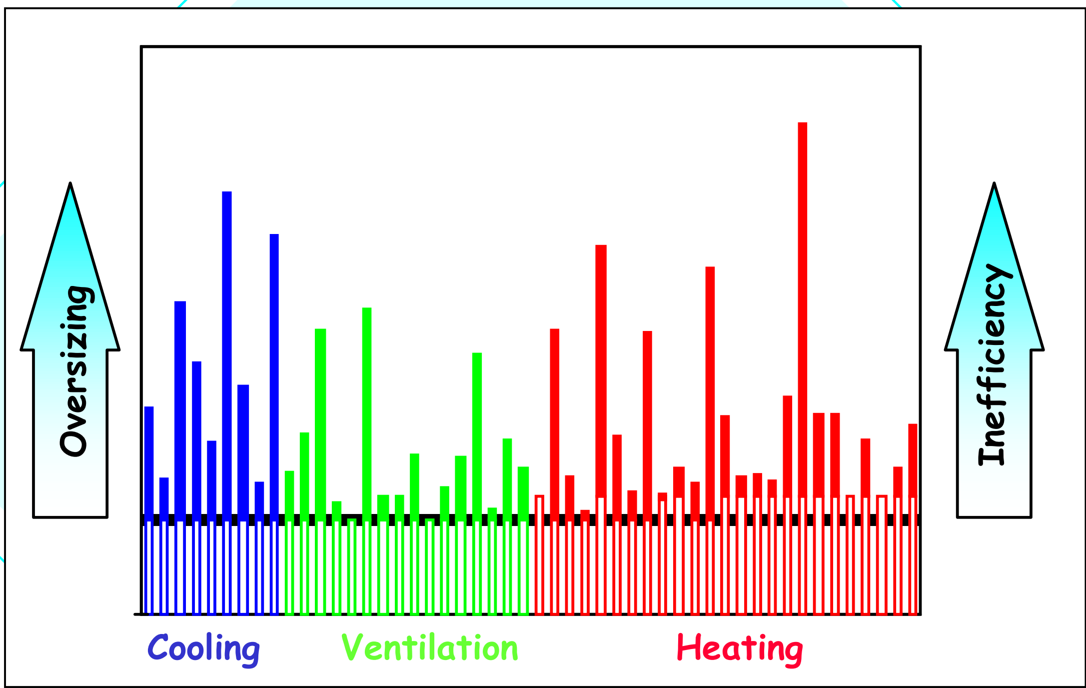
Application Guide AG 1/2000

# ENHANCING THE PERFORMANCE OF OVERSIZED PLANT

Barry Crozier




DETR
Environment
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BSRIA

Application Guide AG 1/2000

# ENHANCING THE PERFORMANCE OF OVERSIZED PLANT

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Gatwick Airport

Acknowledgement is also given to the original BSRIA publication by J. Brittain on plant oversizing on which this research has been based. Some of the original assessment methodology has been reproduced here to aid understanding.

This publication is issued with the agreement of the DETR and every opportunity has been taken to incorporate the views of the editorial panel, but final editorial control of this document rests with BSRIA.

# EXECUTIVE SUMMARY

This guidance document provides an examination of the issues relating to oversized heating, ventilation and air conditioning (HVAC) plant. It also presents a methodology for identifying where plant is oversized and for assessing the extent of this. Recommendations for remedial measures are suggested, plant monitoring tips are provided and case studies are also included. In addition to assisting facilities managers in their continuous quest to reduce energy expenditure, the guidance will also prove an invaluable source of information for those considering plant refurbishment. The methodology can be used to assist in establishing the appropriate plant requirements for the building before any refurbishment is implemented. Monitoring plant and systems for the oversizing surveys can also establish faults or control anomalies which may otherwise have gone unnoticed. Designers of HVAC systems will also find the document useful. By highlighting the origins of oversizing, the document aims to make designers more aware of their responsibility when it comes to sizing HVAC plant and systems. In particular, designers are requested to minimise the use of design margins by closer evaluation of their validity.

The main steps to reduce the energy consumption of oversized plant described in this document are:

- Target plant oversizing by considering basic indicators
- Assess and optimise operation and maintenance practice
- Accurately quantify plant overcapacity by using plant monitoring
- Implement remedial measures in order to improve system performance.

In general, it is worth investigating plant for oversizing if:

1. CAV fans consume at least 6 kWe (operating throughout the year).
2. VAV fans use a capacity control technique other than a speed controlled electrical drive system and consume at least 10 kWe (operating throughout the year).
3. Heating/cooling performance is poor at low load.
4. Boiler plant is made up of old/inefficient or conventional boiler types (capacity greater than 400 kW) that are not sequence controlled.
5. All steam boilers are continuously primed.
6. Domestic hot water calorifiers are poorly insulated (capacity greater than 300 litres).
7. Chiller plant is not sequence-controlled or consists of large chillers (chiller compressor power rating greater than 50 kW).
8. Pumps consume at least 10 kWe (operating through the heating or cooling season).

Investigating oversized plant can result in significant energy and financial savings:

- A constant air volume system with 15 kWe fans may typically cost £6,000 a year to run. If the fans generate 20% more flow than required, £2,500/year could possibly be saved by simply derating system flow rate.

- The fuel for a boiler installation heating a building floor area of 5,000 $m^2$ may typically cost £10,000 a year. If the boiler plant is 20 years old and oversized by a factor of 2.5, £3,500/year could possibly be saved by improving boiler control.
- Chiller plant providing comfort cooling for an area 5,000 $m^2$ may typically cost £20,000 a year in energy. If the plant is oversized a factor of 2, £4,000/year may possibly be saved by derating chiller capacity.

Remedial measures are described for each plant system. Typical measures suggested in this document include:

- downsizing and de-rating plant
- isolation of superfluous plant
- adding loads to systems
- plant replacement
- system balancing
- improved control techniques and strategies.

If plant utilises good capacity control measures, then there may be little benefit in monitoring plant unless it is particularly inefficient and being considered for replacement.

This guidance recommends targeting air handling plant before other systems since thermal load depend on air handling system performance.

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# LIST OF ABBREVIATIONS

| | |
|---|---|
| **BL** | Base load |
| **CAV** | Constant air volume |
| **CS** | Control state |
| **DHW** | Domestic hot water |
| **$D_t$** | Damper position |
| **FCU** | Fan coil unit |
| **HVAC** | Heating, ventilation and air conditioning |
| **IAT** | Internal air temperature |
| **KWE** | Kilowatt (electrical power) |
| **OAT** | Outside air temperature |
| **OF** | Oversizing factor |
| **$OF_D$** | Adjusted oversizing factor |
| **$OF_{HS}$** | Heating system oversizing factor |
| **P** | Power |
| **PB** | Proportional band |
| **$P_e$** | Extract fan power consumption |
| **PPM** | Parts per million |
| **$P_s$** | Supply fan power consumption |
| **$P_{sm}$** | Steam main pressure |
| **$P_t$** | Pump total pressure |
| **Q** | Flow rate |
| **$Q_e$** | Extract flow rate |
| **$Q_p$** | Evaporation rate |
| **$Q_s$** | Supply flow rate |
| **R** | Reference condition |
| **$SDT_o$** | Outside air temperature |

| | |
|---|---|
| **SP** | Setpoint |
| **$T_f$** | Flow temperature |
| **$T_i$** | Inside air temperature |
| **$T_o$** | Outside air temperature |
| **$T_r$** | Return temperature |
| **TU** | Terminal unit |
| **VAV** | Variable air volume |
| **$V_c$** | Cooling valve position |
| **$V_h$** | Heating valve position |
| **$V_i$** | Index terminal unit control valve position |
| **VSD** | Variable speed drive |
| **W** | Proportion of weather related load |

# 1 INTRODUCTION

Building owners and operators endeavour to provide satisfactory comfort conditions at minimum operating cost. However, this objective can be severely hampered by the significant amount of oversized plant installed in the UK.

## 1.1 Oversizing factors

This application guide refers to excess capacity in heating, cooling and ventilation systems in terms of oversizing factors (OFs). Plant with an OF of 1 can be defined as plant with the 'ideal' capacity sized for steady state design conditions. Any OF above the selected 'ideal' will imply overcapacity in the system. For example, in a continuously operated heating system, an OF of 2.5 would imply that the plant has a superfluous margin of 150% above the selected 'ideal' of 1. This does not allow for any margins, or stand-by capacity, or preheating requirements for intermittently operated heating systems. Any additional requirement for preheat gives an 'ideal' OF greater than 1.

It should be remembered that plant sized for steady state design conditions will have excess capacity when the outside conditions are less severe than the design day and therefore margins will exist for the majority of the time.

## 1.2 Extent of oversized plant

Fifty HVAC systems in the UK have been monitored and analysed using the methodology set out in this guidance and the extent of oversizing established. It was found that 80% of the heating plant, 88% of the ventilation plant and 100% of the chiller plant incorporated capacity above that necessary to meet design requirements. A summary of the findings is highlighted in Figure 1 and more information may be found in Appendix A. A maximum 25% margin has been allowed for preheating requirements of the intermittently operated heating systems.

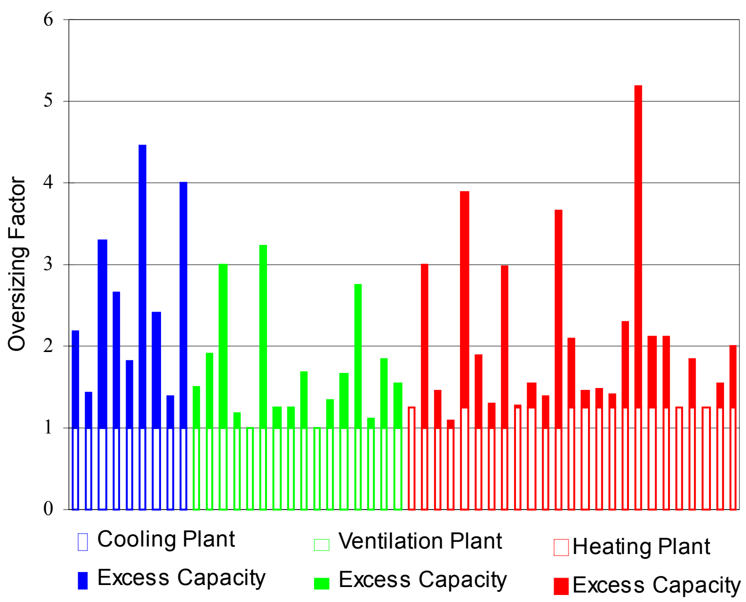



**Figure 1**
Summary of oversizing in 50 HVAC systems

# 2 HOW TO USE THIS DOCUMENT

The document has 4 main parts:

- An overview of oversizing and the implications for system operation. This section incorporates the findings from the oversizing and energy saving surveys including case study examples (Section 3).
- System sections providing procedures for identifying excess capacity and information on practical remedial measures that can be adopted for air handling (Section 4), cooling (Section 5), pumping (Section 6) and heating systems (Section 7).
- Guidance on plant monitoring and assessment procedures (Section 8).
- A summary of 50 oversizing surveys and 9 energy saving surveys (Appendix A).

It is recommended to study Section 3 to gain an insight to the issues concerned with oversizing before progressing to any of the other sections. Once this is complete, choose the section relevant to the type of plant you want to look at (Sections 4, 5, 6 or 7). It is recommended that air handling plant (Section 4) is targeted first as thermal load will be affected by air handling performance.

Within the four main plant sections there are details on -

- Specific implications of oversizing for specific types of plant
- Basic assessment - a simple method to give an initial indication of whether a detailed assessment may be worthwhile
- Detailed monitoring and assessment procedures - including case studies
- Remedial measures - what can be done and the savings that can be achieved.

The assessments and the calculations required for the basic and detailed assessments are displayed in a boxed table method to simplify the process. Figure 2 illustrates how these boxes are displayed.

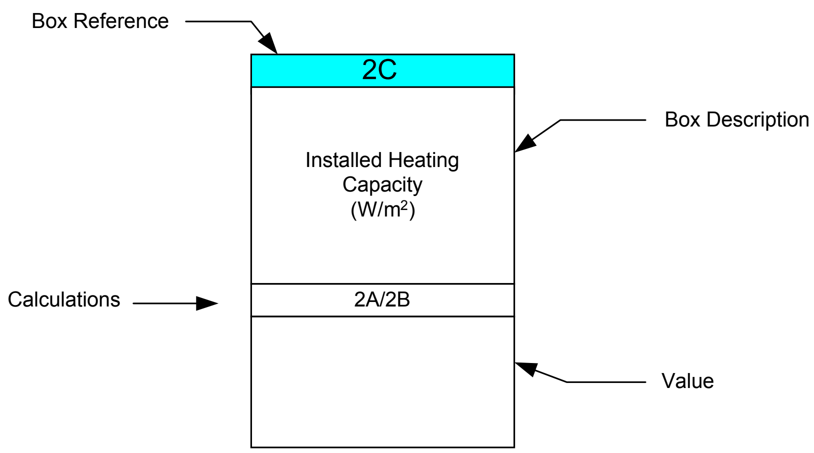



**Figure 2**
Example boxed calculation table

Where calculations are given they are based on box references.

Pages containing these boxed calculations are intended for photocopying before use.  This leaves the originals in the document as 'clean' for re-use.  Any photocopies made must be for use by the document owner only, otherwise copyright will be infringed.

Some of the calculations are quite lengthy and so alternatively the formula can be used in a computer spreadsheet.  This way the document originals are left clean and any future studies can be easily calculated using the spreadsheets.

# 3 OVERSIZING

## 3.1 Origins of oversizing

Oversized plant is defined as plant with maximum operational duty greater than that actually required.

It is generally accepted that the design process promotes oversizing. Systems are usually designed to provide and maintain comfortable internal conditions no matter what outside conditions prevail. Therefore the capacity requirements of the plant are dictated by the need for the system to cope with design peak outside conditions. Hence systems automatically have excess capacity and operate under part load when the external temperature is above or below the design condition (for cooling and heating respectively).

The problem is further exacerbated through the use of design margins, limitations of manufacturers' stock sizing, or change in building use, and more often than not a combination of all of these, as seen by the following:

**1 The design process**

- ⇒ unnecessarily restrictive design parameters
- ⇒ design over-specification
- ⇒ poorly designed capacity control
- ⇒ plant selection (use of next size up)
- ⇒ use of safety margins
- ⇒ increasing use of software design packages without competent knowledge of the calculations and margins they use
- ⇒ security against plant break down and maintenance
- ⇒ overestimated internal gains, particularly with reference to office equipment
- ⇒ subsequent reductions in plant loads, for example, from
  - ♦ increased building insulation
  - ♦ change in building use.

**2 Management**

- ⇒ inadequate commissioning
- ⇒ poorly managed systems
- ⇒ inappropriate control strategies.

### Design margins

Design margins can be defined as 'any percentage change to a design value, parameter or calculation result whether a deliberate and valid design decision, a contingency or safety factor or an inadvertent addition'[1]. The use of margins without adequate justification can be summarised as poor design and is certainly not recognised as 'good design practice'. Given the current interest in value engineering and the emphasis in the Latham report[2] on design responsibilities and best practice it is important that designers should minimise the need for the addition of margins by closer evaluation of their validity.

| | |
|---|---|
| ***Ventilation Plant Margins*** | Margins to allow for installation variations tend to vary between engineers. UK engineers typically add 10% to the calculated fan total pressure[3]. |
| ***Cooling Plant Margins*** | CIBSE Guide A[4] advises that the maximum coincident gain should be the basis of plant sizing. However, if accurate control of room temperature is required it is advised that some additional capacity may be needed. |
| ***Heating Plant Margins*** | There are certain cases where plant oversizing is required for satisfactory plant operation such as boiler margins for pre-heating intermittently heated buildings. Boilers sized for steady state heat losses can be operated intermittently for most of the time and the increased margin may be justified by a decrease in energy consumption. The CIBSE Guide B[5] recommends that a balance is struck between the extra cost of oversizing the boiler and the expected energy savings achieved through intermittent operation. Determining the optimum boiler capacity required for an intermittent heating strategy can be complicated. There are a number of factors to be considered such as type of heating system, the building's thermal time constant and the rate of heat loss. Plant oversizing in excess of 25% of steady design requirements is unlikely to be justified and operation at low loads will lead to corrosion and loss in efficiency. |

## 3.2 Implications of Plant Oversizing

The initial cost of oversizing plant appears at the design stage. Generally, fans and pumps are oversized by at least 15%, with more generous overcapacity often given to boilers and chillers. Bigger plant requires more space allocation and costs more money to purchase. Oversized plant is then likely to consume more energy, with problems associated with poor control, occupant discomfort and shortened plant life. It is estimated that this oversizing is typically responsible for approximately 10-15% of HVAC related energy consumption[6]. Greater detail on implications is given in Sections 4, 5, 6 and 7.

## 3.3 Oversizing Surveys

Oversizing surveys were carried out for the following systems:

- cooling/pumping plant
- air handling plant
- heating plant

### 3.3.1 Cooling plant surveys

Nine cooling systems were monitored during the summer period. Following the methodology set out in this document, current transformer clamps were connected to each phase of each contactor serving the cooling plant compressors. Thermistor bead surface probes were attached to the chilled water flow and return pipes serving the chiller unit and the data was logged at one-minute intervals. For accurate results immersion probes should be used where possible, since the use of surface probes can result in poor thermostat contact thus compromising accuracy. External and internal air temperatures were also monitored and logged every 15 minutes using temperature loggers.

The existence and extent of oversized cooling plant was then established by plotting the control state of each compressor during the warmest hour of the week during occupancy. The oversizing factor (OF) is the ratio of the maximum number of available chiller plant stages to the average number of stages operating over peak load. Where analysis has been conducted for a period with peak outside air temperature (OAT) below the external design temperature, then formulae have been applied to adjust the OFs appropriately. Figure 3 illustrates a typical cooling plant operating profile.

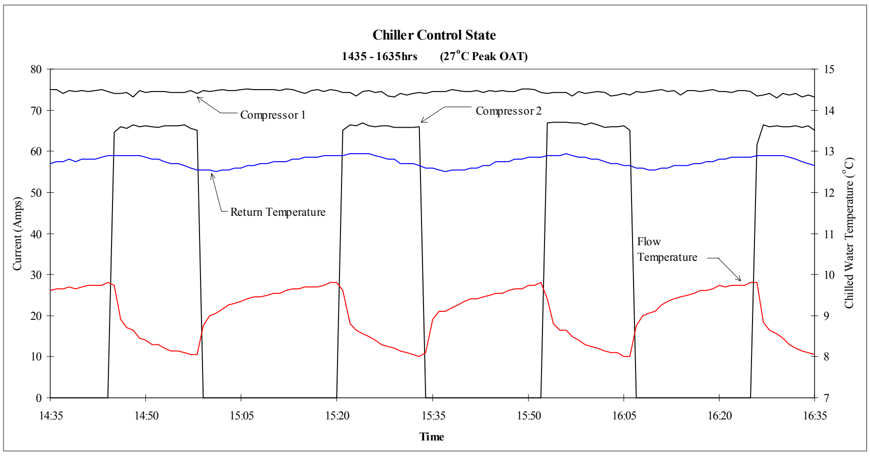



**Figure 3**
Typical chiller operation profile

The results from the surveys are illustrated in Figure 4. The findings established that all the chiller plants studied suffered from degrees of excess cooling capacity, as much as 4.5 times the required capacity in one instance. The two worst cases were found to be in the newer and the older of the systems thus implying that there has been no improvement in the sizing of these systems in the last 10 years.

62% of the installed cooling capacity in these surveys was superfluous. In 1998 it was estimated that the chillers sold in the UK for that year was equivalent to an average load of 635 MW[7]. If the percentage excess capacity found in the surveys was applied throughout the UK, then this implies that there is 394 MW of chiller capacity unnecessarily installed in the UK for 1998 alone.

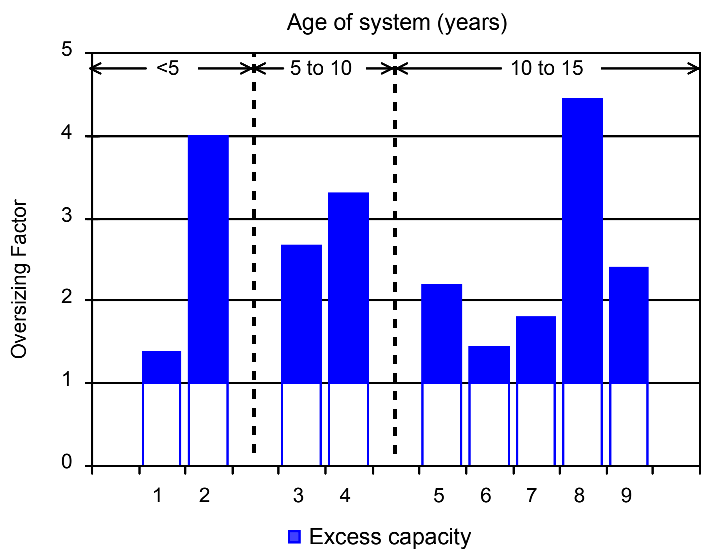



**Figure 4**
Cooling plant survey results

In many of the surveys a high OF was not an issue even though 100% oversizing was established in terms of oversizing factors. All of the chiller plant offered sufficient capacity control through cylinder unloading. Such capacity control allows cooling demands to be matched through staged compressor loading. This offered capacity control through multiple staging meaning that there is no drawback to having excess capacity other than initial capital and space costs. This does not alleviate the designers' responsibility of ensuring that over-engineered systems are not specified.

The main concern was not excess cooling capacity, or the available control of this capacity but the loading and unloading strategy within a chiller, and the sequencing between multiple chiller installations. What was found in many of the case studies was that although the systems had sufficient capacity control to ensure cooling demand was matched appropriately, many of the systems were not utilising the control in the most efficient manner.

In many cases it was found that chillers were operating a number of compressors at part-load when a lesser number of compressors at full load would prove a more efficient method of operation. Similarly, in multiple chiller installations the same scenario was occurring but this inefficiency was compounded by poor sequencing between chillers. For example, poor interaction between 2 chillers serving one zone often resulted in one chiller coming on-line to compensate for the other switching off. This often resulted in an on/off cycling profile between the 2 chillers, when utilisation of one chiller with a continuous profile would offer a more efficient operating pattern.

The most efficient combination of chillers and chiller stages should be determined from manufacturer's load efficiency characteristics. It may be that operating two chillers at three-quarters load can be more efficient than running one at full load and one at half-load.

### 3.3.2 Air handling plant surveys

Air flow rates were measured in 16 constant air volume (CAV) systems by undertaking a velocity traverse in the supply and extract ducts. Measurements were taken using a pitot tube and micromanometer. Any oversizing of the fans was then calculated in terms of excess flow generated which is generally the ratio of actual to design air changes per hour. For plant which solely serves to maintain minimum fresh air requirements such as plant serving fan coil units, then a ratio of actual to minimum design fresh air requirement (l/s/person) is used. The results are illustrated in Figure 5.

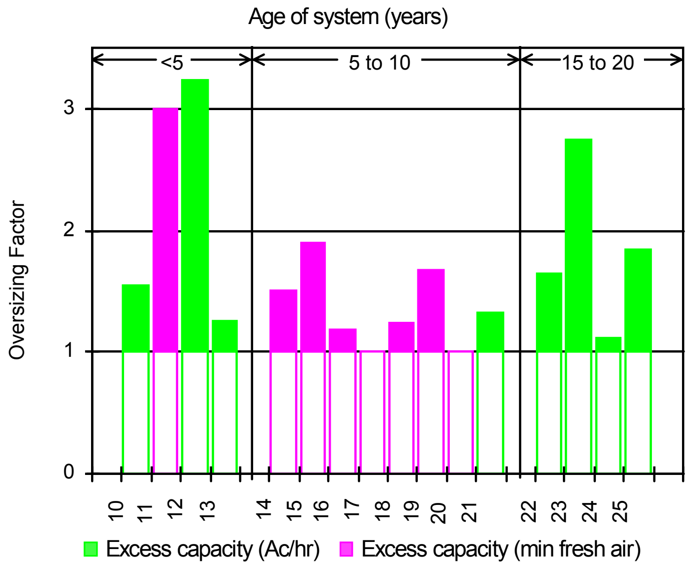



**Figure 5**
Air handling plant survey results

From the surveys it was found that 88% of the ventilation plants had more capacity than was required. There was some very significant oversizing found with up to as much as three times the necessary fan size in some instances. These most extreme cases of oversizing were found in the newer of the systems indicating that there has been no significant improvement in the sizing of air handling plant in the last 20 years. Figure 5 shows that the least oversized of the plants studied fell into the band of 5 to 10 year old systems although they were still oversized in many instances. It is difficult to identify whether there is any correlation between the degree of oversizing and this particular age band of systems or whether the lower results are being influenced by the fact that most of the systems in this band are serving fan coil systems. Oversizing analysis for such systems is based on minimum fresh air requirements rather than air changes per hour in the space. This would imply that fans for providing minimum fresh air requirements are more appropriately sized than other systems.

### 3.3.3 Heating plant surveys

Heating plant monitoring was conducted during the winter period. Surface temperature sensors and data loggers monitored the flow and return temperatures of the heating plant at one-minute intervals, with care being taken to ensure good thermal contact between the sensors and the pipe surface. External and internal air temperatures were also monitored every 15 minutes. The existence and extent of oversized heating plant was then established by plotting the action of each burner (control state) during the coldest hour of the week during occupancy. Where analysis has been conducted for a period above the external design temperature then formulae have been applied to adjust the oversizing factors (OFs) appropriately.

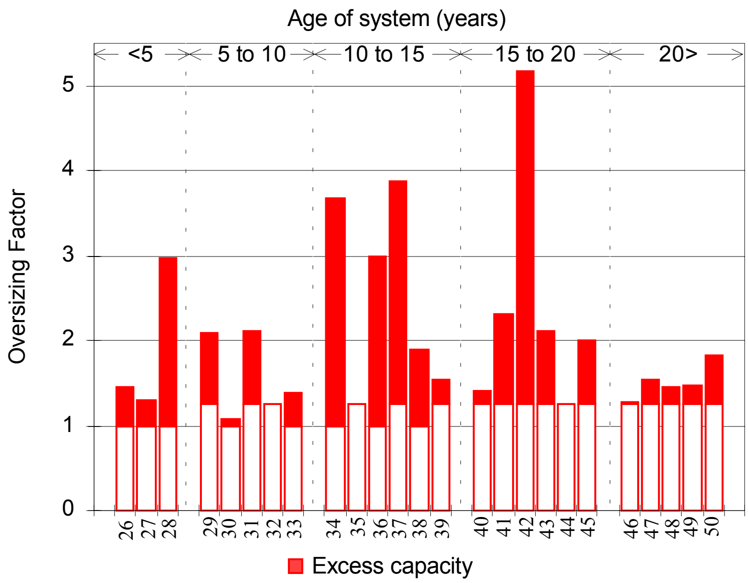



**Figure 6**
Heating plant survey results

The survey showed that 88% of the continuously operated systems and 76% of the intermittent systems had more capacity than was required. The required capacity line for intermittent systems used in Figure 6 is based on a 25% margin. Actual preheating requirements will be as individual as the buildings and will depend on several factors including expected internal heat gains, building heat loss and thermal time constants. The methodology for calculating preheating requirements is detailed in the guidance.

45% of the installed heating capacity in these surveys was superfluous. In 1998 it was estimated that the boilers sold in the UK commercial sector for that year had a total heat capacity of 3.9 GW[8]. If the percentage excess capacity found in the surveys was applied throughout the UK, this implies that there was 1.8 GW of boiler power unnecessarily installed in the UK for 1998 alone.

The results from the surveys showed that many of the systems had sufficient capacity control to make the performance issues related to excess capacity negligible. However, in many instances poor implementation of the controls available meant that the excess capacity became a problem.

### 3.3.4 Summary of findings

From the results of our surveys it was found that there was little indication that either HVAC system age, or the building type were of any great significance for the extent of oversizing.

The problem of oversizing at the design stage is greater in speculative buildings. In such cases the actual internal gains are unknown at the design stage and will be very much dependent on the tenants' working practices. Incorrect assessment of internal heat gains may be a possible explanation for the degree of oversizing determined in the chiller plant surveys. Small power loads, particularly for office equipment, are generally far lower than design calculations predict[9,10,11].

Better design and building practices over recent years combined with an ever increasing focus on energy efficiency would imply that newer buildings should show a decrease in oversizing. However, the monitored results show no apparent improvements in the extent of oversizing. Trends in designing for flexibility, increased litigation, increasing internal gains with better insulated buildings, or the increasing use of software packages which tend to incorporate margins usually unknown to the user, may all contribute to the OFs remaining high in the newer systems. The findings have indicated that there is no trend when it comes to oversized plant, which suggests that over-engineering and poor system evaluation are still present in modern building services design practices.

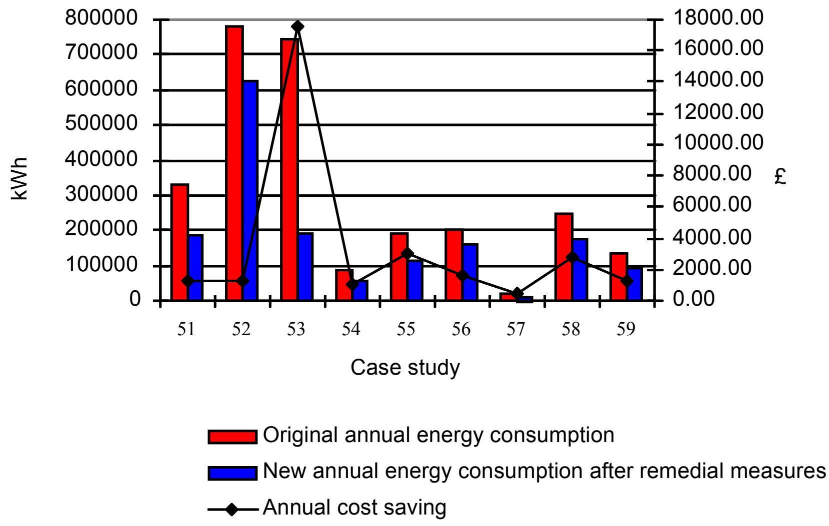
The implications of the excess 2.2 GW of installed heating and cooling capacity in 1998, in terms of capital costs, space requirements, running and maintenance costs are very significant. If all existing installations were considered in addition to this then the total excess capacity installed in the UK would be staggering. However, despite the initial installation costs and space implications of oversized plant, what has emerged from the findings is that oversized plant is not of great concern if modular and/or good capacity control techniques are adopted. A heating installation may have 4 times the required heating capacity, but if this capacity is made up from 4 well controlled and sequenced boilers then only the one required boiler will be running anyway. The same applies to multi-staged or speed control chillers, and fans or pumps with variable speed drives. However, the benefits of good capacity control measures will not be achieved if the control strategies are poor or inappropriate. In many of the surveys conducted for this project a lack of an appropriate control strategy resulted in systems running inefficiently with unnecessary excess capacity. It is important to incorporate optimum capacity control techniques at the design stage, particularly in speculative buildings. In existing buildings remedial measures should be applied wherever possible to improve oversized plant efficiency through good capacity control.

### 3.3.5 Remedial measures

Generally, fans and pumps are oversized by at least 15%, with more generous over capacity often given to boilers and chillers. It is estimated that this oversizing is typically responsible for approximately 10-15% of HVAC related energy consumption[6].

Nine HVAC systems in the UK were studied where action had been taken to ameliorate plant oversizing and reduce energy consumption. The energy savings amounted to almost £30,000/annum with an average payback in under 2½ years. The results of these surveys are illustrated in Figure 7.

**Figure 7**
Energy savings




These nine case studies had adopted simple measures to improve the performance of existing oversized and poorly performing plant. However these measures alone produced total annual energy savings of 40%, equating to 1115 MWh. Some of these savings cannot be credited to a reduction of oversized plant alone, but are also based on reduced capacity and improved operation. A summary of the case studies is provided in Appendix A and selected case studies are shown below.

## Case Study 55: Variable speed fan

### Existing system

An existing inefficient backward curve fan provided 8.46 $m^3/s$ of air to an office space at 63.5% full load. The fan absorbed 17.8 kW of power and was set to operate continuously 24 hours per day for 365 days per annum.

### New system

It was determined that the required duty could be achieved by a much more efficient fan with a variable speed motor. The new fan could meet this duty whilst only absorbing 10.3 kW of power.

### Savings

The total power input of the new higher efficiency fans was 12.76 kW compared with an original 22.02 kW. Annual energy and running cost savings are detailed in Table 1.

**Table 1**
Fan energy and running cost

| | Original | New | Saving |
|---|---|---|---|
| Annual energy consumption (kWh) | 192895 | 111778 | 81117 |
| Annual running costs (£) | 7089 | 4108 | 2981 |

By replacing this one fan a 63% energy saving was achieved which equated to an annual energy cost saving of £2,981. With an overall capital cost for replacing the fans of £3,540, the new system will pay back in 14 months. This is illustrated in Figure 8.

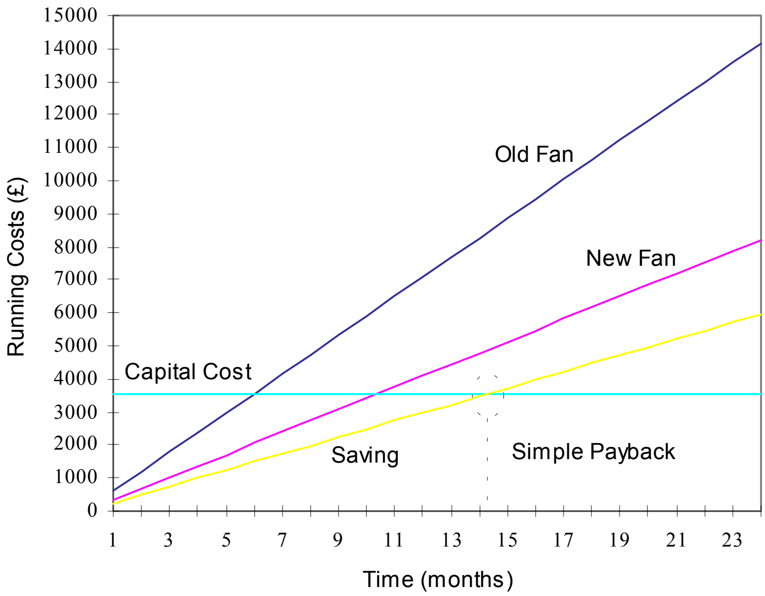



**Figure 8**
Payback on variable speed fan

## Case Study 51: Heating system and control improvements

### Original system

The original system had one boiler providing space heating to 1333 $m^2$ of office space. This single boiler supplied perimeter radiators for the entire building. The system had optimiser control but was only dependent on internal zone temperature and not outside air temperature. The building comprises 5 floors, which were treated as individual zones with a thermostat serving each one. The thermostats operated solenoid valves that shut various runs of the heating circuits, although, only two of these remained connected to the system.

A survey established that not only was the optimiser not operating correctly, but it was also allowing the system to run 24 hours per day, even over weekends. Significant boiler cycling was evident from this survey ,confirming that the plant was providing too much heating, for most of the time. This motivated the building operators to take action in an attempt to reduce the energy waste . The energy manager responsible followed on from the BSRIA findings by conducting a thermal comfort survey in the form of a questionnaire.

### New system

Changes to the boilers, pumps and control system were carried out.

The old boiler was replaced by one gas fired 140 kW condensing boiler and one gas fired 105 kW conventional boiler.

The original control system was replaced with a network-accessible optimiser/compensator. The boiler sequencing was arranged to always lead with the condensing boiler and new zone pumps had variable speed control which varies the volume flow rate in each zone in relation to the load. Each floor was also split into six zones, each controlled via an individual thermostat, based on the results from the thermal comfort survey.

### Savings

During a change in the working patterns of the occupants, there was a period in 1998 when main areas of the building became unoccupied. The building returned to full occupation in May 1998. For this reason a comparison between space heating energy consumption has been made between June 1996 to May 1997 and June 1998 and May 1999.

It can be seen from Table 2 and Figure 9 that since the changes were made in 1997 there has been a dramatic reduction of 43% in energy consumption for space heating. This is equivalent to a saving of £1300 per annum. The old system could not be constantly monitored by the energy manager, who is not based at this particular building. The new networked control system remotely allows the system to be monitored and controlled as required. Although there has been increased capacity installed, improved capacity control techniques prevents the problem of oversized plant from becoming an issue. The two boilers have, individually, less capacity than the original single boiler, thus permitting the lead boiler to operate for longer periods at a more efficient full load. The second boiler has capacity available for more extreme loads when required and can act as a stand-by when not. Improved zoning and thermostatic control of the zones, in conjunction with the use of variable speed pumps, allows a more precise delivery of heating to the spaces as required. Proper operation of the main optimiser control determines a more efficient operation period for the heating system compared to the continuous operation of the previous system.

**Table 2**
Space heating gas consumption

| Month | 1998 - 1999 | 1996 - 1997 | Variation | |
|---|---|---|---|---|
| | kWh | kWh | kWh | % |
| Jun | 2146 | 80 | 2066 | 2582.50 |
| Jul | 743 | 11 | 732 | 6654.55 |
| Aug | 1089 | 0 | 1089 | 0 |
| Sep | 3198 | 0 | 3198 | 0 |
| Oct | 16511 | 27496 | -10985 | -39.95 |
| Nov | 29732 | 57798 | -28066 | -48.56 |
| Dec | 36654 | 45912 | -9258 | -20.16 |
| Jan | 33292 | 74615 | -41323 | -55.38 |
| Feb | 27886 | 47474 | -19588 | -41.26 |
| Mar | 20337 | 28860 | -8523 | -29.53 |
| Apr | 11499 | 38484 | -26985 | -70.12 |
| May | 2438 | 7657 | -5219 | -68.16 |
| Total | 185525 | 328387 | -142862 | -43.50 |

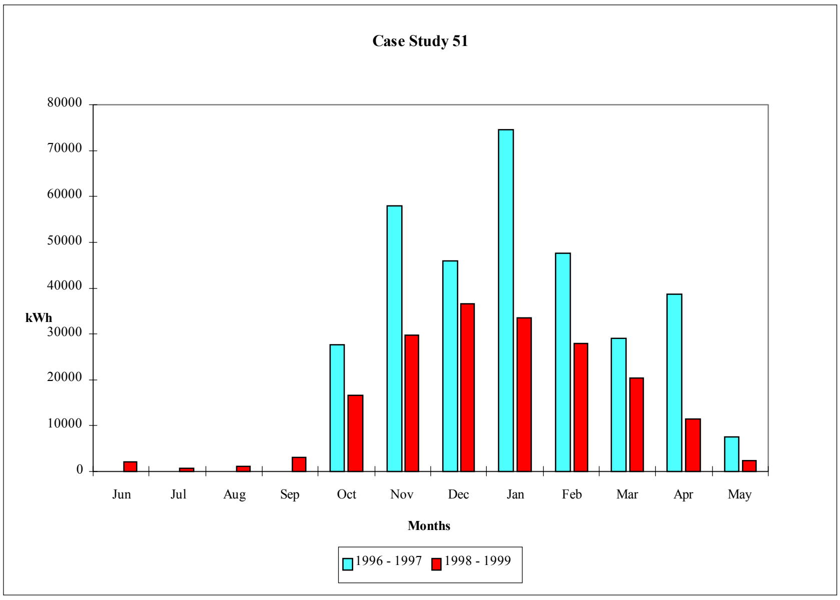



**Figure 9**
Space heating gas consumption before and after remedial measures

The savings achieved since the improvements to the heating system can be clearly seen in Figure 10. The trend line after modifications were implemented falls below the before-modifications trend line. This shows that for periods of similar degree day values the system is now using less fuel. The scatter of the after-modification points is now much closer to the trend line, which implies that closer control is now being achieved. This is confirmed in Figure 11 which emphasises the now close relationship between the outside temperature and energy consumption.

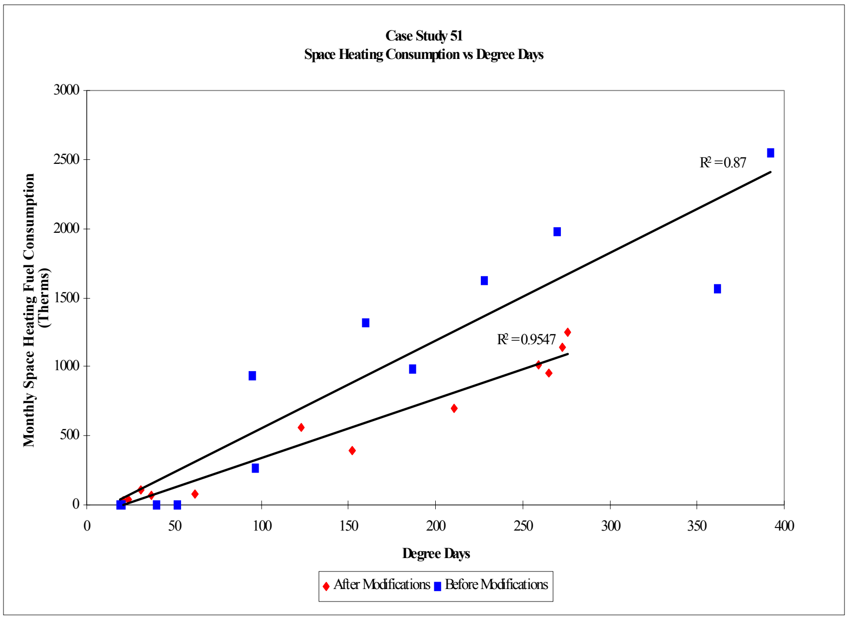



**Figure 10**
space heating fuel consumption vs degree days

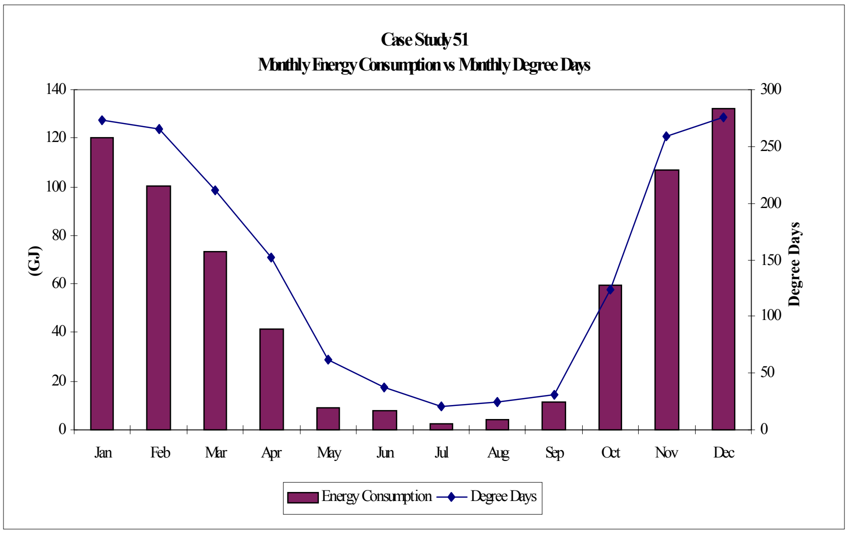



**Figure 11**
Monthly energy consumption vs degree days

Despite the new heating plant having greater capacity than that already installed, the system now operates more efficiently and has produced an energy saving of 43%. This shows how the problems of oversized plant can be dealt with through the implementation of good capacity control. Improved capacity control in conjunction with higher efficiency boilers and an improved control strategy produced significant energy savings in this case.

## Case Study 60: Improved motor control for chiller system

### Original system

An office building had three chillers, one absorption type and two compression type. Each compression type chiller has two compressors, each with two stages.

### New system

Motor controllers were fitted to both compressor motors on one of the chillers.

### Savings

The zone, air supply and chilled water set-points were reduced to the minimum permitted values and an offset was added to the outside air temperature sensor to ensure that the system operated on minimum fresh air (to minimise free cooling). These conditions ensured that the chiller operated, but throughout the test period it was only on one stage of one compressor.

Figure 12 shows the total chiller current consumption for the entire monitoring period including the compressors and condenser fans (condenser fans are a very small proportion of the total load). It can be seen that there was a small reduction in current with the motor controller enabled.

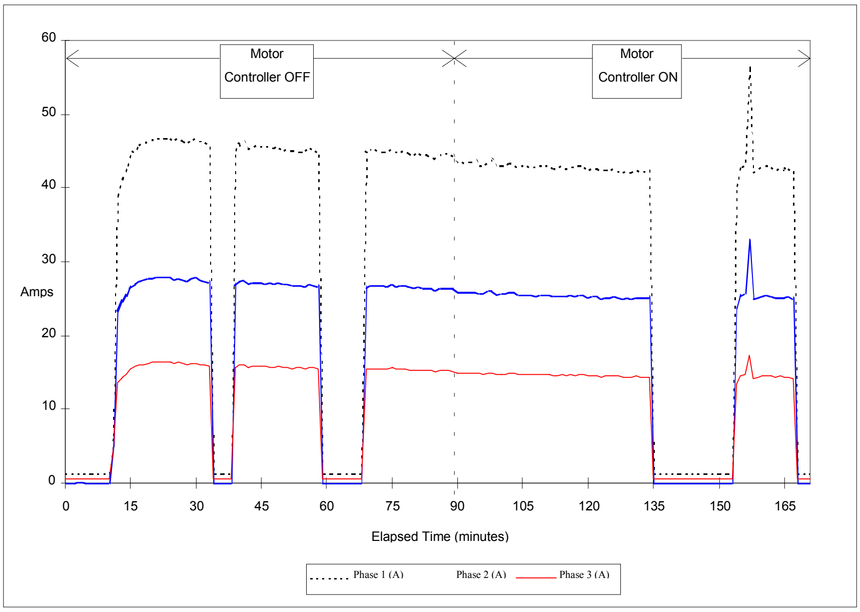



**Figure 12**
Chiller current with and without motor controller operating

Figure 13 shows the power consumption with and without the motor controller on. To allow easier comparison the two plots are shown over the same period.

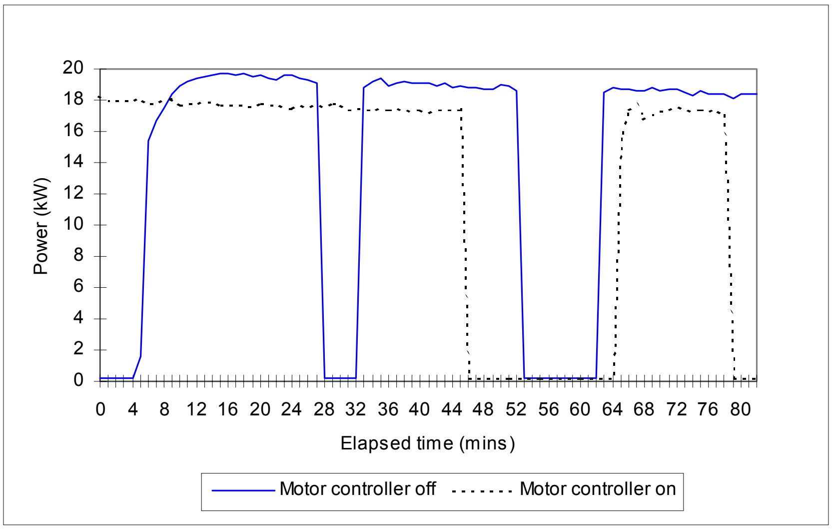



**Figure 13**
Chiller power consumption with and without the motor controller

Table 3 shows the average power consumption with and without the motor controller operating and also the saving. Throughout the test the chiller operated on only one stage of one compressor.

**Table 3**
Power consumption

| Average power-controller off (kW) | Average power-controller on (kW) | Average saving (kW) |
|---|---|---|
| 18.85 | 17.45 | 1.40 |

In order to gauge how the reduction in power from the limited monitoring could translate to an annual saving and a simple payback on the capital cost, a potential operational scenario was developed. This attempted to estimate the annual operation at each stage (two compressors each with two stages) based on outside air temperature. It should be stressed that this scenario contains a number of assumptions and the results should not be used as an indicator of actual savings but merely an example of potential savings. Also, in this building the absorption chiller is the lead chiller and therefore the hours of operation of the chiller tested may be less than in the example. The assumptions made were as follows:

- cooling season : May to September (20 weeks)
- operation:

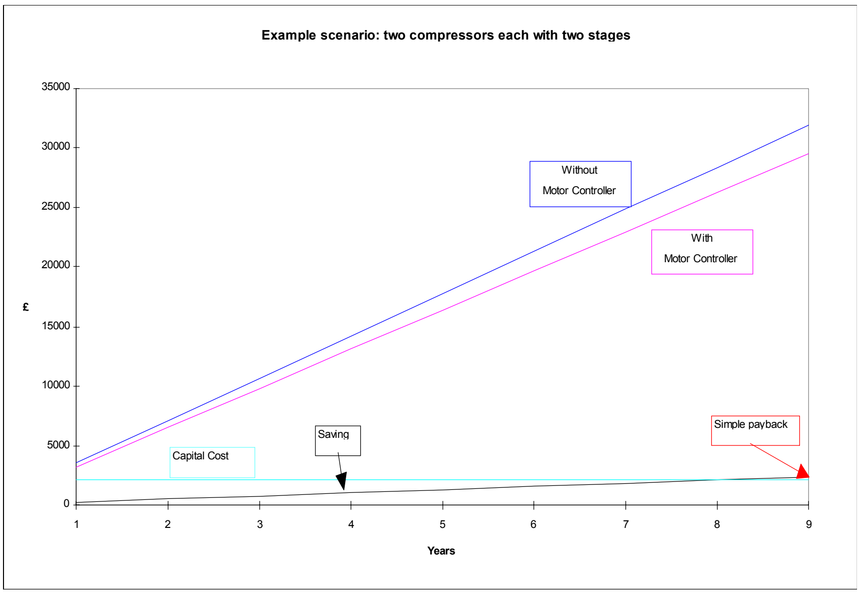
| Ambient temperature | Stages operating |
|---|---|
| 14°C to 18°C | 1 |
| 19°C to 22°C | 1, 2 |
| 23°C to 25°C | 1, 2, 3 |
| over 25°C | 1, 2, 3, 4 |

- the power consumption of the second stage of each compressor was set to 150% of the first stage (this was typical of chillers monitored during the oversized plant project)
- no free cooling
- electricity tariff 5p/ kWh.

The periods of operation at each stage were calculated from ambient temperatures monitored at the BSRIA site in Bracknell, Berkshire during 1999.

Figure 14 shows the savings from the scenario described above. The result was an estimated simple payback of 9 years. Actual savings may be better or worse than this.

**Figure 14**
Estimated simple payback from example scenario



# 4 AIR HANDLING SYSTEMS

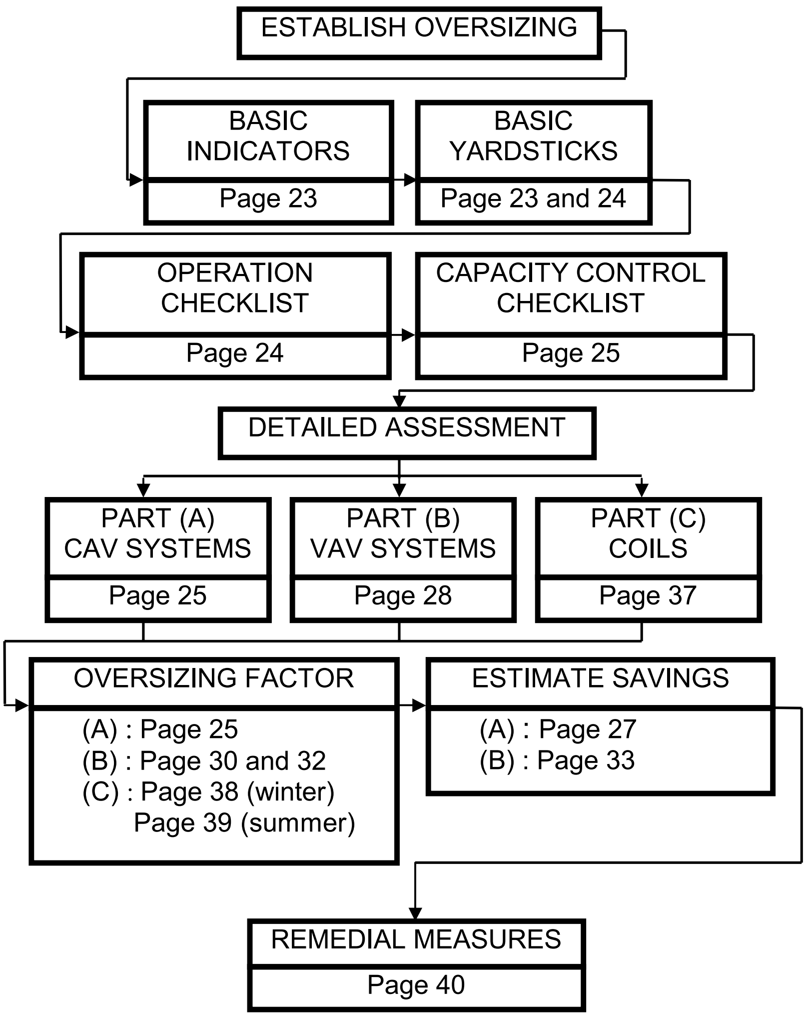
This section provides a methodology for reducing the energy consumption of oversized constant air volume (CAV) or variable air volume (VAV) air handling plant. The guidance in this section should be implemented before that for heating and cooling systems as thermal load depends on air handling system performance. The work is based on the method developed in *Oversized air handling plant*[12].

The section is set out as follows:

⇒ 4.1 Implications of oversized air handling systems
⇒ 4.2 Establish oversizing - target air handling plant oversizing using:
- 4.2.1 Basic indicators
- 4.2.2 Basic yardsticks
- 4.2.3 Assess and optimise operation and maintenance practice

⇒ 4.3 Quantify air handling plant overcapacity using plant monitoring.
- Part A : CAV Systems
- Part B : VAV Systems
- Part C : Cooling and heating coils

⇒ 4.4 Remedial measures - implement remedial measures in order to improve system performance.

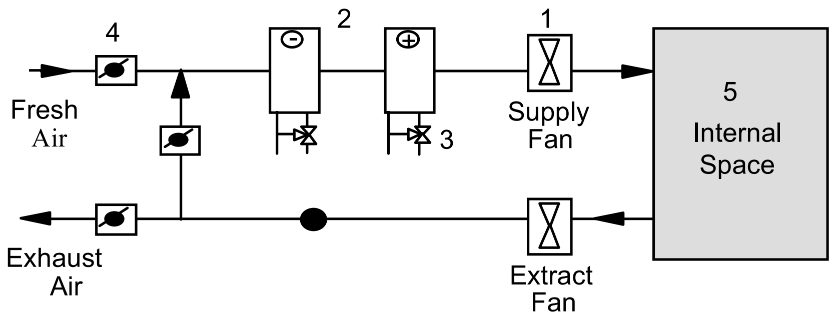


## 4.1 Implications of Oversized Air Handling Systems

Oversized air handling systems and components can incur increased space requirement, capital costs and energy consumption. Furthermore, difficult plant control can occur leading to compromised occupant comfort and shortened plant life. More detailed implications for constant air volume (CAV) and variable air volume (VAV) systems are illustrated in Figure 15 and Figure 16 respectively.

**Figure 15**
CAV system




Notes:

1. Oversized fans provide excessive flow rates and operate inefficiently, increasing power requirements.
   They also put more heat into the air and so increase the cooling load on the system (but heating load is reduced).
2. Oversized coils increase the fan pressure requirements and are likely to have oversized pumps.
3. Oversized valves reduce effective control. High valve cycling rates reduce effective valve life.
4. Oversized dampers reduce effective control. Excess fresh air can be supplied, increasing the loads on the heating and cooling coils and their respective systems, thereby increasing energy costs. Reduced fresh air could compromise indoor air quality.
5. Oversized terminal units increase fan power requirements and can compromise comfort.

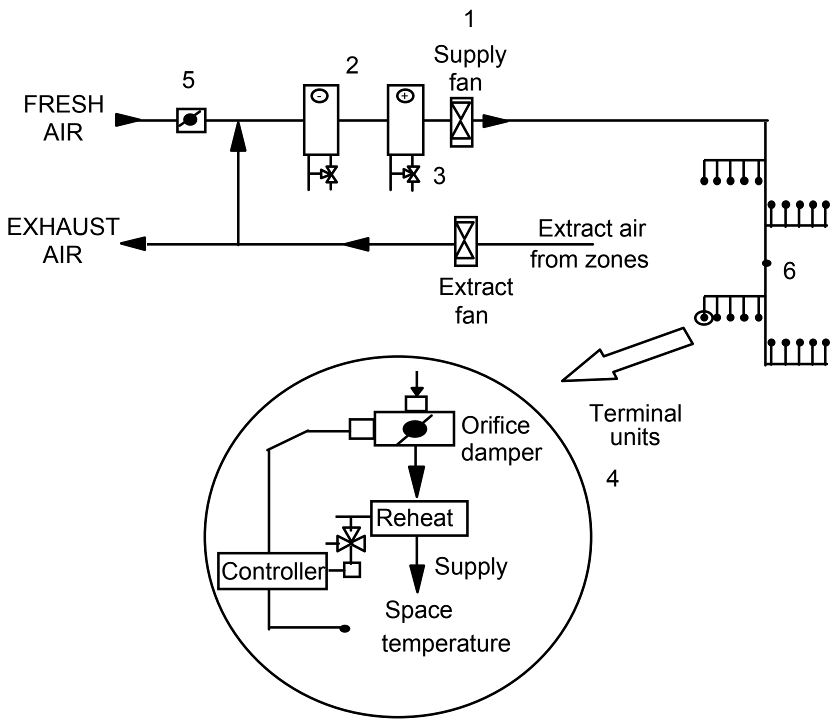



**Figure 16**
VAV system

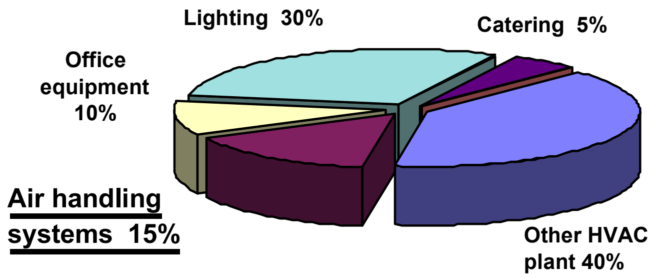
Notes:

1. Oversized fans do not operate at their most efficient operating point and consume additional energy.
   At minimum fresh air settings, oversized fans put more heat into the air increasing the cooling load on the system (heating load is reduced).
2. Oversized coils increase the fan pressure requirements and the amount of water pumped in the hydronic systems.
3. Oversized valves reduce effective control. Simultaneous heating and cooling increases energy consumption. High valve cycling rates reduce effective valve life.
4. Oversized terminal units reduce control. Orifice dampers never fully open, and the turndown ratio of the box is limited to ensure the Coanda effect is not destabilised. Energy consumption increases. Excessive minimum VAV terminal unit settings increase the fresh air provision to the internal space thereby increasing the loads on the hydronic systems, especially for reheat control, and the power requirements of the fan.
5. Oversized dampers reduce effective control. Excess fresh air can be supplied, increasing the loads on the heating and cooling coils and their respective systems, thereby increasing energy costs. Reduced fresh air could compromise indoor air quality.
6. If static pressure is set too high, fan energy consumption increases due to the increased pressure consumed by the system.

### 4.1.1 Cost implications of air handling systems

Generally fans are oversized by at least 15%. Due to the relatively high price currently paid for electricity in comparison to other energy sources, it is worthwhile targeting oversized air handling plant.

**Figure 17**
Typical energy cost example - air handling system




(Based on EEO figures[13])

Where total building annual energy cost = £120,000

Plant oversizing at typically 10-15% of HVAC energy consumption approximates to:

= £8,250/year

**Examples of why air handling plant oversizing costs money**

1. **REDUCTION IN SYSTEM EFFICIENCY:** System efficiency is different to an individual plant's dynamic efficiency. The excess output from oversized plant may be consumed by the system without doing any useful work. In this way, the individual plant's efficiency may still be high but system efficiency has been compromised.

   For example: a fan oversized by 25% in terms of generating excess flow reduces fan system efficiency as it can consume twice the energy that is really necessary (derived from fan laws).

2. **APPROPRIATE CAPACITY CONTROL:** Capacity control techniques are used to control plant output for part-load operation. Plant output must match the demand profile. The capacity control method installed may be suitable for the original design load profile but not for actual operating conditions (especially if the control method is inefficient at low loads). An alternative control method may be more appropriate.

   For example: it is estimated that a VAV fan oversized by 35%, using guide vanes, may consume more than twice the annual energy than if a speed-controlled electrical drive system were installed (derived using data supplied by the EEO[13]).

3. **OVERALL SYSTEM CONTROL:** System control is used to minimise energy consumption and maintain satisfactory comfort conditions. Oversized plant, operating for most of the time at part-load, induces plant cycling and so compromises control stability.

For example: oversizing heating or cooling coils and their associated valves can reduce valve authorities to below minimum limits prescribed for effective control. Systems often become unstable.

4. **PLANT LONGEVITY:** Oversized plant permanently operating at low loads can reduce plant life. Accelerated wear can also arise from unstable control caused by plant oversizing.

   For example: many oversized control valves fail prematurely because they operate close to their closed position for most of the time.

## 4.2 ESTABLISHING OVERSIZING OF AIR HANDLING SYSTEMS

### Basic assessment

Oversized plant is defined as plant that has more capacity than is required.

Air handling plant can therefore be categorised as oversized if:

a) The fan generates an air flow rate in excess of building requirements.
b) The fan does not operate at its most efficient operating condition because it was designed to generate a larger air volume and/or pressure.
c) The full capacity of cooling or heating coils is never fully utilised.
d) Air terminal units are able to supply a greater air volume than required for maximum cooling demands.
e) The minimum setting on air terminal units supplies an air volume in excess of fresh air or air change rate requirements or that necessary to ensure adequate heating.

There are, of course, certain cases where more plant capacity is desirable for satisfactory plant and building operation. Examples include margins on motor sizing to cope with high start-up torques and possible supply voltage drops during peak load operation.

By focusing on building performance during periods of high load, air handling plant oversizing can be very apparent. The observations in Table 4 are examples of basic indicators that point to air handling plant overcapacity for both constant and variable air volume systems.

### 4.2.1 Basic indicators of air handling systems

**Table 4**
Basic indicators of air handling systems

(Tick as appropriate)

| CRITERIA | TICK |
|---|---|
| A larger flow rate supplied than expected | |
| Summer internal space temperatures easily maintained | |
| Small cooling load such as maximum building electrical demand is much smaller than the design value | |
| Complaints of draughts in occupied spaces | |
| Noisy air diffusers | |
| Duct static pressure easily maintained during periods of high cooling demand | |
| Fans never operate close to full capacity (especially if continuously throttled by dampers) | |
| Low fan motor power factor | |
| Full heating /cooling coil capacity is never utilised (for example, only up to 70%) | |
| VAV terminal unit orifice dampers never/rarely fully open during occupancy | |
| Carbon dioxide levels below 600 ppm in the space when fresh air and terminal unit orifice dampers are at their minimum positions and the building is fully occupied (note: spot $CO_2$ readings are not representative). | |
| *One or more ticks : proceed to Box A* | |

### 4.2.2 Basic yardsticks of air handling systems

The yardsticks are used to establish whether a more detailed oversizing assessment is likely to be worthwhile. If air handling plant is serving a purely fresh air system, eg serving fan coil units use Part 2 of Boxes A and B, otherwise use Part 1.

**Box A (part 1)** Installed capacity

| A1 | A2 | A3 |
|---|---|---|
| Ventilated space ($m^3$) | Fan duty (l/s) | Installed fan capacity (l/s/$m^3$) |
| | | A1 / A2 |
| | | |

*Proceed to Box B.*

**Box A (part 2)** Minimum fresh air capacity

| A4 | A5 | A6 |
|---|---|---|
| Number of occupants* | Fan duty (l/s) | Installed fan capacity (l/s/person) |
| | | A4 / A5 |
| | | |

**If not known 14 $m^2$ net area of ventilated floor space per person is recommended [14].*

**Box B (part 1)** Basic yardsticks

| B1 | B2 | B3 |
|---|---|---|
| Building type | Capacity yardsticks* (l/s/m³) [4] | Expected oversizing factor (OF) |
| | | A3 / B2 |
| Offices | 1.4 | |
| Retail | 2.1 | |
| Halls and theatres | 2.1 | |
| Restaurants | 3.5 | |

**Extract should be generally 10% to 20% less than supply (where ingress is unlikely to be a problem).*
*If B3 > 1 it is likely the system has excess capacity. Proceed to Table 5.*

**Box B (part 2)** Minimum fresh air capacity

| B4 | B5 | B6 |
|---|---|---|
| Level of smoking | Capacity requirements* (l/s/person)[5] | Expected oversizing factor (OF) |
| | | (A4 x B5) / A5 |
| No smoking | 8 | |
| Some smoking | 16 | |
| Heavy smoking | 24 | |
| Very heavy smoking | 36 | |

**Extract should be generally 10% to 20% less than supply (where ingress is unlikely to be a problem).*
*If B6 > 1 it is likely the system has excess capacity. Proceed to Table 5.*

### 4.2.3 Operational review of air handling systems

Having established through the basic assessment that the plant is likely to have excess capacity, then plant monitoring is likely to be worthwhile. However, before undertaking plant monitoring it is important to review the operational measures of the system. This allows the status of the air handling system to be ascertained and helps put plant monitoring into context. Use the checklist in Table 5.

**Table 5**
Air handling system - check list

| Category | Check |
|---|---|
| General air supply | • Unoccupied zones are isolated from air handling system.<br>• Control sensors are properly located and are functioning correctly (accuracy checked in the last 12 months).<br>• Air infiltration through windows, doors and surrounds is minimised.<br>• Duct and damper leakage is minimised.<br>• Air system is balanced. |
| Thermal load | • Fresh air dampers and coil control valves operate effectively and filters are clean.<br>• Economy cycle/enthalpy control is utilised where possible.<br>• Blinds, solar shades and solar film are used on windows.<br>• Energy efficient PCs, lights, and other electrical equipment are utilised. |
| Cooling demand controlled air supply | • Supply temperature schedule is reasonable.<br>• Terminal unit orifice dampers operate appropriately.<br>• Terminal unit temperature set-points fixed as specified.<br>• Use of night cooling where possible, to remove internal heat gains absorbed by the building fabric, furniture, and fittings. |

Consideration should be given to implementing operational measures from Table 5 where appropriate, otherwise an inaccurate assessment of plant oversizing will result. This will also optimise the savings that can be achieved.

If the plant utilises good capacity control measures as indicated in Table 6, then there may be little benefit in proceeding with plant monitoring unless the plant is particularly inefficient and being considered for replacement. Complete Table 3 as appropriate. Oversized plant should never be replaced on a like-for-like basis. Where necessary, plant should be tagged as being oversized for when plant replacement arises.

**Table 6**
Fan capacity control - check list

| Item | Tick |
|---|---|
| • Multi-air volume system controlled to occupancy or cooling demands. | |
| • Capacity controlled by variable speed drive systems (investigation will only be justified if the fan is at least 35% oversized). | |

## 4.3 Plant monitoring of air handling systems

A more accurate assessment should be carried out by monitoring air handling plant once oversizing has been identified using the basic indicators. Follow the appropriate sub-steps, dependent on the type of air handling system installed. Reference to the case studies may aid understanding.

1. Install plant monitoring as outlined in Figure 18 and Figure 19. Monitoring may be simplified by using a BMS (if present).
2. Use portable logging equipment for other required parameters Check that accuracy of sensors is within 5%. Follow all safety guidelines. Refer to Section 8 for hints and tips for conducting monitoring of air handling systems.
3. Document and archive results and conclusions for future reference.

The process of monitoring and assessing air handling systems differs depending on whether or not the system is a CAV or VAV system. For CAV fans follow Part (A) of this section for VAV fans follow (Part B).

**Part (A) CAV systems**

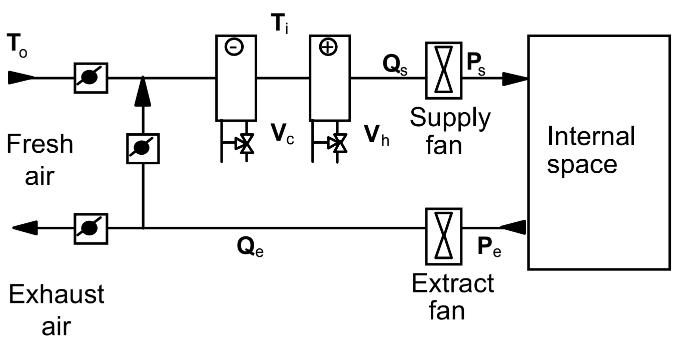



**Figure 18**
Monitoring positions for CAV systems

**Spot readings**

$Q_s$ - Flow rate through supply fan ($m^3/s$)
$Q_e$ - Flow rate through extract fan ($m^3/s$)

Measure flow rate by undertaking a velocity traverse in the duct, at a position that conforms to recommendations (see *Commissioning of air systems in buildings*[15])

Monitor fan power consumption, if possible:
$P_s$ -Supply fan power consumption (kW)
$P_e$ -Extract fan power consumption (kW).

### Assess excess air flow rate

Take a series of measurements of air flow for both supply and extract fans as specified as above. System characteristics can vary depending on the position of the fresh air damper. Therefore take readings for different amounts of fresh air if the damper modulates.

The fan oversizing factor is then calculated in terms of excess flow generated.

Excess air flow rate is the ratio of actual to specified air change rates per hour. Calculate supply fan oversizing factor in Box C. Extract air flow rate will generally be proportional to the supply air flow rate. This is site-specific and should be noted at the bottom of Box C.

**Box C** Oversizing factor

| C1 | | C2 | C3 |
|---|---|---|---|
| Recommended total air supply rate* (air changes per hour) | | Minimum monitored air flow rate (l/s) | Oversizing factor (OF) |
| | | | (3.6 x C2) / (A1 x C1) |
| Offices | *4 - 6* | | |
| Retail stores | *3 - 8* | | |
| Libraries, museums | *3 - 4* | | |
| Conference rooms | *6 - 10* | | |
| Canteens | *8 - 12* | | |
| Restaurants | *10 - 15* | | |
| Extract air flow rate | | | |

**From CIBSE Guide B*[5]
*Apply to both supply and extract fans then proceed to Box D.*

**Box D** Airflow rate reduction

| D1 |
|---|
| Flow rate reduction (%) |
| [(1 - (1/C3)) x 100] |
| |

| D2 |
|---|
| Actual required flow rate (l/s) |
| (D1/100) x C2 |
| |

### Assess fan energy consumption

Take a series of current readings to assess the power consumption of each fan (or estimate by consulting with the manufacturer). Convert to an annual energy consumption by multiplying the power by the average number of hours that the fan operates each day and number of days during the year.

For example, supply and extract fans consume 10 kW of power each for, on average, 12 hours a day, 365 days a year. The annual energy consumption of both fans is therefore:

$= 2 \times 10 \times 12 \times 365 = 87{,}600$ kWh/annum
At 2000 prices the two fans cost about £6,000 a year to run.

**Estimate available savings**

Fan efficiency depends on the capacity control method employed. To determine the actual savings that will be achieved the change in fan operating efficiency must be incorporated. Fan efficiency can easily be established by completing the following equation:

$$\text{Fan Efficiency} = \frac{\text{Fan total pressure ( kPa) x Flow rate (l/s)}}{\text{10 X Power consumption (kW)}}$$

The current operating efficiency can be found by substituting the current operating values as monitored into the above equation. However, establishing the new efficiency is somewhat more complex. The operating conditions of the fan at the required flow rate will be dependent on the method of flow rate regulation.

- Regulation using a dampers alters the system curve and increases the operating pressure.
- Regulation using a variable speed drive reduces the flow rate on the same system curve and thus decreases operating pressure.

The only accurate method of establishing the new operating efficiency and hence actual achievable saving is by using fan curves which can be obtained from operation and maintenance manuals or from the manufacturer. If these are not available then use Box E to provide 'typical' savings based on the excess air flow generated as established in Box C. A summary of typical savings for various degrees of oversizing is provided in Table 7.

Based on the excess air flow generated as established in Box C convert the savings to £/annum saved (based on the assessed energy consumption figures calculated above) and then consider the remedial measures that can be implemented (Section 4.4).

**Box E** Available savings

| E1 | E2 | E3 |
|---|---|---|
| Power consumption* (kW) | New power consumption** (kW) | Saving*** (%) |
| | $E1 / (C3)^3$ | [(E1 - E2) / E1] X 100 |
| | | |

**Power consumption by monitoring or from manufacturers.*
***It is assumed that a maximum air flow rate reduction of 20% is possible for oversized fans without affecting system balancing. Larger reductions in air flow may be possible. If a 20% limited reduction is assumed, the maximum value that should be used for C3 should be 1.25.*
****This assumes that fan efficiency is constant. This will not be the case but allows typical savings to be established. Axial flow fans have a typical efficiency range of 60% - 75%, whereas, for centrifugal fans, efficiencies can be within a range of 45% to 85% depending on the type of impeller[5].*

**Table 7**
Typical savings available through CAV air handling units

| Oversizing factor (OF) | | | | |
|---|---|---|---|---|
| 1.05 | 1.1 | 1.15 | 1.2 | 1.25 |
| 14% | 25% | 34% | 42% | 49% |

Notes:
The figures are derived using the savings equation shown in Box E, assuming that fan efficiency is constant.
These savings do not include the additional savings associated with a reduced thermal load on the boiler and chiller plant that may result from a reduction in flow rate.

## Case study 61: CAV system

Table 8 calculates the oversizing factors for the supply fans for four air handling units serving an office building. It is estimated that about £3,100 can be saved each year if the oversized fan capacity is eliminated (assuming similar savings apply to the extract fans as well). A large proportion of these savings is likely to be achieved by simply adjusting the pulley size on the fans so that the required flow is achieved.

**Table 8**
CAV oversizing analysis

| Supply Fan | AHU 1 | AHU 2 | AHU 3 | AHU 4 |
|---|---|---|---|---|
| Measured power (kWe) | 9.81 | 3.20 | 2.21 | 1.03 |
| Annual energy consumption (kWh) | 42986 | 14024 | 9683 | 4503 |
| Ventilated building volume ($m^3$) | 6940 | 3640 | 2180 | 1080 |
| Duct area ($m^2$) | 1.44 | 0.64 | 0.56 | 0.36 |
| Measured air velocity (m/s) | 7.9 | 7.5 | 6.5 | 5.3 |
| Measured air volume ($m^3$/s) | 11.4 | 4.8 | 3.7 | 1.9 |
| Measured air changes per hour | 5.9 | 4.8 | 6.1 | 6.4 |
| Recommended air changes per hour | 5 | 5 | 5 | 5 |
| Excess flow oversizing factor | 1.18 | 0.95 | 1.21 | 1.27 |
| Estimated savings* (%) | 40 | - | 43 | 49 |
| Estimated £/annum saved | £1204 | - | £291 | £154 |

***It is assumed that a maximum air flow rate reduction of 20% is possible for oversized fans without affecting system balancing. Larger reductions in air flow may be possible.*

### Part (B) VAV Systems

Both fans and terminal units are targeted in a VAV system oversizing assessment.

The maximum flow through VAV fan installations normally occurs to satisfy the largest cooling load to a specified supply air temperature. If these fans are able to supply a greater flow than this maximum the fan is said to be oversized.

VAV terminal units can be oversized if they are able to supply a greater flow rate than required by the maximum cooling demand. In addition to this, the minimum setting on VAV terminal units should also supply an adequate air flow to sustain the minimum fresh air requirements of the internal space. Under such conditions, if the flow rate is excessive for fresh air requirements, then the terminal units that are supplying the space are oversized. Therefore, VAV terminal units should be assessed in terms of both maximum and minimum flow rates. This requires summer and winter monitoring. Monitoring of $CO_2$ levels within the space will indicate if minimum fresh air requirements are being achieved.

### Summer monitoring

During summer (when the building is fully occupied and functional): Install plant monitoring as outlined on Figure 19, and log for at least 7 days (go to Boxes F and G).

Ideally this should be undertaken during the hottest and sunniest period of the year. However, as this is not usually practical, plant monitoring can be installed during cooler weather and the observed oversizing adapted to determine the oversizing factor at design conditions.

Generally, the hottest period of the year occurs in either July or August. Basic 5-day weather forecasts are freely available from most media. In addition, the UK weather centres can provide more detailed and longer termed weather forecasts (at a cost) if required.

### Winter monitoring

For a period of minimum VAV load (when all the dampers are at their minimum position):

Install plant monitoring over 2 or 3 days (go to Boxes L and O).

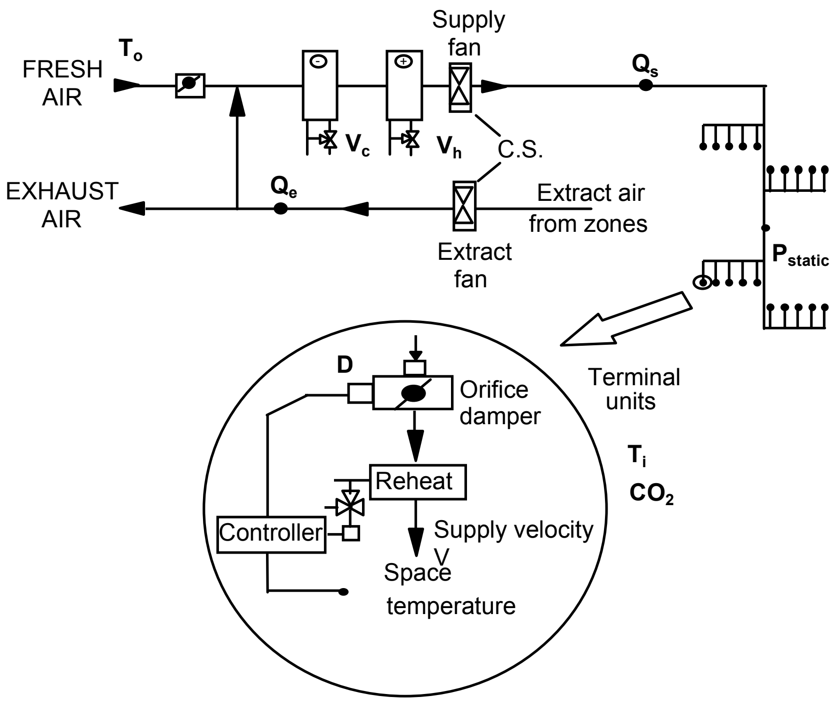



**Figure 19**
Monitoring positions for VAV systems

**At 15 minute intervals**

$T_o$ - Outside air temperature (°C)
CS - Control signal of each fan (% of full capacity).

For speed controlled fans, monitor fan speed.

For other capacity control techniques (eg inlet guide vanes, damper throttle control, pitch adjustment, etc) install position monitoring:

$P_{static}$ - Static total pressure as measured in the duct
$Q_s$ - Flow rate through supply fan ($m^3/s$)
$Q_e$ - Flow rate through extract fan ($m^3/s$).

**For a selected number of terminal units:**

Monitor as many terminal units as possible (ideally 1 in 4) to establish if the velocity flow rate they supply is greater than that required by the maximum cooling demand. $CO_2$ levels must also be monitored to establish if the units are satisfying minimum fresh air requirements. Pick them throughout the system, as representative of the different thermal loads and types of unit installed:

$T_i$ - Internal space temperature (°C)
Vel - Terminal unit air velocity (m/s)
or $D_t$ - Damper position (%).

**Zone $CO_2$ assessment:**

Divide up the zones supplied by the system and assess $CO_2$ concentration at representative points as an indicator of ventilation rate. Measure:
$CO_2$ - Carbon dioxide concentrations in each zone (ppm)

$CO_2$ monitoring must be undertaken over a period of time - spot readings are not representative.

**Coil valve monitoring**

All systems at 15 minute intervals:

$T_o$ - Outside air temperature (°C)
$T_i$ - Internal space temperature (°C)
$V_c$ - Cooling valve position (%)
$V_h$ - Heating valve position (%).

**Assess observed fan oversizing**

Plot supply and extract fan demands and supply air volume, for a day near the end of the hottest summer's week monitored. Complete Box F.

**Box F** Outside air temperature (OAT)

| F1 | |
|---|---|
| Recommended external design temperature[4] | |
| Location | Dry Bulb (°C) |
| Belfast | 23 |
| Birmingham | 26 |
| Cardiff | 25 |
| Edinburgh | 23 |
| Glasgow | 24 |
| London | 29 |
| Manchester | 26 |
| Plymouth | 25 |

| F2 | | |
|---|---|---|
| Peak monitored OAT* | | |
| Date | Time | (°C) |
| | | |

**Average for an afternoon near the end of a hot week.*

**Part 1 - Fans**

The calculation used depends on whether the external temperature is within 2°C or 5°C of the design external temperature.

If F2 is not less than 2°C lower than F1 - complete Box G only.
If F2 is lower than F1 by between 2°C and 5°C then complete Boxes H to L. Boxes H to L adapt the oversizing factor calculated when the OAT is not at the design condition. It cannot be used if the measured external temperature is more than 5°C lower than the design external temperature due to the inaccuracy that prevails outside of this limit. *(NB the calculation Boxes should be completed for both supply and extract.)*

For the monitored period assess the peak external temperature during an afternoon to determine whether the outside air temperature is not less than 2°C below the design external temperature, or whether it is between 2°C and 5°C below the design external temperature. If it is not less than 2°C then complete Boxes G1 to G3. If it is between 2°C and 5°C below the design external temperature complete Boxes G1 and G2 only and proceed to Box H.

**Box G** Assess excess fan capacity

| G1 | G2 | G3 |
|---|---|---|
| Average fan demand** (%) | Fan oversizing (OF) | Possible demand reduction (%) |
| | 100 / G1 | [(G2 -1) / G2] x 100 |
| | | |
| | | |

*** Average monitored control state during an hour of afternoon peak external temperature. If possible, check that the control state of the fan is in proportion to the flow rate by taking 2 or 3 flow rate readings and comparing them to the fan control signal monitored. If there is not a linear relationship between capacity control and flow volume (or control state monitoring is hard to implement), monitor the flow rate at full flow to assess plant oversizing instead. High flow rates can be assessed, for example, when the system is precooling on Monday morning after a hot weekend, or by simulating high internal temperature.*

**Box H** Assessment when peak OAT is between 2°C and 5°C below external design temperature

| H1 | | | H2 | H3 |
|---|---|---|---|---|
| Monitored OAT* [when OAT< (design OAT - 2°C) and OAT > (design OAT -5°C)] | | | Does IAT = OAT? (°C) (yes/no) | Assumed constant internal temperatures if OAT increases (°C) |
| Date | Time | (°C) | | |
| | | | | |

** From the monitored data find a Thursday or Friday afternoon during occupation when peak outside air temperature (OAT) is equal to internal air temperature (IAT). This is important as the assessment of average fan demand in Box J1 relies on monitoring the system when the peak OAT is equal to the internal temperature, ideally during an afternoon near the end of a hot week when internal heat gains and building fabric temperature have stabilised. This condition represents the amount of casual heat gain to the space. The external temperature should not be more than 5°C below the design external temperature.*

Note: In Box H3 the internal temperature is assumed to be that maintained when the OAT rises above that in Box H1.

**Box J** Assess fan reference condition when Box H2 is satisfied

| J1 | J2 |
|---|---|
| Average fan demand (%) | Fan reference condition (R) |
| | J1 / 100 |
| | |
| | |

Note that the average fan demand signal established in Box G1 above will not be the same as in J1. This is because Box G1 is based on peak outside temperature during the period whereas Box J1 is based on the average fan demand when the internal temperature is the same as outside temperature.

**Box K** Fan excess capacity (OAT below design)

| K1 | K2 | K3 |
|---|---|---|
| Proportion of weather related load (W) | Adjusted oversizing factor ($OF_D$) | Possible demand reduction (%) |
| 1 - (J2 x G2) | G2 x {1 / (1 - K1) + K1 [(F1 - H3) / (F2 -H3)]} | [(K2 -1) / K2] x 100 |
| | | |

Table 9 shows potential savings for fans at design conditions, and below.

**Table 9**
Matrix showing potential savings under different OAT and fan demands

| F1 | F2 | G1 | G2 | G3 | H1 | H3 | J1 | K1 | K2 | K3 |
|---|---|---|---|---|---|---|---|---|---|---|
| 29 | 29 | 90 | 1.1 | **10** | | | | | | |
| 29 | 28 | 82 | 1.2 | **18** | 26 | 25 | 77 | 0.07 | 1.4 | **30** |
| 29 | 27 | 72 | 1.4 | **28** | 25 | 23 | 69 | 0.04 | 1.5 | **35** |
| 29 | 26 | 75 | 1.3 | **25** | 24 | 23 | 65 | 0.13 | 2.0 | **49** |
| 29 | 25 | 48 | 2.1 | **52** | 21 | 20 | 45 | 0.06 | 2.5 | **60** |

F1 - Recommended external design temperature (°C)
F2 - Peak monitored OAT (°C)
G1 - Average fan demand (%)
G2 - Fan oversizing (OF)
G3 - Possible demand reduction (%)
H1 - Temperature for a period when OAT is equal to IAT (°C)
H3 - Assumed constant IAT if OAT for period H1 increases (°C)
J1 - Average fan demand for period used in H1
K1 - Proportion of weather related load (W)
K2 - Adjusted oversizing factor ($OF_D$)
K3 - Actual possible demand reduction (%)

**Box L** Fan base load

| L1 | | L2 | L3 |
|---|---|---|---|
| Period of minimum load* | | Average fan demand (%) | Base load (%) |
| Date | Time | | L2 x G2 or K2 |
| | | | |

**When all dampers are at their minimum position*

**Box M** Annual fan energy consumption

| M1 |
|---|
| kWh* |
| $3640 \times \{1 + [0.75 + (25 / L3)]^3\}$ |
| |

**Power consumption at minimum load (derived assuming that VAV system operates at minimum flow for half VAV system operation). For an accurate assessment, monitor the cumulative fan energy consumption over the year.*

### Part 2 - Terminal units

#### Assessment of oversizing at maximum flow

Note the simultaneous internal space temperature associated with each terminal unit. If the VAV terminal units are not controlled to velocity (or the velocity readings cannot be trusted) then assess terminal unit oversizing for maximum load in terms of damper position. The linearity of damper position and flow can be checked by measuring flows in a sample number of terminal units on 2 or 3 occasions.

**Box N** Assess at maximum flow

| N1 | N2 | N3 |
|---|---|---|
| Velocity at full capacity (m/s)* | Maximum air flow rate (m/s)** | Oversizing factor (OF)*** |
| | | N1 / N2 |
| | | |
| | | |

**As measured when the terminal unit is fully open during a pre-cooling period, (or if necessary in terms of terminal unit design).*
***Recorded during the period highlighted in Box F2 or H1.*
****It is necessary to ensure that OF at maximum flow is not below 1. Otherwise, it is likely that inadequate flow will be provided to the index terminal unit, thus compromising the cooling capability of the system in summer.*

### Assess at minimum flow

**Box O** Assess at minimum flow

| O1 | O2 | O3 | O4 |
|---|---|---|---|
| Maximum indoor $CO_2$ concentration (ppm)* | Monitored internal $CO_2$ concentration (ppm)** | Outside $CO_2$ concentration (ppm)*** | Oversizing factor$_{(min)}$ (OF) |
| | | | (O1 - O3) / (O2 - O3) |
| | | | |

**As specified in design (if in doubt take as 850ppm).*
***As monitored during period specified in Box K1.*
****If not monitored use 400ppm.*

This equation represents the state when $CO_2$ levels peak in the building during full occupancy. The equation will be more accurate for office buildings than for buildings with a larger volume and smaller air supply rate.

### Assess at available savings

Use Table 10 to assess the available savings. Convert the savings into £/annum saved (based on the energy consumption calculated in Box M1) and then consider the remedial measures that can be implemented. Available savings may increase if the maximum monitored flow rate is much lower than the fan design flow rate specified on the fan nameplate. If this is the case, then estimate the fan oversizing and consult the fan manufacturer for the associated reduction in fan maximum efficiency.

**Table 10**
Typical (estimated) savings available for VAV systems

**Speed controlled electrical drive systems**

| BL% (box L) | TU OF (min. flow) | Fan Oversizing Factor 1.1 | 1.2 | 1.35 | 1.5 |
|---|---|---|---|---|---|
| 80% | 0% | 0% | 5% | 8% | 11% |
| 60% | 0% | 2% | 3% | 12% | 20% |
| 80% | 10% | 9% | 16% | 19% | 23% |
| 60% | 10% | 10% | 17% | 20% | 27% |

**Fan blade pitch adjustment (axial fans only)**

| BL% (box L) | TU OF (min. flow) | Fan Oversizing Factor 1.1 | 1.2 | 1.35 | 1.5 |
|---|---|---|---|---|---|
| 80% | 0% | 0% | 11% | 19% | 29% |
| 60% | 0% | 9% | 17% | 27% | 39% |
| 80% | 10% | 11% | 20% | 27% | 36% |
| 60% | 10% | 16% | 24% | 33% | 44% |

**Guide vane adjustment**

| BL% (box L) | TU OF (min. flow) | Fan Oversizing Factor 1.1 | 1.2 | 1.35 | 1.5 |
|---|---|---|---|---|---|
| 80% | 0% | 11% (33%) | 30% (47%) | 46% (59%) | 58% (68%) |
| 60% | 0% | 18% (59%) | 36% (68%) | 50% (75%) | 62% (81%) |
| 80% | 10% | 18% (59%) | 36% (68%) | 50% (75%) | 62% (81%) |
| 60% | 10% | 20% (62%) | 38% (70%) | 51% (77%) | 63% (82%) |

**Damper adjustment**

| BL% (box L) | TU OF (min. flow) | Fan Oversizing Factor 1.1 | 1.2 | 1.35 | 1.5 |
|---|---|---|---|---|---|
| 80% | 0% | 18% (52%) | 37% (63%) | 52% (72%) | 63% (78%) |
| 60% | 0% | 18% (59%) | 36% (68%) | 51% (83%) | 62% (87%) |
| 80% | 10% | 21% (58%) | 38% (67%) | 53% (75%) | 64% (81%) |
| 60% | 10% | 22% (74%) | 39% (80%) | 53% (85% | 64% (88%) |

***Note to table:***

1. *The savings available present approximate savings from increased system efficiency and fan dynamic efficiency and they should only be considered as a guideline. They are derived assuming a load profile of minimum flow for 6 months of the year and are based on a correctly sized fan being installed. Other combinations (for example using the original fan but reducing minimum flow) can be extrapolated for savings as necessary.*
2. *The figures do not take into account savings associated with reductions in thermal loads and maximum demand charges.*
3. *The figures in brackets represent the savings if the current capacity controlled method was replaced by a speed controlled electrical drive system.*

## Case study 62: VAV system

An 11 kW supply fan and 7.5 kW extract fan (both using variable speed drives) serve 17 VAV terminal units responsible for 240 $m^2$ of fully occupied office space. The VAV system has been designed to provide 7.2 $l/s/m^3$. In comparison to the capacity yardstick on Page24 (and even accounting for the cooling demand) this suggests that the system is oversized.

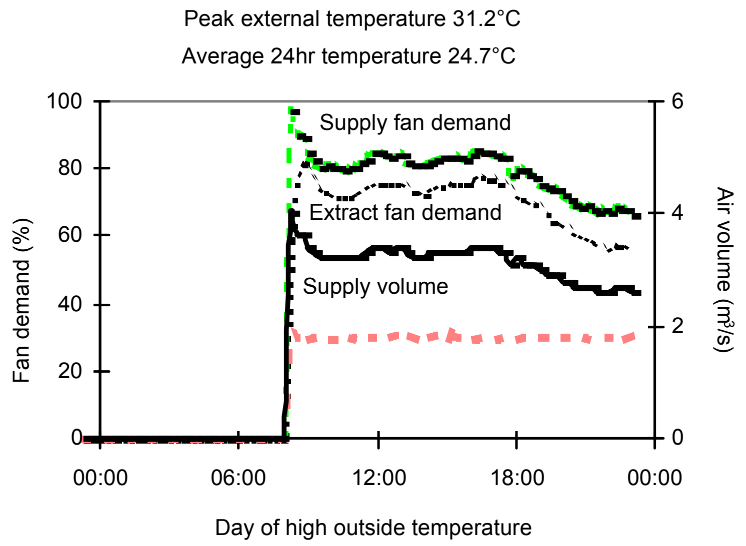



**Figure 20**
VAV system performance on a hot day

Figure 20 plots the system characteristics for the hottest day monitored. It shows that during the afternoon the supply fan demand peaked at 85%, and the extract fan peaked at 77% of full speed. This indicates that for these conditions (from *Box* G2):

The supply fan oversizing factor = 1.18
The extract fan oversizing factor = 1.39
The terminal units monitored were also found to be oversized.

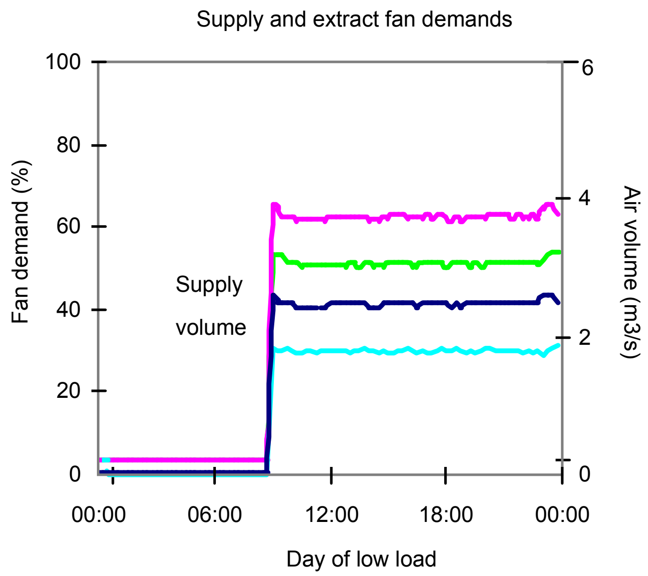



**Figure 21**
VAV system performance at low load

Figure 21 presents the same monitored parameters but for a period of minimal VAV system operation. The base load expressed as a percentage of maximum load (from *Box* L) is:

BL% = 59% × 1.18 = 70%.

At this base load, with all the terminal units in their minimum positions (and also the system operating at minimum fresh air of 10%), the internal $CO_2$ levels (at full occupancy) were measured as 690ppm with outside $CO_2$ levels of 400ppm during the afternoon. The terminal units can therefore be considered to be approximately 1.6 times oversized for minimum flow fresh air rate if the maximum internal $CO_2$ limit is specified as 850ppm (see Box O).

It is estimated that if the system flow was reduced by 10% at minimum load (even though the original fans are kept), approximately 10% savings can be achieved by adjusting the terminal unit schedules and the system control static pressure set-point.

### Part (C) Cooling and heating coils

A coil and its control valve should act in unison. If the valve and coil are equally oversized, the valve position is a good indicator of the percentage of coil capacity used (assuming an equal percentage valve is used). However, if the valve acts more like an on/off controller across all loads, not offering uniform sensitivity as it should, then it is likely that the valve is oversized in comparison to the coil. A correctly sized valve must first be installed before coil capacity can be assessed. Consult the valve supplier for expert advice. Also see Heating Plant section for more details.

### Install plant monitoring

(a) During summer (when the building is fully occupied and functional):

Install coil monitoring as outlined in Figure 18 and Figure 19; where:

$V_c$ - cooling valve position
$V_h$ - heating valve position

log for at least 7 days. Plot the coil demand (valve position) for a day at the end of the hottest/coldest week monitored.

(b) For a period of minimum coil demand (generally during autumn/spring):

Install coil monitoring for 2 or 3 days and then assess low load coil performance by completing Box P.

**Box P** Assess coil oversizing

| P1 | P2 |
|---|---|
| Average coil valve position (%)* | Coil oversizing factor (OF) |
| | 100 / P1 |
| | |

**Taken during an hour of most extreme afternoon external temperature. This assumes that valve position is an accurate indicator of coil output - even though this is the basis of design, this may not be true in practice.*

### Assess low load coil performance

The consequence of oversizing coils and associated valves is that unstable control can occur at low loads. Plot the action of the coil valves and assess coil performance for a day of low load. Consider remedial measures if the performance has been compromised.

The coils are likely to be undersized if the internal temperature is not maintained under highest load.

**Box Q** Winter-time coil monitoring

| Q1 Recommended external design temperature[4] (°C) | | |
|---|---|---|
| Location | Low Thermal Inertia | High Thermal Inertia |
| Belfast | -5.0 | -4.0 |
| Birmingham | -7.5 | -5.5 |
| Cardiff | -5.5 | -4.0 |
| Edinburgh | -7.0 | -6.0 |
| Glasgow | -6.5 | -4.5 |
| London | -5.5 | -4.5 |
| Manchester | -6.0 | -5.0 |
| Plymouth | -3.5 | 2.5 |

*Low thermal inertia - typically, single storey buildings.*
*High thermal inertia - typically multi-storey buildings with solid intermediate floors and partitions.*

| Q2 |
|---|
| Monitored OAT* (°C) |
| |
| |

** Average OAT for period used in Box P*

Is Q2 lower or equal to Q1? — Yes → Oversizing Factor (OF) = P2

No ↓

Is Q2 higher than Q1 but within 2°C? — No → Go back to Box P1 using data from a colder day. Re-monitor if required.

Yes ↓

| Q3 | Q4 | Q5 |
|---|---|---|
| Space Temperature (°C) | Base External Air Temperature (°C) | Correction Factor |
| | Q3 - 3.5 | $\frac{(Q4 - Q2)}{(Q4 - Q1)}$ |
| | | |

| Q6 |
|---|
| Adjusted OF |
| P2 x Q5 |
| |

**Box R** Summer-time coil monitoring

| R1 | |
|---|---|
| Recommended external design temperature[4] (°C) | |
| Location | Dry Bulb (°C) |
| Belfast | 23 |
| Birmingham | 26 |
| Cardiff | 25 |
| Edinburgh | 23 |
| Glasgow | 24 |
| London | 29 |
| Manchester | 26 |
| Plymouth | 25 |

| R2 |
|---|
| Monitored OAT* (°C) |
| |
| |
| ** Average OAT for period used in Box P* |

Is R2 higher or equal to R1? — Yes → Oversizing factor (OF) = P2

No ↓

Is R2 lower than R1 but within 2°C? — No → Go back to Box P1 using data from a warmer day. Re-monitor if required.

Yes ↓

**Box S** OF Adjustment

*From the monitored data find a Thursday or Friday afternoon, during occupation when Peak OAT is equal to internal air temperature (IAT).*

| S1 | S2 | S3 |
|---|---|---|
| Temperature when OAT = IAT (°C) | Average coil valve position (%) | Reference condition |
| | | S2 / 100 |
| | | |

| S4 | S5 |
|---|---|
| Average internal temperature when OAT > F1 (°C) | Proportion of weather related load (W) |
| | 1 - (S3 x P2) |
| | |

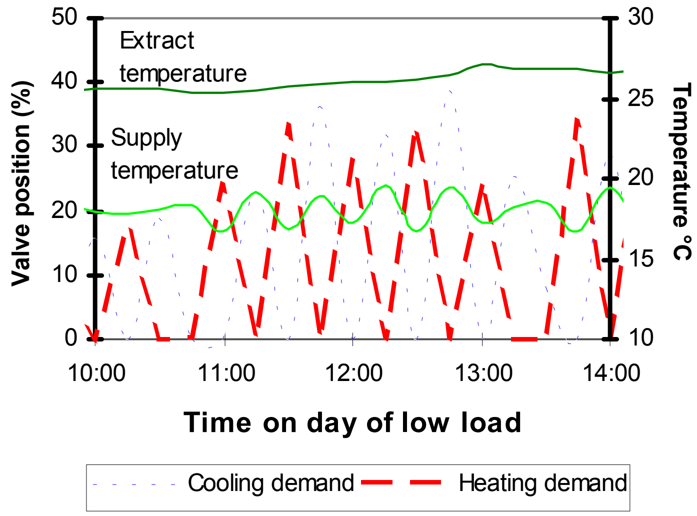
| S6 |
|---|
| Adjusted Oversizing Factor ($OF_D$) |
| P2 x {1 / (1 - S5) + S5 [(R1 - S4) / (R2 - S4)]} |
| |

**Case Study 63: Coil performance**

The performance of the cooling coil used in a restaurant air handling unit was monitored during a hot summer day (peak external temperature: 31.2°C). The investigation found that the coil control valve never fully opened during the afternoon despite normal occupancy levels and the high external temperature. The measured average load for this day was significantly smaller than full capacity (at 55%) which indicates oversizing.

Figure 22 illustrates the poor control of the cooling and heating coils for this AHU at low load conditions. Control is nearly reduced to on/off - both the heating and cooling coil valves cycle as the supply temperature under-and over-shoots the set-point required. Clearly energy is wasted, probably due to coil oversizing. Further re-tuning of the control loop during low load conditions would improve performance.

**Figure 22**
Heating and cooling coil performance on day of low load




## 4.4 Remedial Measures of Air Handling Systems

The objective of implementing remedial measures is to reduce energy consumption and to improve system control and occupant comfort. Reducing energy cost is mainly achieved by targeting fans when Box E (available savings) has shown that savings are available. As a rule of thumb, it is usually worth implementing remedial measures if CAV fans are oversized by 10% or more and consume at least 6 kWe (operating throughout the year). Oversizing factors above 10% are worth targeting for smaller CAV fans. VAV fans that use guide vanes are worth retrofitting with a variable speed drive (VSD) if they are oversized by at least 10% and consume 10 kWe or more at maximum demand. When undertaking changes to ventilation air flow rates it is important to check the overall air flow movement, which may be configured to maintain areas of positive, negative or balanced air pressure, or to limit the spread of smoke in the event of a fire.

Improving cooling/heating coil performance is also good practice if low load control is difficult. Benefits include fewer occupant complaints and reduced energy consumption if cycling of the control valve occurs.

In general terms, if plant is old and in state of disrepair, then plant replacement will clearly improve performance and provide significant savings (never replace oversized plant on a like-for-like basis - if necessary, tag plant as oversized when plant replacement arise). However, significant savings can also arise from the following simple low cost measures, if appropriate.

### 1. Ensure fans are off when they are not required

Introduce a time schedule, with plant override, so that fans only operate during occupantion. Alternatively, fan operation can be controlled by an inexpensive occupancy sensor if occupancy is unpredictable.

### 2. Reduce flow rate through the CAV system

Both supply and extract fans should be de-rated to reduce the airflow through the system to meet actual requirements (see ‘Practical considerations’ Section 4.4.1). The system should either remain constant volume or be converted to a multi-volume system, by installing a VSD or multiple speed motor, controlled to demand by, for example, using a space $CO_2$ sensor, (see ‘Measures to reduce fan output’in Section 4.4.1).

### 3. Optimise fan efficiency (see ‘Increasing fan efficiency’)

(For VAV systems, introducing a more efficient load control method will increase available savings.)

### 4. Reduce VAV system control static pressure set-point

It is good practice to minimise the system static set-point so that the terminal unit orifice dampers are as open as possible (as long as an adequate pressure is maintained at all terminal units).

Such practice promotes:

a) Improved VAV terminal unit control and occupant comfort.
b) Reductions in fan power as pressure requirements drop.

From knowledge of measured fan and terminal unit oversizing, fix the required static pressure and adjust the terminal unit control schedule (see Figure 23) so that the maximum airflow required coincides with maximum terminal unit damper position (see ‘Practical considerations’). Consult VAV system manufacturers if required.

### 5. Reduce air flow to space when VAV terminal unit dampers are at their minimum position

If the terminal unit oversizing occurs at minimum flow rate (Box O), consider adjusting the terminal unit control schedules (see Figure 23) to reduce the air flow rate at its minimum setting. The system static pressure set-point can often be reduced, with savings in fan energy consumption and system thermal loads. However, when system flow rates are reduced at minimum demand, system balancing may be affected and so the index terminal unit may become starved of air and thus unable to maintain satisfactory air distribution in the space (see ‘Practical considerations’). If the system load is unbalanced (for example where some terminal units are oversized and others are undersized) system performance can be improved by reconfiguring the system. Re-commission the system as necessary, (see ‘Practical considerations’).

### 6. Improve cooling/heating coil performance

a) The control loop can be fine-tuned to improve low load performance. This should only be undertaken by suitable personnel.
b) Replace the coil/valve with a correctly sized system. The coil and valve should be redesigned and replaced together for effective control.

Oversized two-port valves should be replaced to match coil output. Assess the size by correlating the oversizing factor of the system (Box P2) with spot measurements of flow rate and temperatures across the coil to establish coil capacity. Consult coil manufacturer.

#### 4.4.1 Practical considerations of air handling systems

**System balancing:** If the air flow rate is reduced or the system configuration is altered (such as replacing a selection of terminal units), system balancing may be affected and certain zones may be starved of air. Refer to BSRIA *Commissioning of air systems in buildings*[15] for checking air system balancing. Any rebalancing that is necessary should only be undertaken by suitable personnel and only if absolutely required.

**Variable speed drives (VSDs):** Inverters used in VSDs can generate harmonic currents (due to their non-linear nature) that distort the voltage in the mains supply. Utilities can require heavy polluters (in hi-tech areas) to clean up the supply by installing filters or traps to restrict the harmonics. Some latest generation VSD's use a 12 pulse inverter drive or better still harmonic cancelling drives which significantly reduce the likelihood of harmonic production. If harmonics are likely to be an issue, then such VSD's should be considered, consult BSRIA publication AG 2/000, *The BSRIA power quality guide*[16]. Otherwise another variable speed device, other than VSD, should be chosen.

**VAV terminal units:** The minimum flow setting of a VAV terminal unit can only be reduced so long as the good air flow into the space is maintained. A well designed terminal unit is still able to be effective at distributing air (ie sustaining the Coanda effect) at turn-down ratios of approximately 40% of design flow. Consult the terminal unit supplier about terminal unit performance if minimum flow rates are reduced to below 40% of terminal unit capacity. Also establish whether any methods can enhance this air distribution. Air distribution from terminal units can be checked by using a simple smoke test.

**Measures to reduce fan output**

- modify the pulley size on belt driven fans
- install a VSD to the fan motor and control fan speed to meet demand
- install a correctly sized fan/motor.

**Increasing fan efficiency**

The EEO *Good Practice Guide 2*[17] gives guidance notes on reducing the energy consumption of motors and their drive systems. Poor low load performance associated with oversized motors on fans can be improved by a number of techniques, such as:

**Reducing available motor power and iron losses by using a delta to star reconnection** (savings are typically available (for minimum cost) if the motor is oversized by at least 70%).

**Using motor voltage controllers** (typically increased efficiency when operating below 50% of rated output).

**Using variable speed systems** (multi-speed motors, variable drive systems).

**Correctly sized motor replacement** (High efficiency motors are 2% to 3% more efficient than their standard counterparts.) Make reference to *Good Practice Guide 2*[13] or other detailed guidance material.

**Figure 23**
VAV Terminal unit control schedule

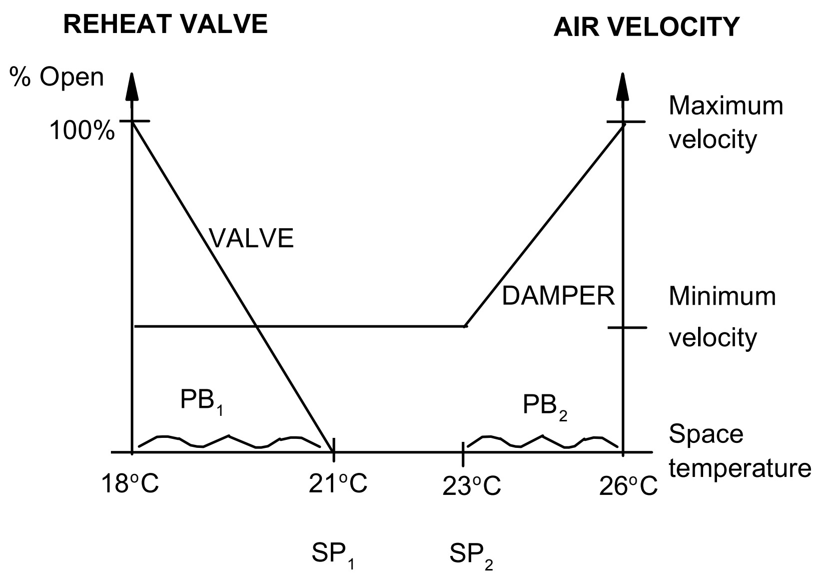



Figure 23 illustrates a simple terminal unit control schedule with space temperature, in terms of reheat valve position, and terminal unit air velocity (or damper position). In order to adjust the schedule, determine the heating and cooling set-points and proportional bands required (SP1 and SP2, and PB1 and PB2 respectively) so that the terminal unit is as open as possible during maximum load, and as closed as possible on minimum demand while still maintaining adequate air distribution. Fine-tune the control loop for stable control and co-ordinate with the supply temperature schedule.