Subject :Air-Conditioning system design	**موضوع : تصميم منظومات تكييف**
Weekly Hours : 2 Theoretical	**الساعات الأسبوعية : نظري : 2**
Tutorial :	**مناقشة :**
Experimental: 2	**عملي :2**
UNITS:6	**الوحدات :6**

week	**Contents**	**المحتويات**	**الأسبوع**
1.	**Principle of Air-Conditioning system design.**	مبادىء تصميم منظومات التكييف.	1.
2.	**Advanced Psychometric of Air Conditioning processes.**	تطبيقات متقدمة على مخطط المصردي.	2.
3.	**The Choice of Supply Design Conditions**	اختيار ظروف التجهيز التصميمية	3.
4.	**Ducting Design**	تصميم منظومات مجاري الهواء.	4.
5.	**Static Regain Method**	منظومة أعادة الكسب الاستاتيكي	5.
6.	**Room Air Distribution**	توزيع الهواء في الغرف.	6.
7.	**Air Handling Unites**	وحدة معالجة الهواء	7.
8.	**Fans**	المراوح	8.
9.	**Noise Sources in air conditioning systems**	مصادر الضوضاء في منظومات تكييف الهواء	9.
10.	**Filtration**	المرشحات	10.
11.	**Evaporative air cooling system.**	منظومات التبريد ألتبخيري	11.
12.	**Piping system design.**	تصميم منظومات الأنابيب.	12.
13.	**Valves and Fitting.**	الصمامات والملحقات	13.
14.	**Types of Piping system**	أنواع منظومات الأنابيب	14.
15.	**Water Pumps.**	مضخة الماء	15.
16.	**Air Conditioning systems introduction**	مقدمة لمنظومات تكييف الهواء	16.
17.	**All air system design**	تصميم منظومات الهواء الكلية.	17.
18.	**Single Zone Systems**	منظومات المنطقة الواحدة	18.
19.	**Multi – Zone Systems**	منظومات المناطق المتعددة	19.
20.	**Variable air volume systems**	منظومات حجم هواء المتغير	20.
21.	**All water system design.**	تصميم منظومات الماء الكلية.	21.
22.	**Fan – Coil Unit**	وحدة المروحة والملف	22.
23.	**Air-water system design..**	تصميم منظومات الماء - هواء.	23.
24.	**Induction System**	المنظومة الحثية	24.
25.	**DX-Systems , Packaged Units**	منظومات التمدد المباشر الوحدات المجمعة	25.
26.	**Cooling and freezing of food system**	منظومات تبريد وتثليج الأطعمة.	26.
27.	**Industrial building system design.**	منظومات الأبنية الصناعية.	27.
28.	**Public building system design**	منظومات الأبنية العامة.	28.
29.	**Energy Conservation in A/Systems**	ترشيد الطاقة في منظومات التكييف.	29.
30.	**Heat Recovery Systems**	أنظمة استرداد الطاقة	30.

Subject :Air-Conditioning system design (Tutorial)	**موضوع : تصميم منظومات تكييف (عملي)**
Weekly Hours : - Theoretical	**الساعات الأسبوعية : نظري :**
Tutorial :	**مناقشة :**
Experimental: 2	**عملي :2**
UNITS:2	**الوحدات :2**

week	Contents	المحتويات	الأسبوع
1.	Principle of Air-Conditioning system design.	مبادىء تصميم منظومات التكييف.	.1
2.	Year project.	المشروع السنوي.	.2
3.	Chose the project area, location.	اختيار منطقة المشروع.	.3
4.	Type of Air-Conditioning systems.	أنواع منظومات تكييف الهواء.	.4
5.	Ventilation equipment.	معدات التهوية.	.5
6.	Filtration equipment.	معدات التنقية.	.6
7.	All air system (project 1).	منظومات الهواء الكلية (مشروع 1).	.7
8.	(Project 1).	(مشروع 1).	.8
9.	(Project 1).	(مشروع 1).	.9
10.	All water system (project 2).	منظومات الماء الكلية (مشروع 2).	.10
11.	(Project 2).	(مشروع 2).	.11
12.	(Project 2).	(مشروع 2).	.12
13.	Air-water system (project 3).	منظومات الماء – هواء (مشروع 3).	.13
14.	(Project 3).	(مشروع 3).	.14
15.	(Project 3).	(مشروع 3).	.15
16.	Ventilation system (project 4).	منظومات التهوية (مشروع 4).	.16
17.	(Project 4).	(مشروع 4).	.17
18.	(Project 4).	(مشروع 4).	.18
19.	Evaporative system (project 5).	المنظومات التبخيرية (مشروع 5).	.19
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21.	System selection.	اختيار المنظومة.	.21
22.	Ducting of the year project.	منظومة مجاري الهواء للمشروع السنوي.	.22
23.	Ducting of the year project.	منظومة مجاري الهواء للمشروع السنوي.	.23
24.	Piping system of the year project.	منظومة الأنابيب للمشروع السنوي.	.24
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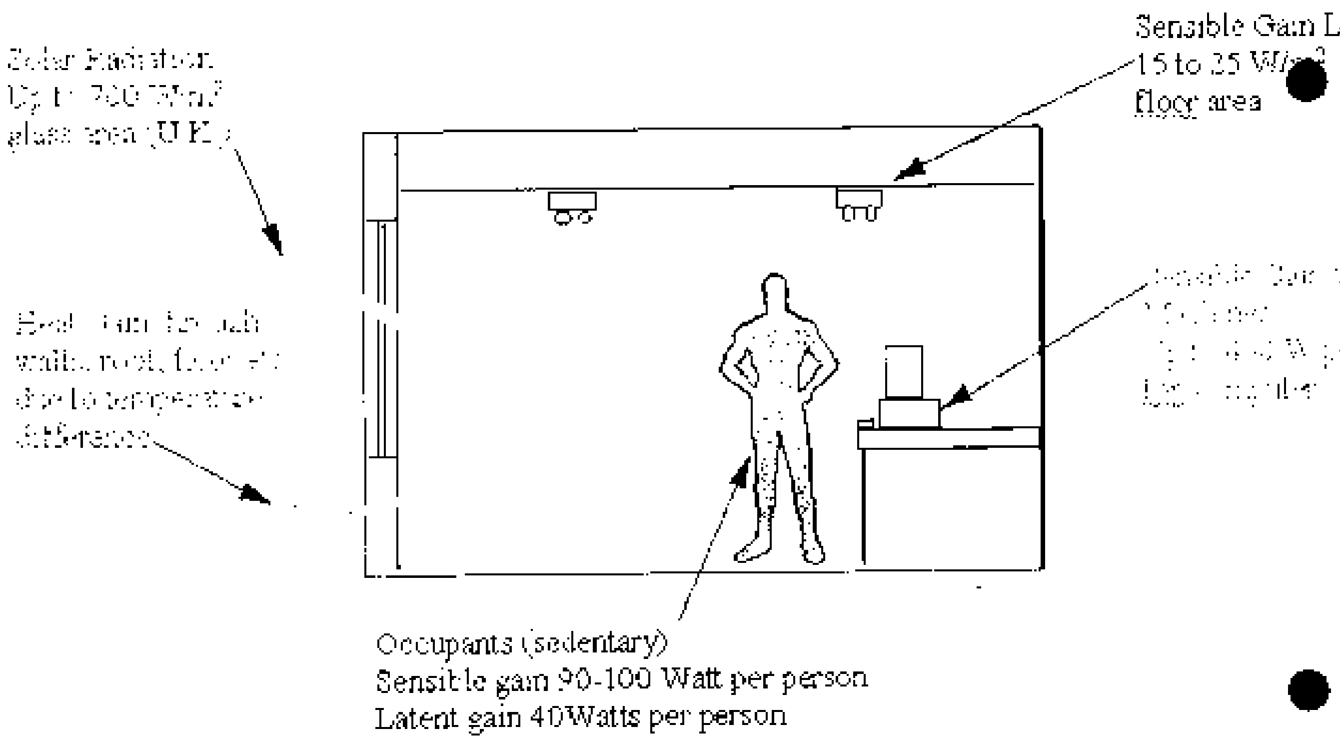
26.	Control system of the year project.	نظام السيطرة للمشروع السنوي.	.26
27.	Control system of the year project.	نظام السيطرة للمشروع السنوي.	.27
28.	Examination.	الامتحان.	.28
29.	Examination.	الامتحان.	.29
30.	Examination.	الامتحان.	.30

AIR CONDITIONING

Introduction to Air Conditioning

Air conditioning may be required in buildings which have a high heat gain and as a result a high internal temperature. The heat gain may be from solar radiation and/or internal gains such as people, lights and business machines.

The diagram below shows some typical heat gains in a room.



If the inside temperature of a space rises to about 25°C then air conditioning will probably be necessary to maintain comfort levels. This internal temperature (around 25°C) may change depending on some variables such as:

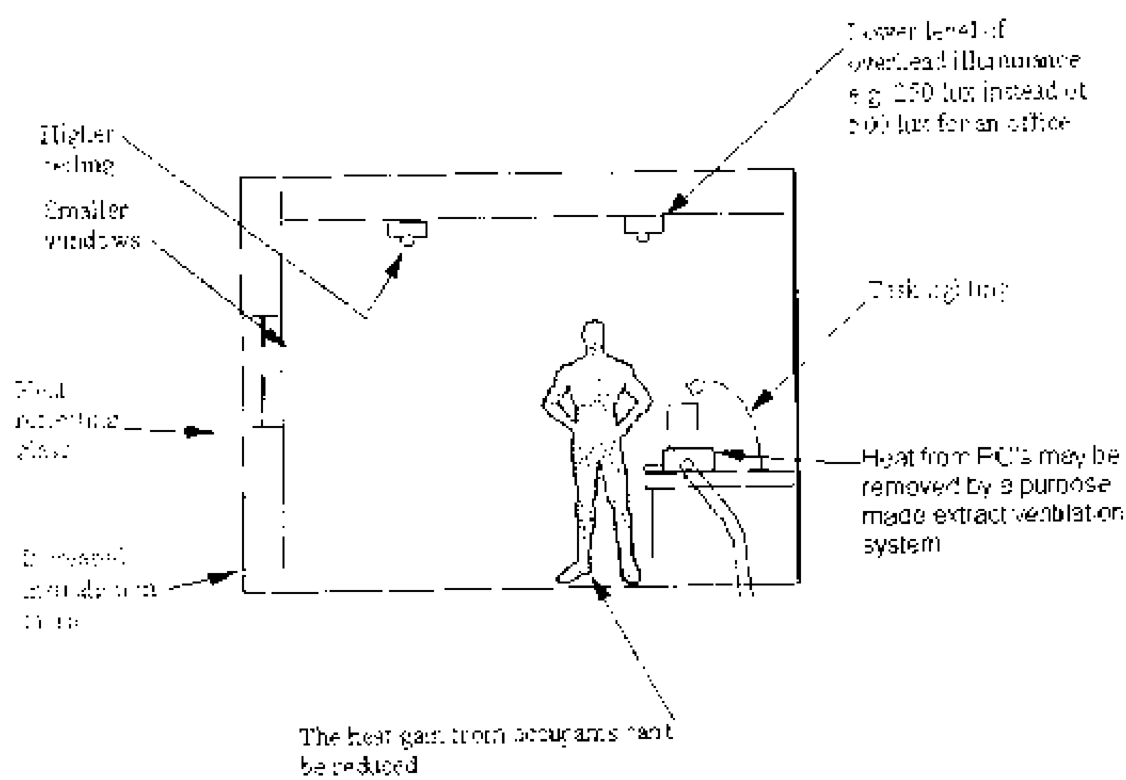
- type of building
- location of building
- duration of high internal temperature
- expected comfort conditions.
- degree of air movement
- percentage saturation

In some buildings it may be possible to maintain a **comfortable environment** with mechanical ventilation but the air change rate will tend to be high (above about 8 air changes per hour) which can in itself cause air distribution problems.

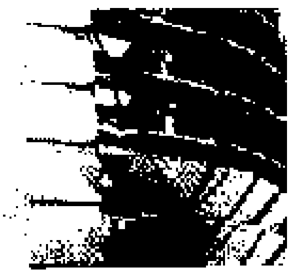
Since air conditioning is both **expensive** to install and maintain, it is best avoided if possible. This may possibly be achieved by careful building design and by utilising methods such as:

- window blinds or shading methods
- heat absorbing glass
- heat reflecting glass
- openable windows
- higher ceilings
- smaller windows on south facing facades
- alternative lighting schemes

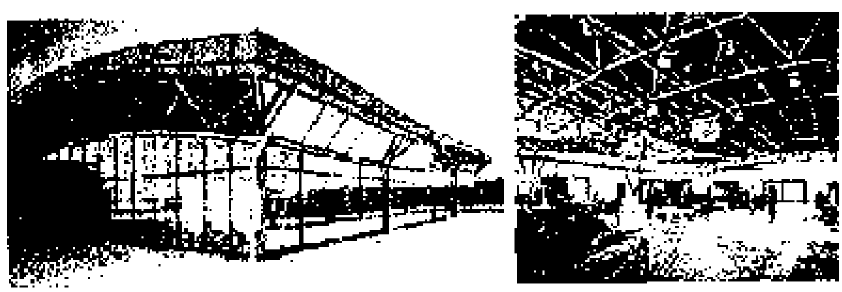
The diagram below shows some of these methods.



FIXED EXTERNAL SOLAR SHADING



MOVABLE FACADE



USE OF SOLAR GLASS IN CAR SHOW ROOM

INTERNAL BLINDS

If air conditioning is the only answer to adequate comfort in a building then the main **choice of system** can be considered.

Full comfort air conditioning can be used in summer to provide cool air (approx. 13°C to 18°C) in summer and warm air (approx. 28°C to 36°C) in winter. Also the air is cleaned by filters, dehumidified to remove moisture or humidified to add moisture.

Air conditioning systems fall into three main categories, and are detailed in the following pages;

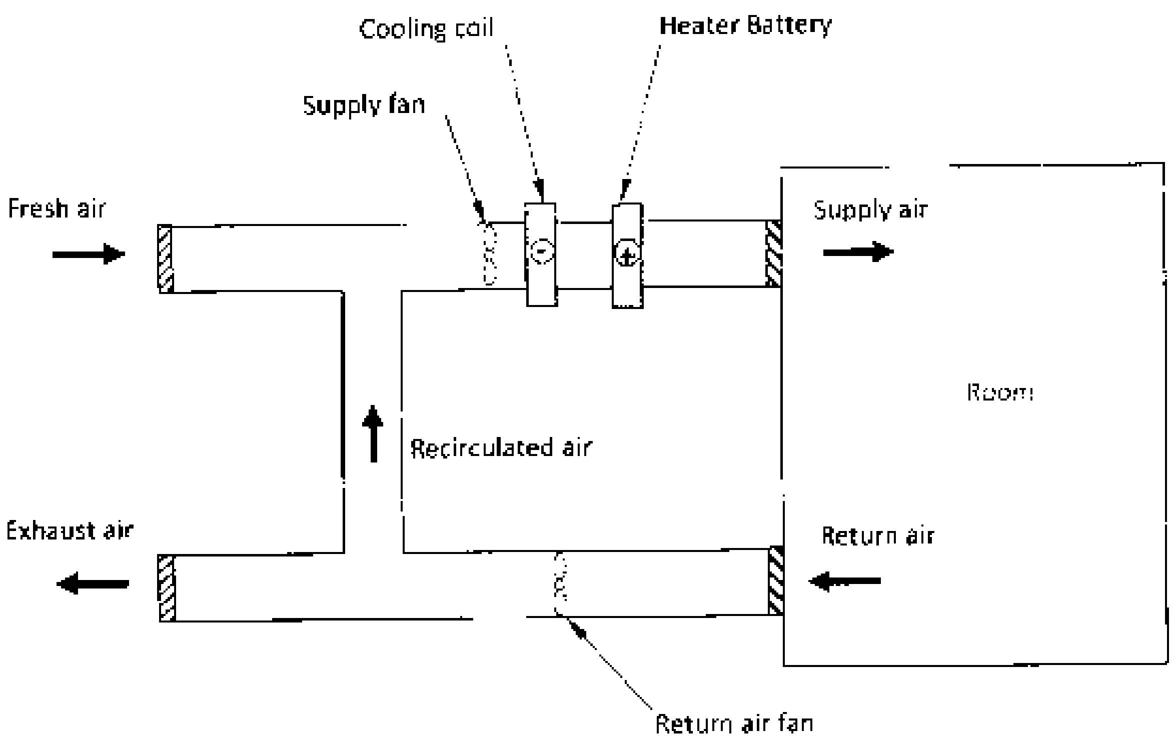
1. Central plant systems.
2. Room air conditioning units.
3. Fan coil units.

Central plant systems have one central source of conditioned air which is distributed in a network of ductwork. **Room air conditioning units** are self-contained package units which can be positioned in each room to provide cool air in summer or warm air in winter.

Fan coil units are room units and incorporate heat exchangers piped with chilled water and a fan to provide cool air.

1.0 Central Plant Systems

A typical central plant air conditioning system is shown below.



Schematic Diagram of Central Plant Air Conditioning System

The system shown above resembles a **balanced ventilation** system with plenum heating but with the addition of a cooling coil.

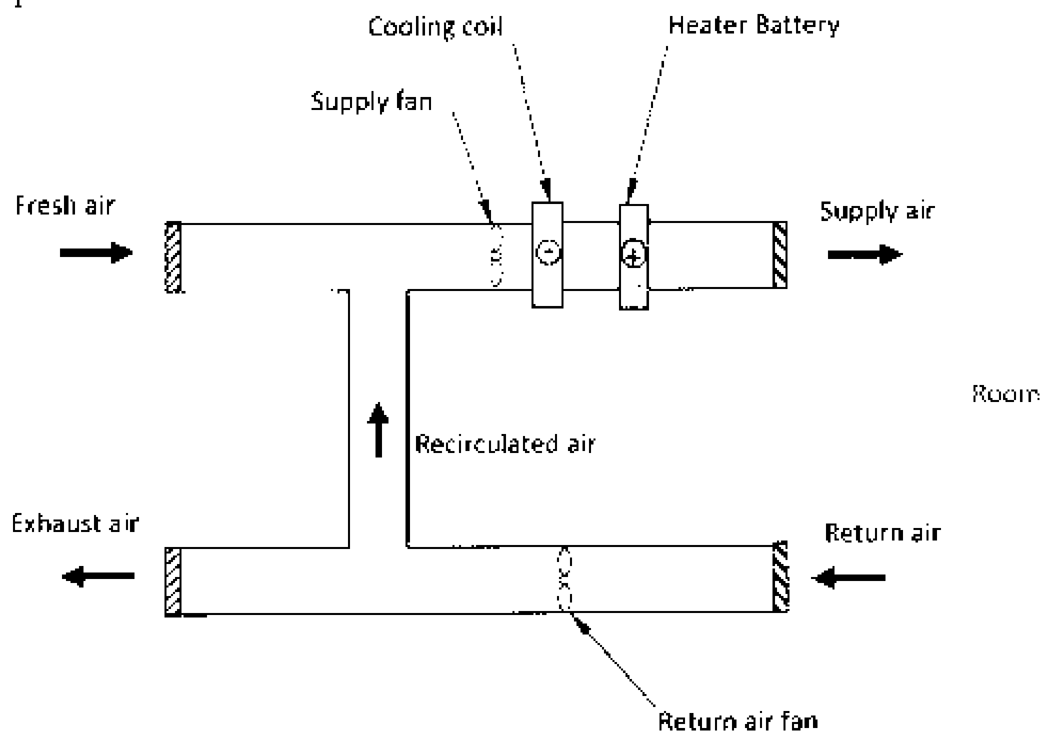
For information on balanced ventilation see VENTILATION section.

In winter the heater battery will be on and the cooling coil will probably be switched off for the majority of buildings. In summer the heater battery will not need to have the same output and the cooling coil will be switched on.

A humidifier may be required to add moisture to the air when it is 'dry'. This is when outdoor air has a low humidity of around 20% to 30%.

In dryer regions humidification is required through most of the year whereas in tropical air conditioning one of the main features of the system is the ability to remove moisture from warm moist air.

Dampers are used in air conditioning central plant systems to control the amount of air in each duct. It is common to have 20% fresh air and 80% recirculated air to buildings. In buildings with high occupancy the fresh air quantity should be calculated based on C.I.B.S.E. data., this may require a higher percentage of fresh air (i.e. more than 20%). See Ventilation section for examples of fresh air rates.



Schematic Diagram of Central Plant Air Conditioning System

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In the U.K. low humidities are rare and therefore humidification is sometimes not used.

In dryer regions humidification is required through most of the year whereas in tropical air conditioning one of the main features of the system is the ability to remove moisture from warm moist air.

Filters are required to remove particles of dust and general outdoor pollution.

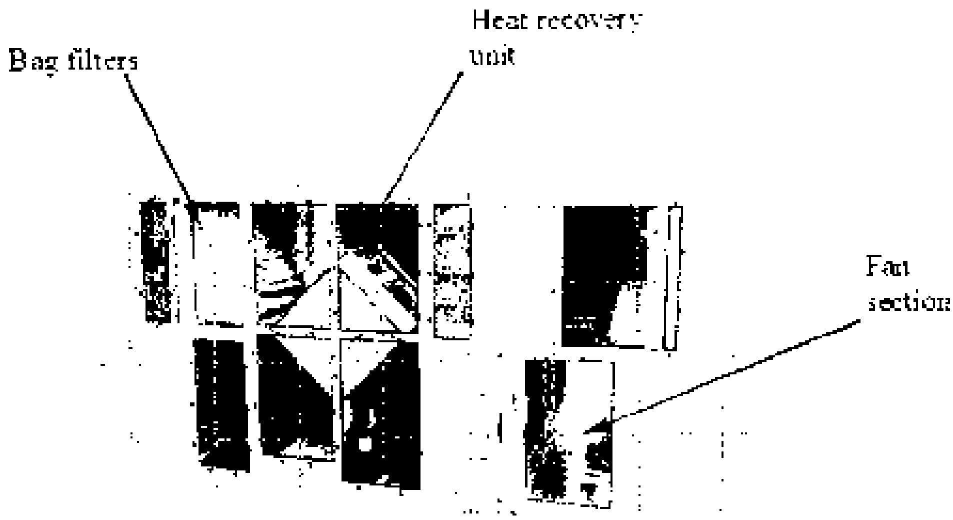
This filter is sometimes called a coarse filter or pre-filter.

A removable fibreglass **dust** filter is positioned in the fresh air intake duct or in larger installation an oil filled viscous filter may be used.

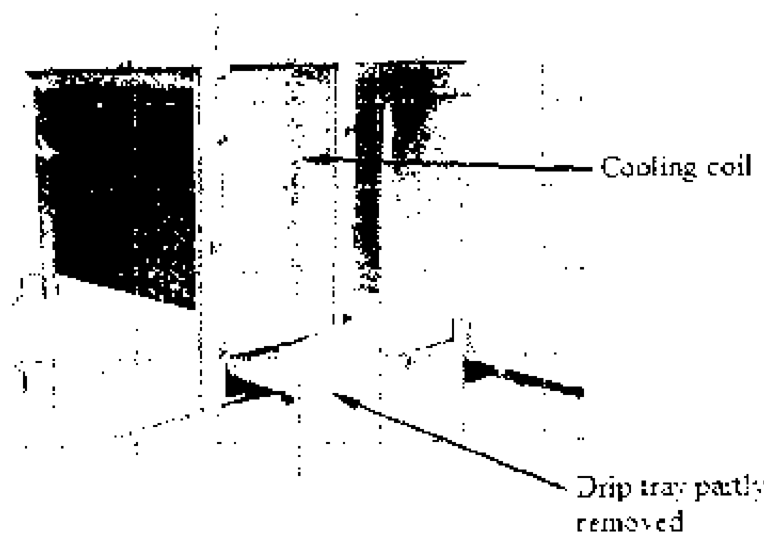
The secondary filter, after the mix point, is used to remove fine dust particles or other contaminant picked up in the rooms and recirculated back into the plant. A removable **bag filter** is generally used for this where a series of woven fibre bags are secured to a framework which can be slid out of the ductwork or air handling unit (A.H.U.) for replacement.

[illegible]

Air handling units (A.H.U.) are widely used as a package unit which incorporates all the main plant items as shown below. Pipework, ductwork and electrical connections are made after the unit is set in place on site. Since air conditioning plant rooms tend to be at roof level, the larger A.H.U.'s are lifted into place by crane before the roof is fixed.



AIR HANDLING UNIT WITH DOORS REMOVED



AIR HANDLING UNIT COOLING COIL SECTION

2.0 Room Air Conditioning Units

These units use refrigerant to transfer cooling effect into rooms.

Room air conditioning units fall into two main categories:

1. Split type

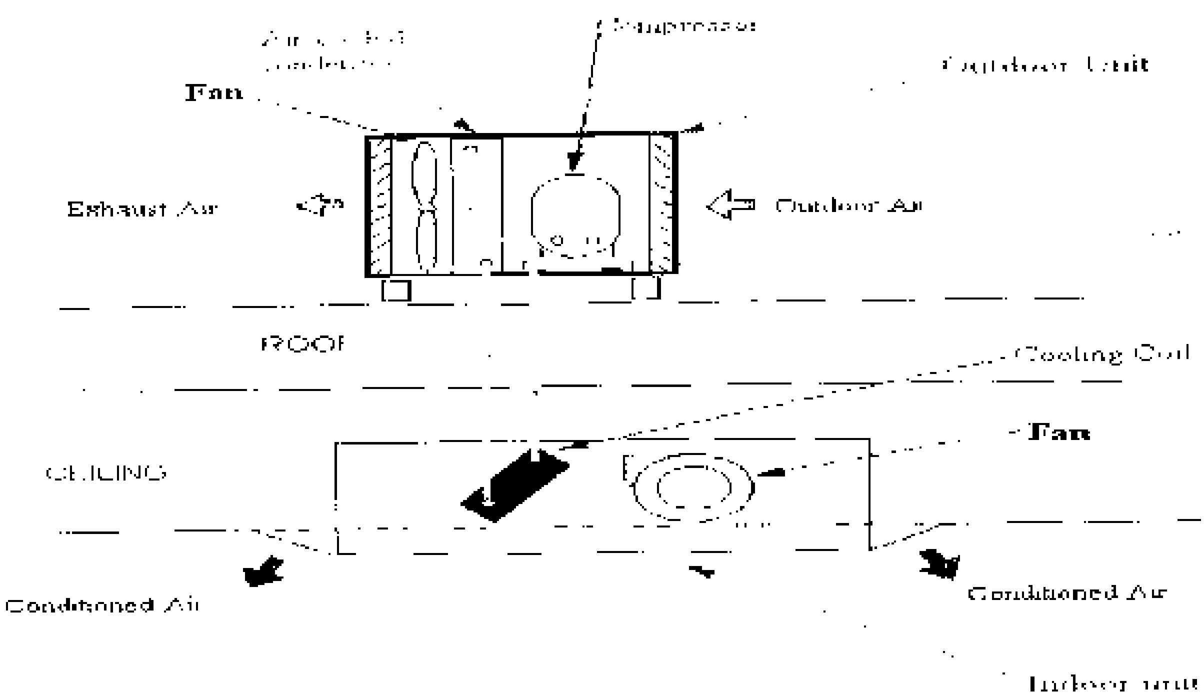
2. Window/wall units.

Split Air Conditioners

Split air conditioners have two main parts, the outdoor unit is the section which generates the cold refrigerant gas and the indoor unit uses this cold refrigerant to cool the air in a space.

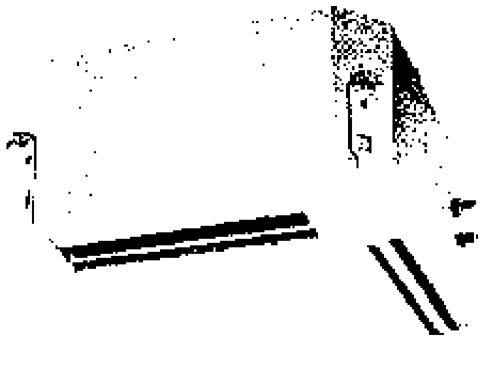
The outdoor unit uses a compressor and air cooled condenser to provide cold refrigerant to a **cooling coil** in the indoor unit. A fan then blows air across the cooling coil and into the room. The indoor unit can either be ceiling mounted (cassette unit), floor mounted or duct type.

The drawing below shows a ceiling mounted (cassette unit).



SPLIT AIR CONDITIOING UNIT

The photographs below show a ceiling mounted cassette and an outdoor unit.

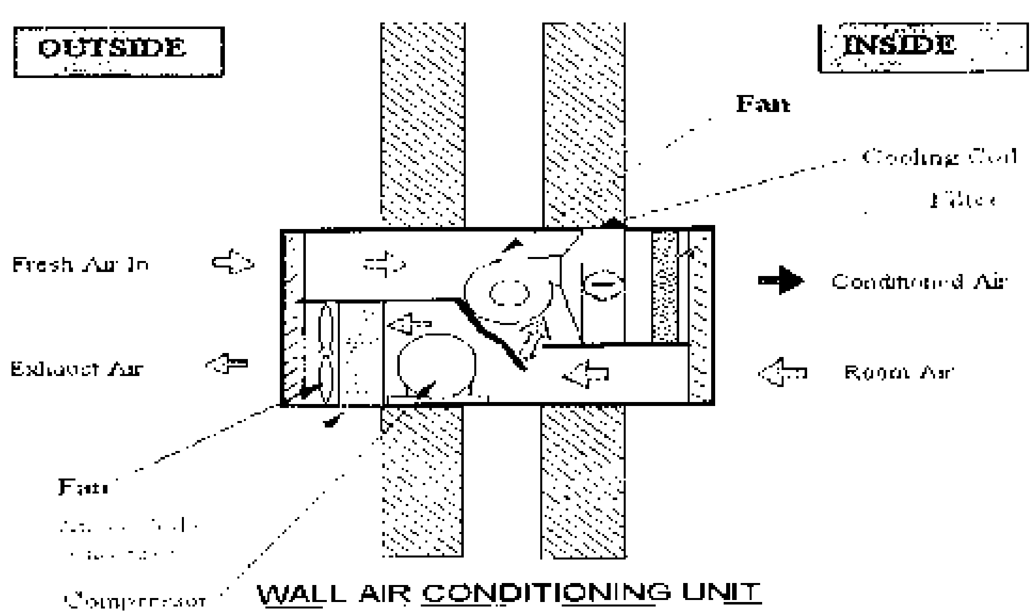


Window / Wall Units

Window or wall units are more compact than split units since all the plant items are contained in one box.

Window units are installed into an appropriate hole in the window and supported from a metal frame.

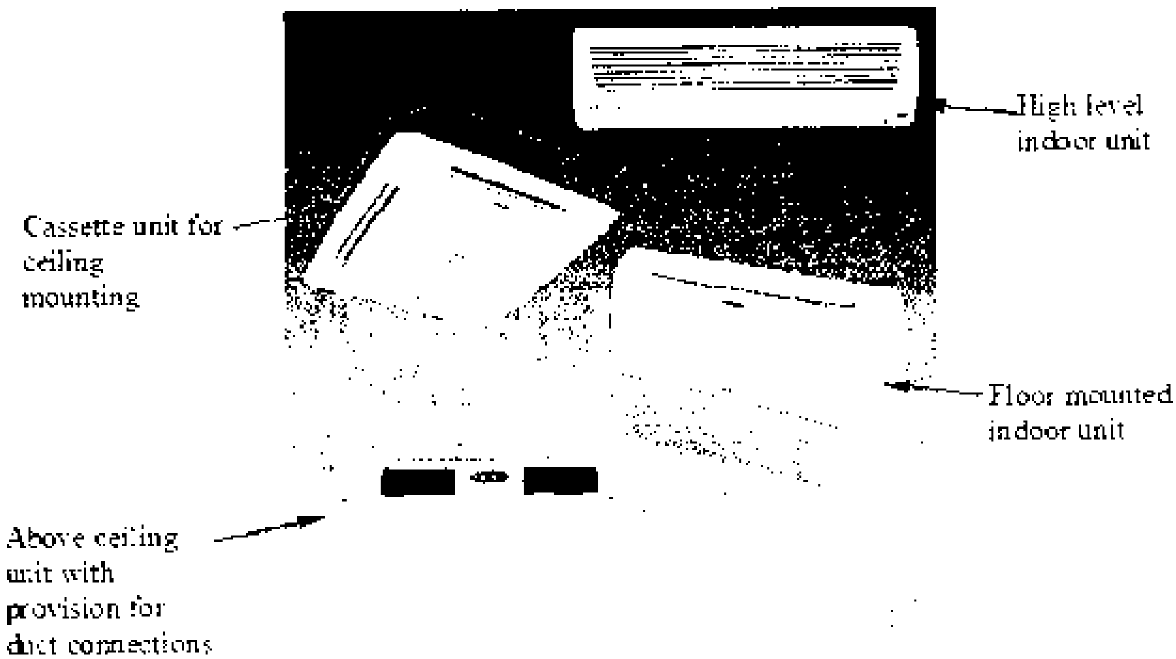
Wall units like the one shown below are built into an external wall and contain all the necessary items of equipment to provide cool air in summer and some may even provide heating in winter.



A small

WALL AIR CONDITIONING UNIT

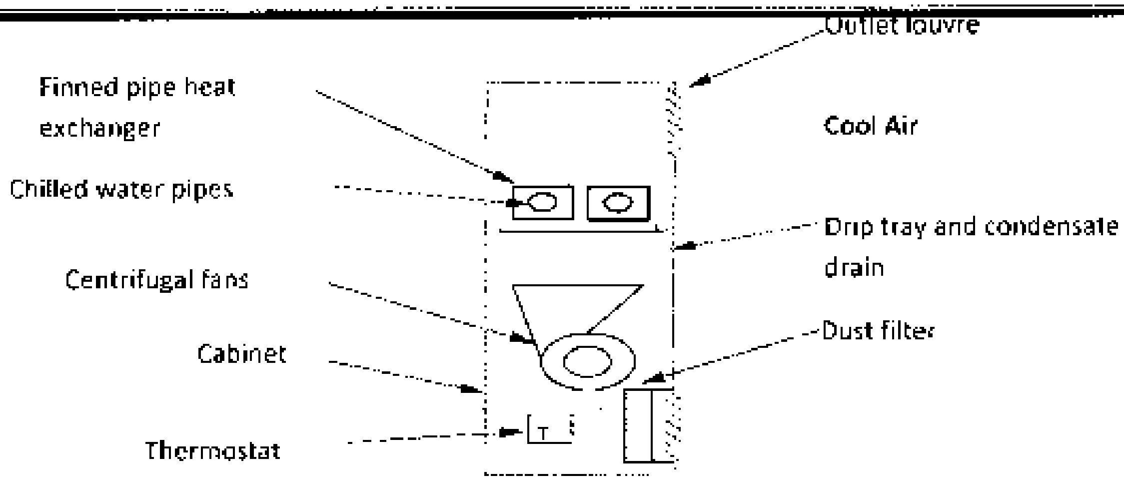
hermetically sealed compressor is used to provide refrigerant gas at the pressure required to operate the refrigeration cycle. The condenser is used to condense the refrigerant to a liquid which is then reduced in pressure and piped to the cooling coil.



VARIOUS TYPES OF ROOM AIR CONDITIONERS - INDOOR UNITS

3.0 Fan Coil Units

These are room air conditioners but use chilled water instead of refrigerant.
Units can be floor or ceiling mounted.
The chilled water is piped to a fin[illegible] heat exchanger as in a fan convector. A fan blows room air across the heat exchanger and cool air is emitted into the room. as shown below.



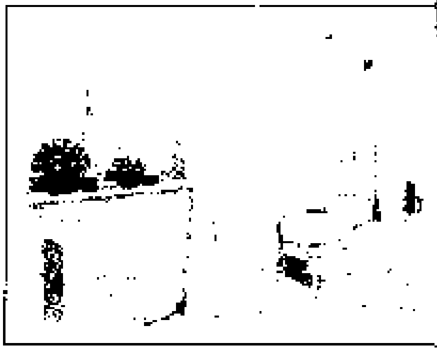
FAN COIL UNIT

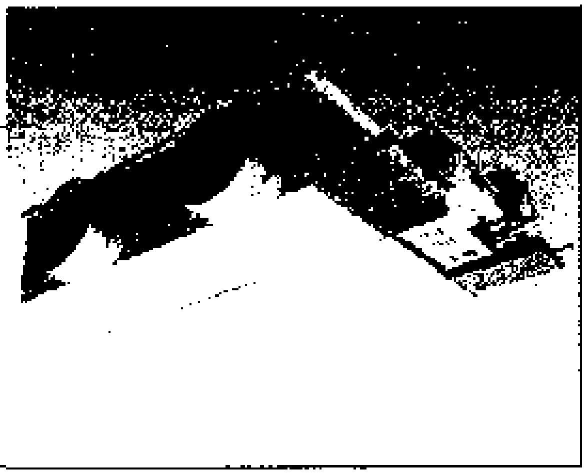
Fan coil units may be looked upon as being small air handling units located in rooms and they can be piped with chilled water for cooling and low temperature hot water (LTHW) for heating if necessary.

The room temperature can be controlled with low, medium and high fan speeds and chilled water flow is varied with a two-port or three-port motorised valve.

Two-pipe, three-pipe and four-pipe systems have been used. The four-pipe system has two heating and two cooling pipes and may have a single heat exchanger or two separate heat exchangers for heating and cooling.

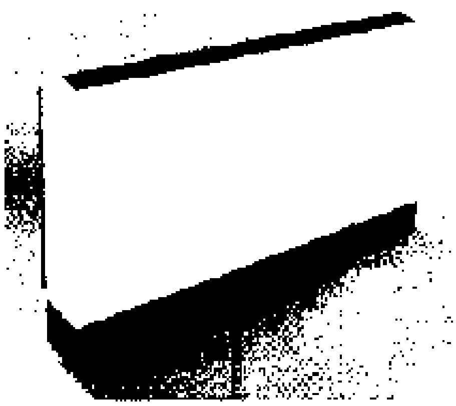
It is useful to have a [illegible] switch in the main control system to avoid both heat exchangers being on at the same time.
A three-pipe system used heating flow, cooling flow and common return pipework.

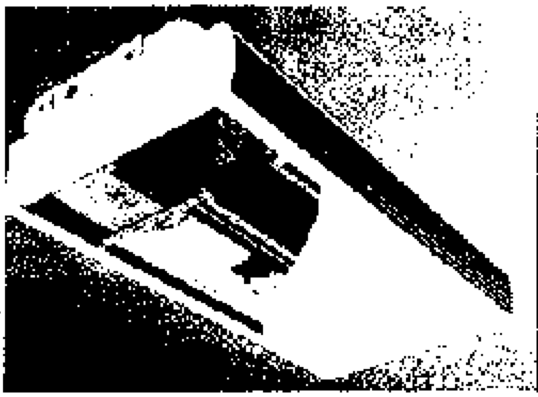




DOMESTIC FAN COIL UNIT

FAN COIL UNIT FOR ABOVE CEILING WITH DUCT CONNECTIONS





FLOOR MOUNTED UNIT

CEILING MOUNTED UNIT

CUT AWAY SHOWING

Psychrometry for Air Conditioning

Psychrometry is the study of air and water vapour mixtures.

Air is made up of five main gases i.e.

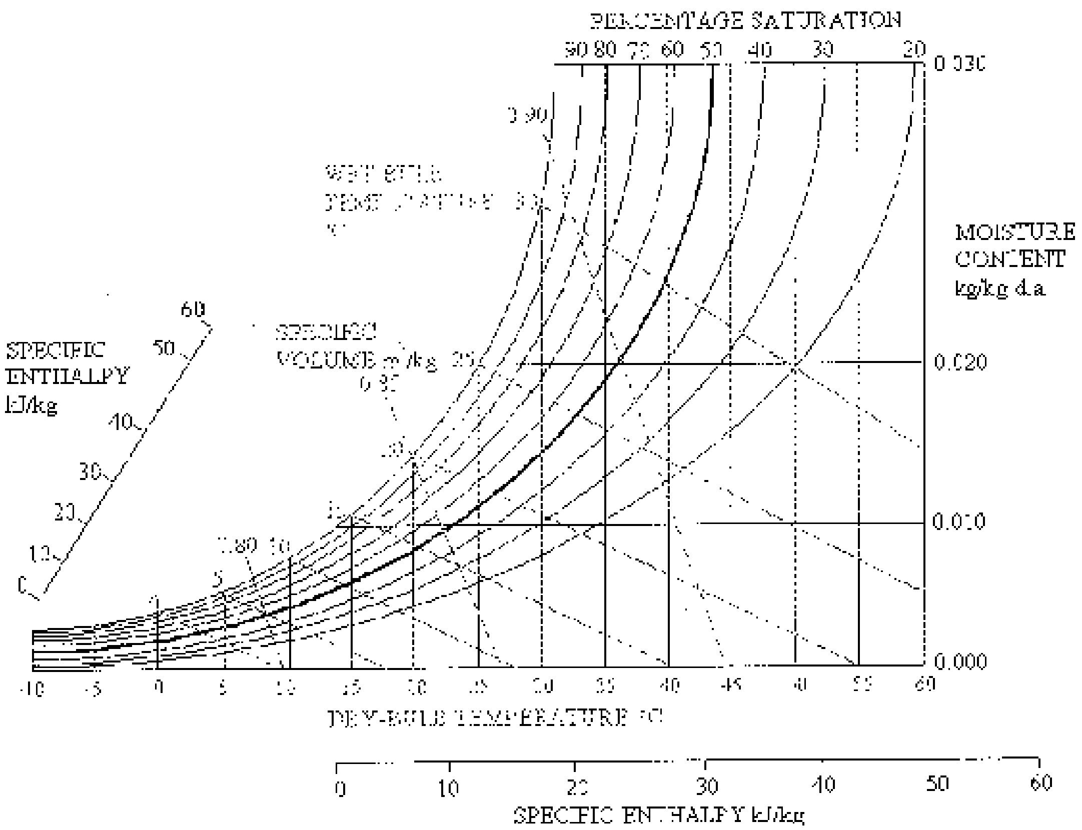
Nitrogen 78.03%, Oxygen 20.99%, Argon 0.94%, Carbon Dioxide 0.03%, and Hydrogen 0.01% by volume.

The Ideal Gas Laws are used to determine psychrometric data for air so that the engineer can carry out calculations.

To make life easier a chart has been compiled with all the relevant psychrometric data indicated.

This is called the Psychrometric Chart.

A typical chart is shown below.



Air at any state point can be plotted on the psychrometric chart.

The information that can be obtained from a Psychrometric Chart is as follows:

1. Dry bulb temperature
2. Wet bulb temperature
3. Moisture content
4. Percentage saturation
5. Specific enthalpy
6. Specific volume.

The following is a brief description of each of the properties of air.

1. Dry bulb temperature

This is the air temperature measured by a mercury-in-glass thermometer.

2. Wet bulb temperature

This is the air temperature measured by a mercury-in-glass thermometer which has the mercury bulb wetted by gauze that is kept moist by a reservoir of water.

When exposed to the environment the moisture evaporates from the wetted gauze, which gives a lower reading on the thermometer.

This gives an indication of how 'dry' or how 'moist' the air is, since in 'dry' air the water will evaporate quickly from the gauze, which depresses the thermometer reading.

3. Moisture content

This is the amount of moisture in air given in kg of moisture per kg of dry air e.g. for room air at 21°C dry bulb and 15°C wet bulb, the moisture content is about 0.008 kg/kg d.a.

This is a small mass of moisture (0.008 kg = 8 grams) per kg of dry air or 9.5

grams per cubic metre of air.

4. Percentage saturation

The Percentage saturation is another indication of the amount of moisture in air.

This is the ratio of the moisture content of moist air to the moisture content of saturated air at the same temperature.

When air is saturated it is at 100% saturation and cannot hold any more moisture.

5. Specific enthalpy

This is the amount of heat energy (kJ) in air per kg.

If heat is added to the air at a heater battery for example, then the amount to be added can be determined from Specific enthalpy change.

6. Specific volume

This is the volume of moist air (dry air + water vapour) per unit mass.

The units of measurement are m^3 per kg.

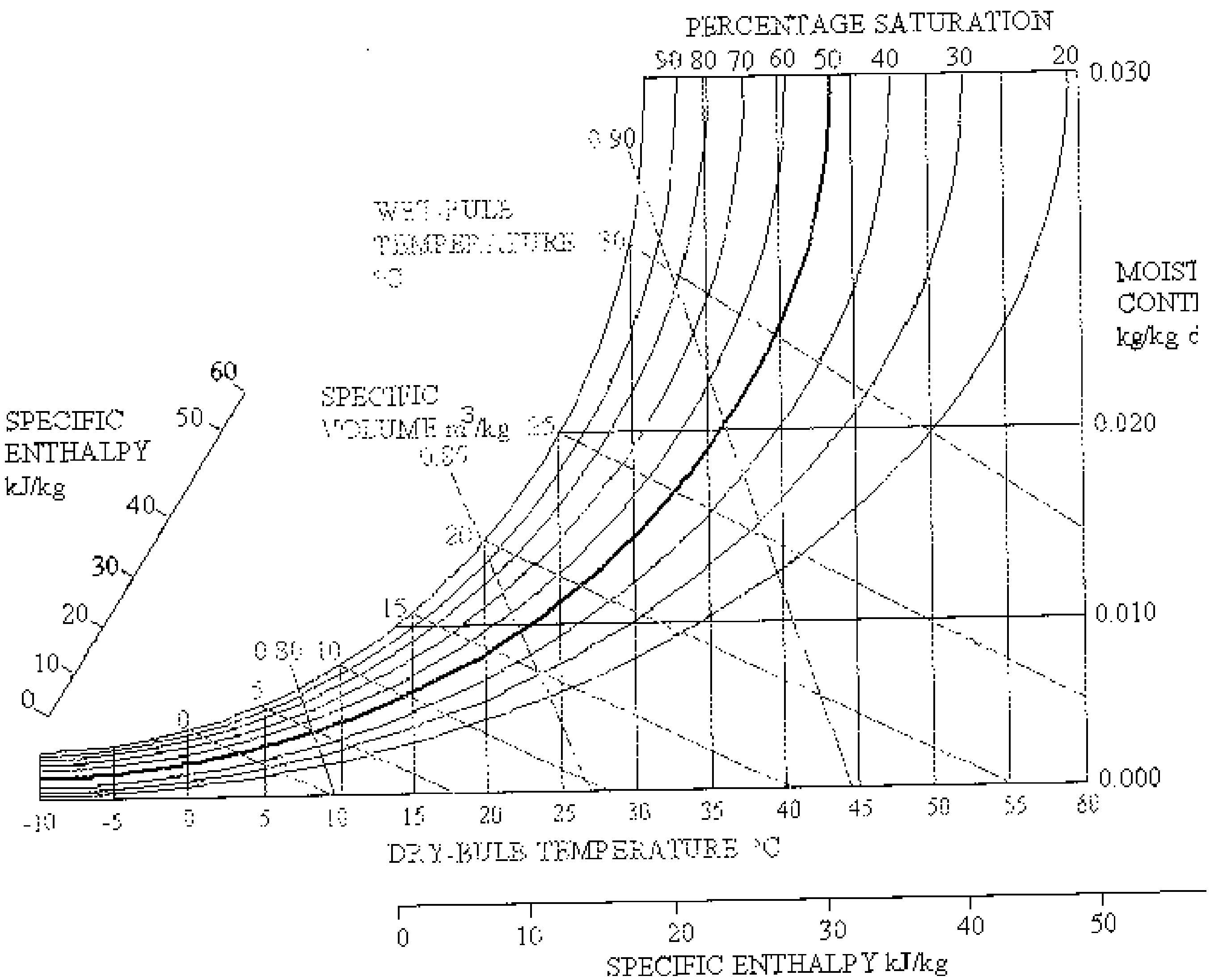
Also specific volume = 1 / density.

THE PSYCHROMETRIC CHART

The six properties of air previously discussed can be shown on one chart called a Psychrometric Chart.

One of the purposes of the Psychrometric Chart is to size heater batteries, cooling coils and humidifiers.

A simplified Psychrometric Chart is shown below.



This chart is only for demonstration purposes.

If accurate assessments are to be carried out use a C.I.B.S.E. chart.

USING THE PSYCHROMETRIC CHART

If any two properties of air are known then the other four can be found from the psychrometric chart.

Examples of Psychrometric Properties

EXAMPLE 1

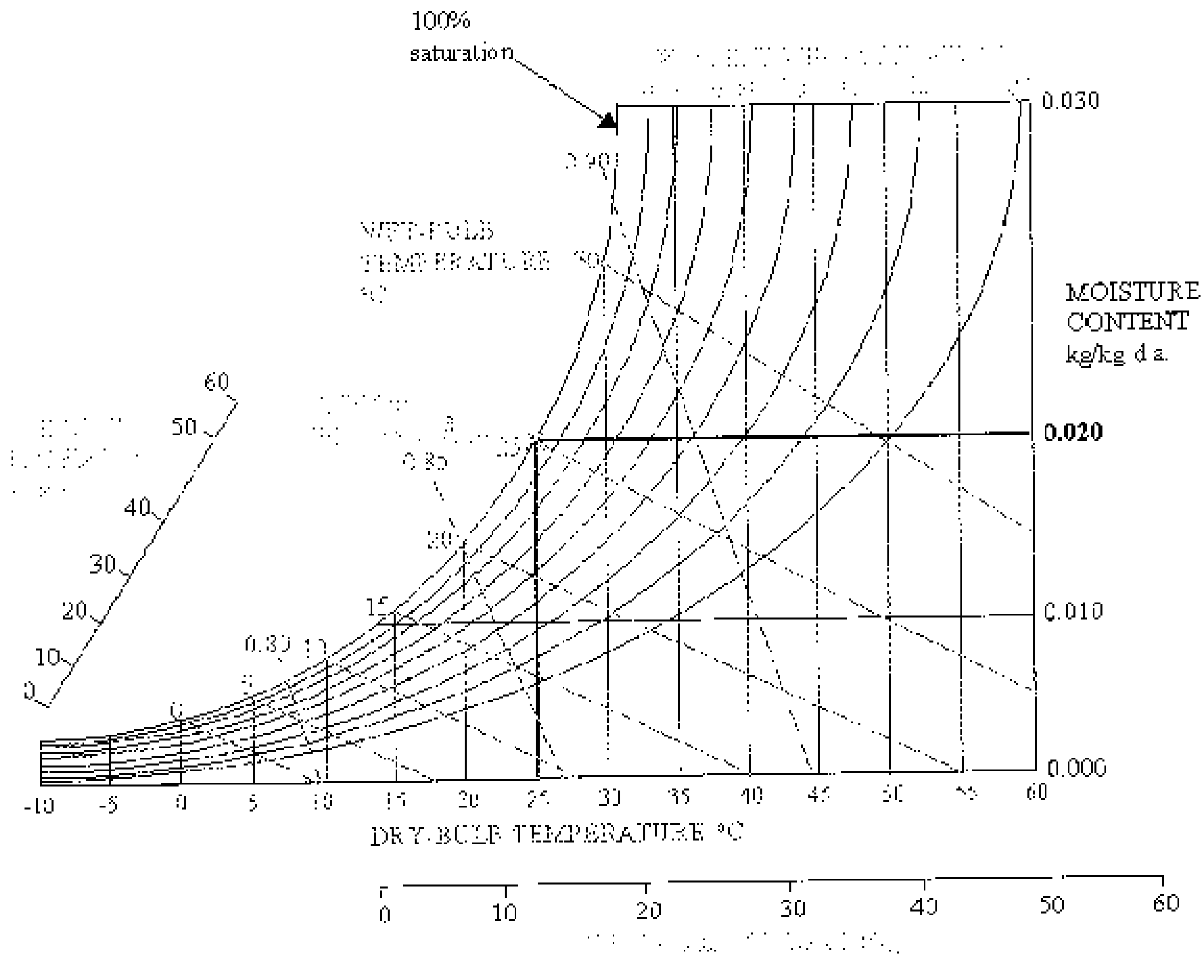
Find the moisture content of air at 25°C dry-bulb temperature and 25°C wet-bulb temperature.

Referring to the chart below, a vertical line is drawn upwards from 25°C dry-bulb temperature until it intersects at 25°C wet-bulb temperature.

This intersection point happens to be on the 100% saturation line.

The intersection point is highlighted and a horizontal line is drawn to the right to find the corresponding moisture content.

The moisture content is therefore 0.020 kg/kg dry air.



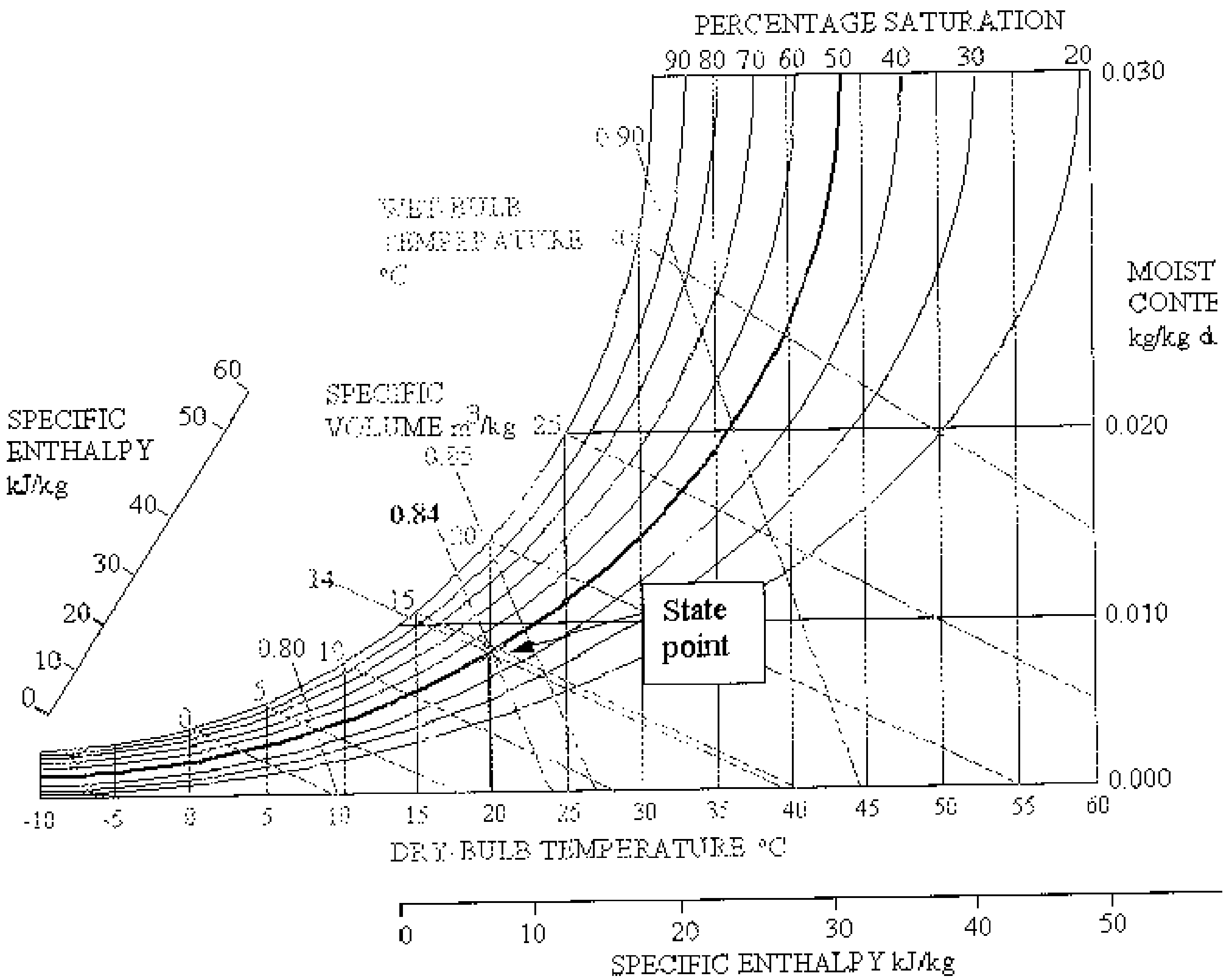
EXAMPLE 2

Find the specific volume and wet-bulb temperature of air at 20°C dry-bulb temperature and 50% saturation.

Referring to the chart below, a vertical line is drawn upwards from 20°C dry-bulb temperature until it intersects with the 50% saturation curve.

The intersection point is sometimes referred to as the state point.

The specific volume is found to be 0.84 m^3/kg and the wet-bulb temperature is 14°C



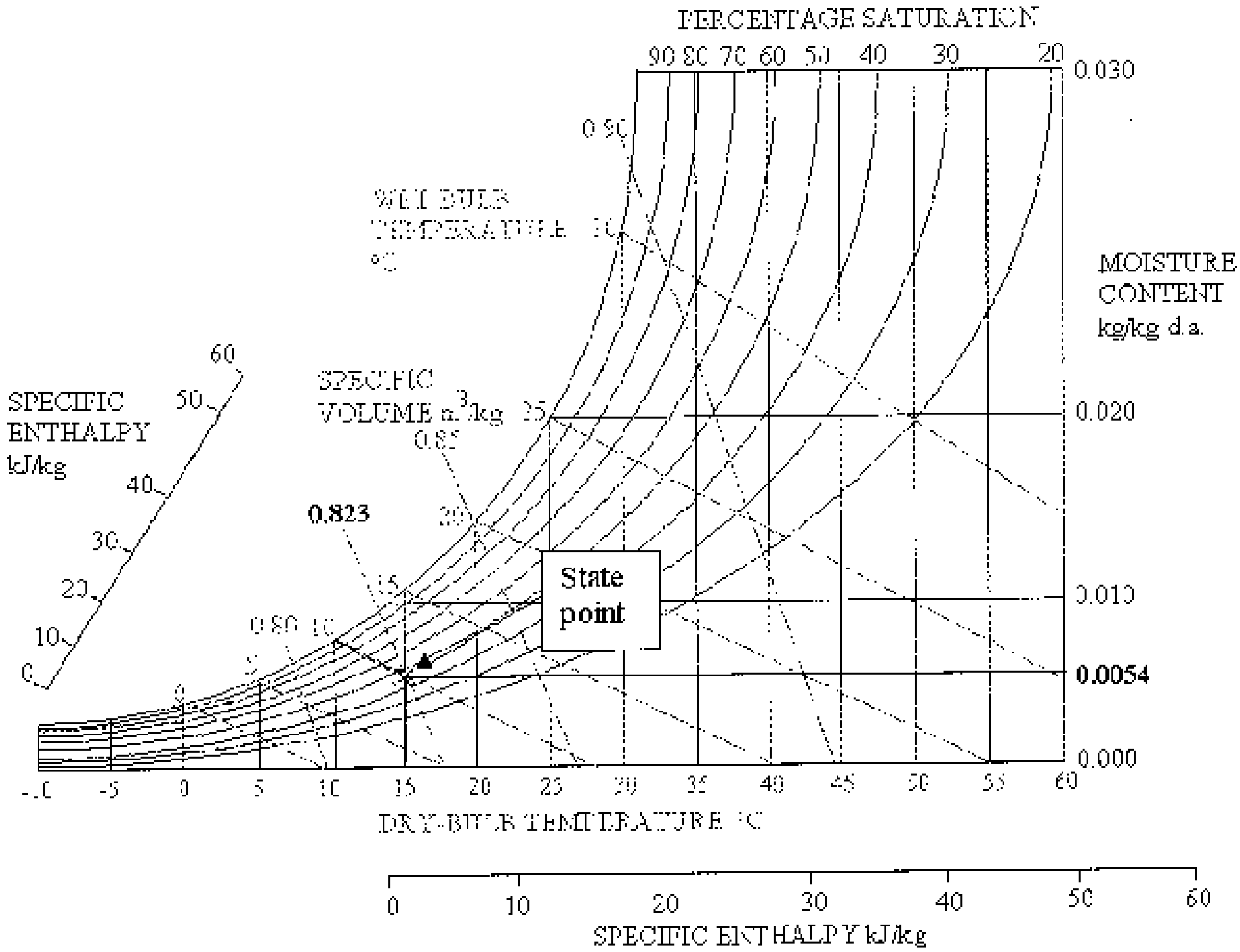
EXAMPLE 3

Find the specific volume, percentage saturation and moisture content of air at 15°C dry-bulb temperature and 10°C wet-bulb temperature.

Referring to the chart below, a vertical line is drawn upwards from 15°C dry-bulb temperature until it intersects with the 10°C wet-bulb temperature line.

This intersection is the state point.

The specific volume is found to be 0.823 m³/kg, the percentage saturation 52% and the moisture content 0.0054 kg/kg d.a.



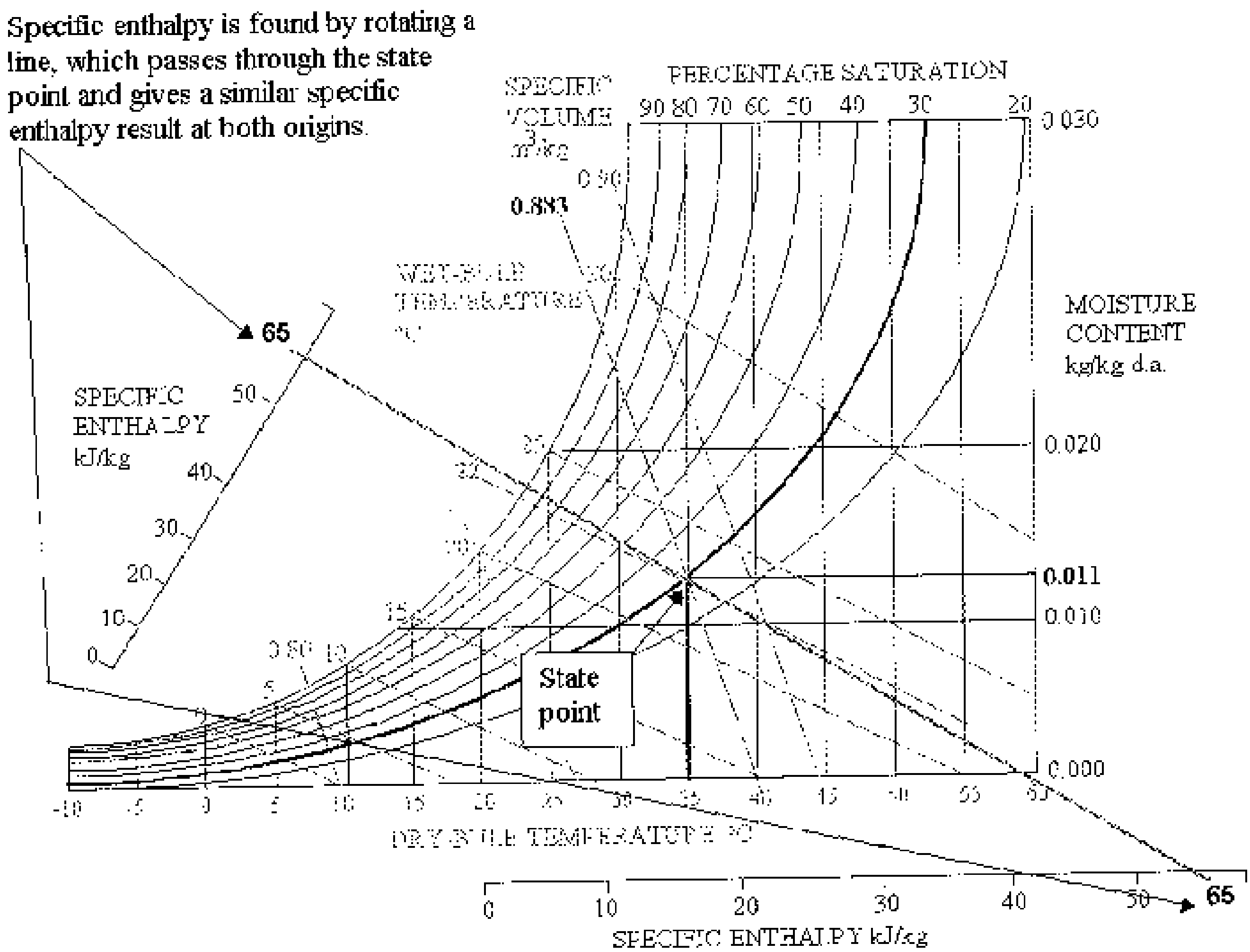
EXAMPLE 4

Find the specific volume, wet-bulb temperature, moisture content and specific enthalpy of air at 35°C dry-bulb temperature and 30% saturation.

Referring to the chart below, a vertical line is drawn upwards from 35°C dry-bulb temperature until it intersects with the 30% saturation curve.

This intersection is the state point.

The specific volume is found to be 0.883 m^3/kg, the wet-bulb temperature is 22°C, the moisture content 0.011kg/kg d.a. and the specific enthalpy 65 kJ/kg.



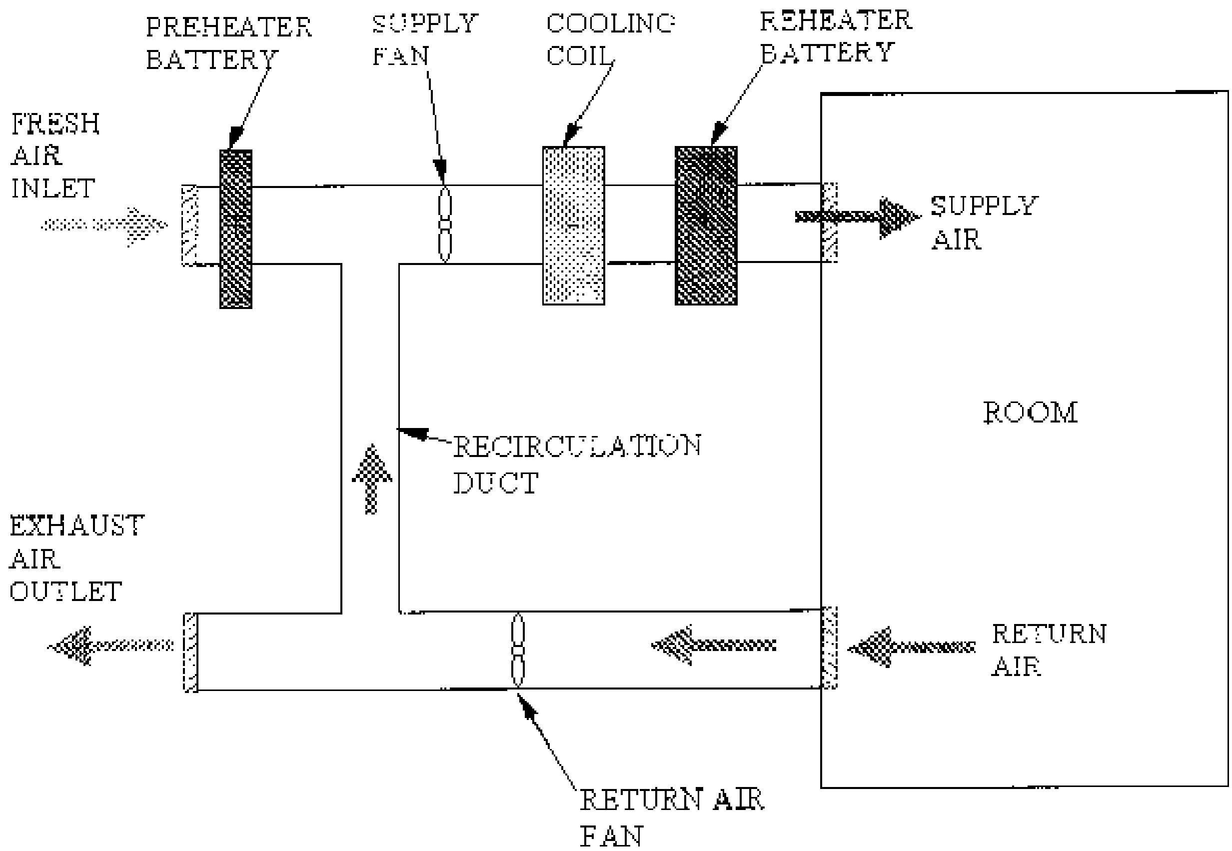
Air Cond. Plant for Summer and Winter

In the summer time when cooling is required by the air conditioning plant it will be necessary to operate the cooling coil, reheater and possibly other plant as well.

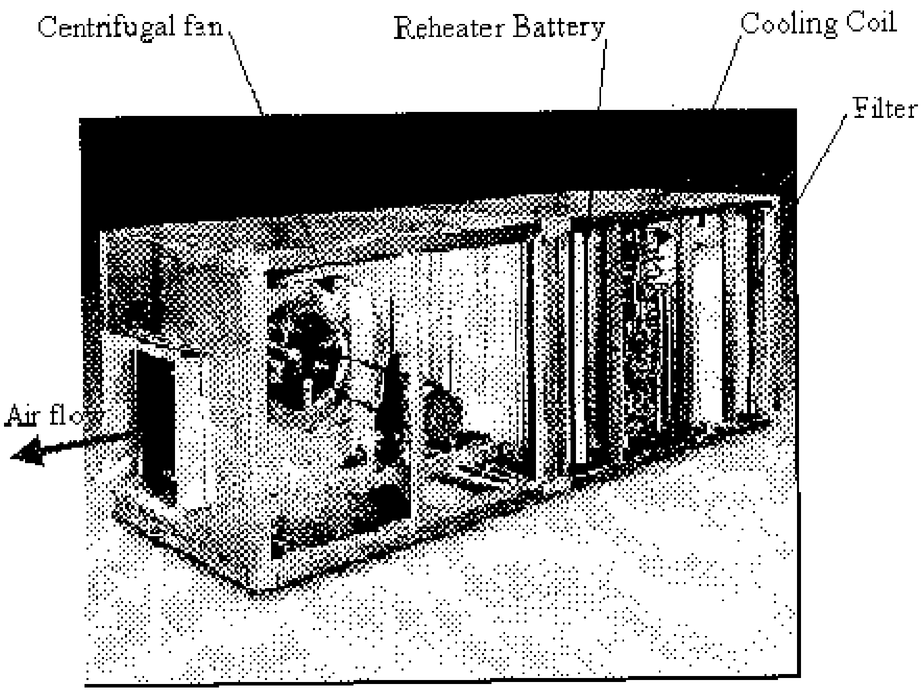
In winter time the preheater and reheater battery will probably be on to provide warm air to overcome heat losses.

Other plant may be switched on as well.

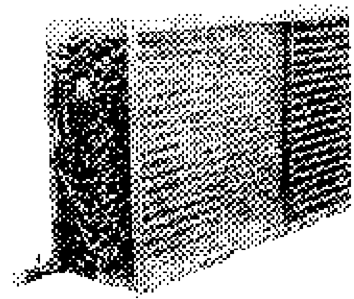
These plant items are shown in the diagram below.



The photographs below show some plant items.



AIR HANDLING UNIT WITH FAN, FILTER, COOLING COIL AND REHEATER.



COOLING COI

Basic Air Conditioning Processes

1. MIXING

Where two air streams are mixed the psychrometric process is shown as a straight line between two air conditions on the psychrometric chart, thus points 1 and 2 are joined and the mix point 3 will lie on this line.

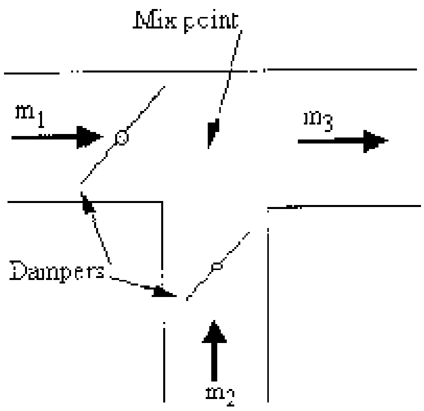
Two air streams are mixed in air conditioning when fresh air (m_1) is brought in from outside and mixed with recirculated air (m_2).

The resulting air mixture is shown below as (m_3).

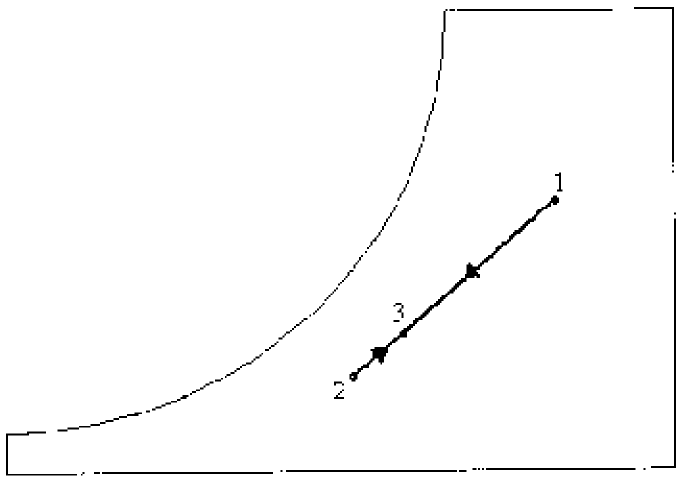
The mixing ratio is fixed by dampers.

Sometimes, in more sophisticated plant, modulating dampers are used which are driven by electric motors to control the mixture of air entering the system.

The diagrams below show mixing of two air streams.



TWO AIR STREAMS MIXING



PSYCHROMETRIC CHART SHOWING TWO AIR STREAMS MIXING

By the conservation of mass formula:

$$m_1 + m_2 = m_3$$

By the conservation of energy formula:

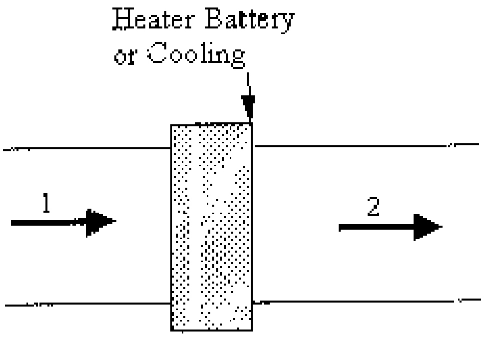
$$m_1 h_1 + m_2 h_2 = m_3 h_3$$

where: m = mass flow rate of air (kg/s)

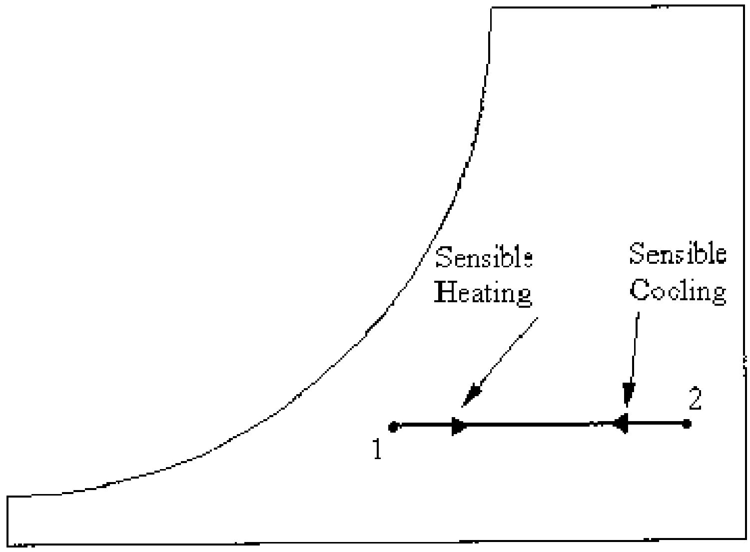
h = specific enthalpy of air (kJ/kg) found from psychrometric chart.

2. SENSIBLE COOLING AND HEATING

When air is heated or cooled sensibly, that is, when no moisture is added or removed, this process is represented by a horizontal line on a psychrometric chart.



AIR STREAM IS HEATED OR COOLED SENSIBLY.



PSYCHROMETRIC CHART SHOWING SENSIBLE HEATING OR COOLING

For sensible heating:

The amount of heating input to the air approximates to;

$$H_{1-2} = m \times Cp \times (t_2 - t_1)$$

Or more accurately from psychrometric chart:

$$H_{1-2} = m \times (h_2 - h_1)$$

For sensible cooling:

The amount of cooling input to the air approximates to;

$$H_{2-1} = m \times Cp \times (t_2 - t_1)$$

Or more accurately from psychrometric chart:

$$H_{2-1} = m \times (h_2 - h_1)$$

where: H = Heat or cooling energy (kW)

m = mass flow rate of air (kg/s)

Cp = Specific heat capacity of air, may be taken as 1.01 kJ/kg degC.

t = Dry bulb temperature of air (^{o}C)

h = specific enthalpy of air (kJ/kg) found from psychrometric chart.

3. COOLING WITH DEHUMIDIFICATION

The most commonly used method of removing water vapour from air (dehumidification) is to cool the air below its dew point.

The dew point of air is when it is fully saturated i.e. at 100% saturation.

When air is fully saturated it cannot hold any more moisture in the form of water vapour.

If the air is cooled to the dew point air and is still further cooled then moisture will drop out of the air in the form of condensate.

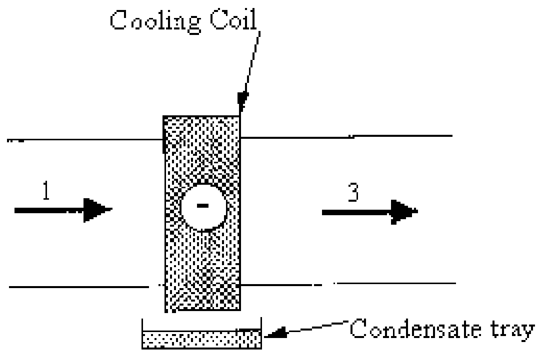
This can be shown on a psychrometric chart as air sensibly cooled until it becomes fully saturated (the dew point is reached) and then the air is cooled latently to a lower temperature.

This is apparent on the psychrometric chart as a horizontal line for sensible cooling to the 100% saturation curve and then the process follows the 100% saturation curve down to another point at a lower temperature.

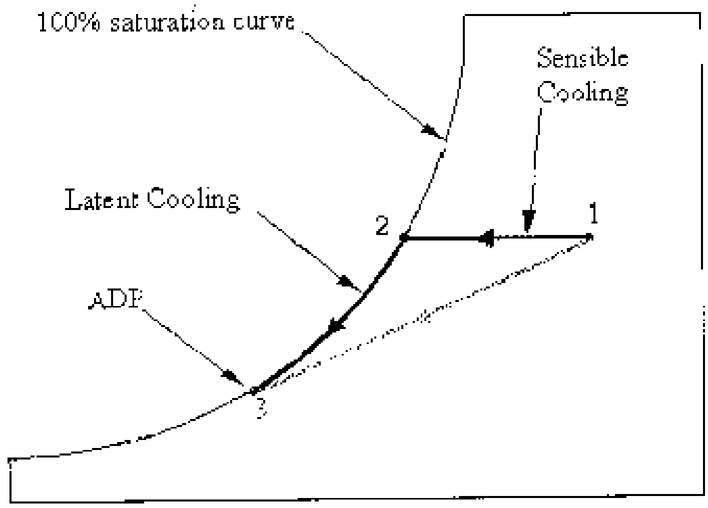
This lower temperature is sometimes called the Apparatus dew Point (ADP) of the cooling coil.

In reality the ADP of the cooling coil is close to the cooling liquid temperature inside the coil. Chilled water or refrigerant may be the cooling liquid.

The psychrometric process from state point 1 to 2 to 3 may be shown as a straight line for simplicity as shown above with a yellow line.



AIR STREAM IS COOLED AND DEHUMIDIFIED.



PSYCHROMETRIC CHART SHOWING COOLING AND DEHUMIDIFICATION

The total amount of cooling input to the air approximates to;

$$H_{1-3} = m \times (h_1 - h_3)$$

The sensible heat removed is:

$$H_{1-2} = m \times (h_1 - h_2)$$

The latent heat removed is:

$$H_{2-3} = m \times (h_2 - h_3)$$

where: H = Cooling energy (kW)

m = mass flow rate of air (kg/s)

h = specific enthalpy of air (kJ/kg) found from psychrometric chart.

In the absence of a suitable psychrometric chart the following formula may be used;

The sensible heat removed is:

$$H_{1-2} = m \times Cp \times (t_1 - t_2)$$

The latent heat removed is:

$H_{2\text{-}3} = m \times hfg \times (g_2 - g_3)$

where: H = Cooling energy (kW)

m = mass flow rate of air (kg/s)

Cp = Specific heat capacity of air, may be taken as 1.01 kJ/kg degC.

t = Dry bulb temperature of air (^{0}C)

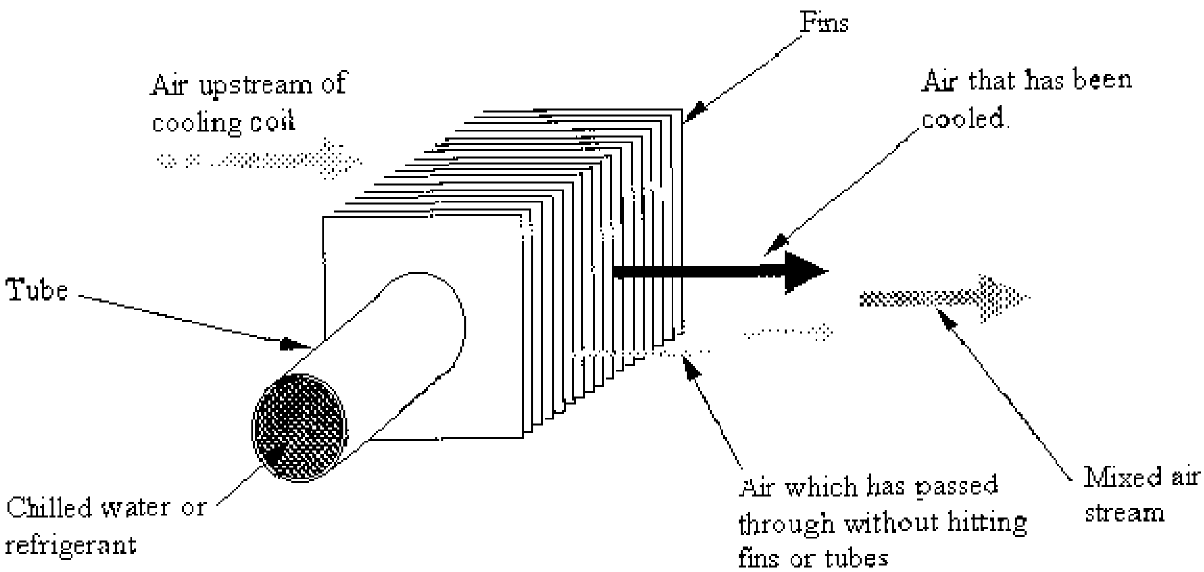
hfg = latent heat of evaporisation, may be taken as 2454 kJ/kg @20^{0}C.

g = moisture content of air from psychrometric chart (kg/kg dry air)

3.1 COOLING COIL CONTACT FACTOR

Some of the air going through a cooling coil does not come into contact with the tubes or fins of the cooling coil and is therefore not cooled to the ADP temperature.

A mixing process therefore takes place as two air streams mix downstream of the cooling coil as shown below.

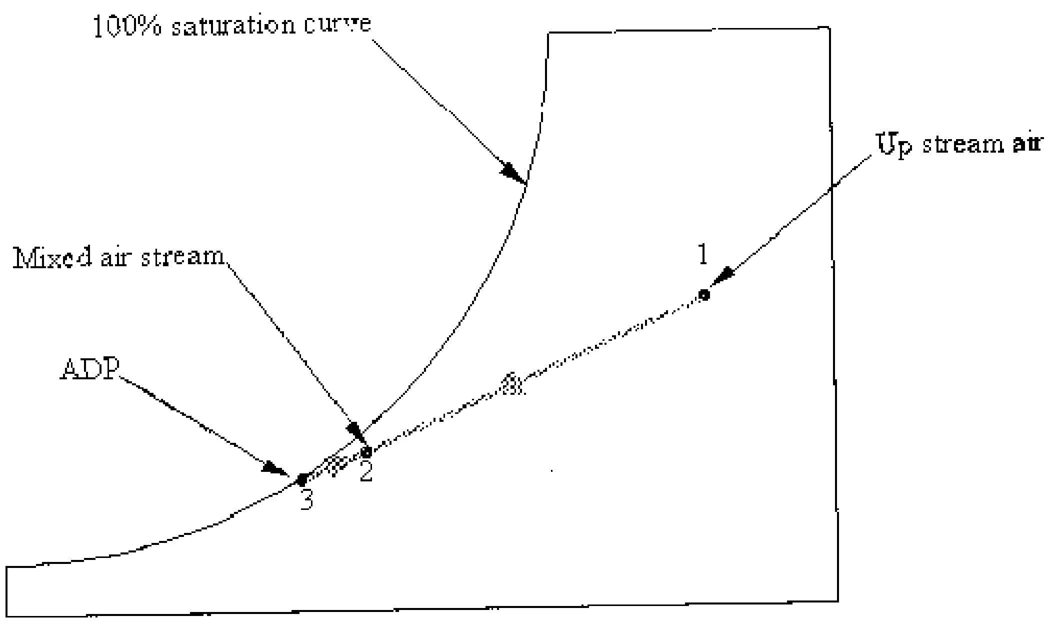


A SECTION OF COOLING COIL SHOWING AIR STREAMS

One air stream is cooled down to the ADP and the other air stream by-passes the coil surfaces to give an off-coil air temperature (mixed air stream) a little higher than the ADP.

This may be looked upon as an inefficiency of the coil and is usually given as the cooling coil contact factor.

The process is shown on the psychrometric chart below.



PSYCHROMETRIC CHART SHOWING COOLING COIL CONTACT FACTOR

The contact factor of a cooling coil may be found from;

$$\text{Contact Factor} = \frac{(h_1 - h_2)}{(h_1 - h_3)}$$

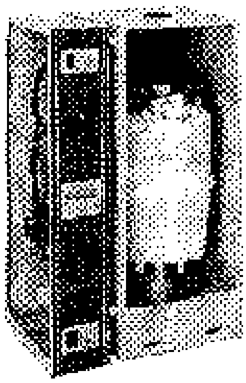
Another expression for contact factor is;

4. HUMIDIFICATION

If is it necessary to add some moisture to the supply air then this is best done by injecting steam into the air stream.

Humidification can be carried out by spraying a fine mist of water droplets into the air but this is not recommended in rooms occupied by people due to the risk of bacteria carry over.

Dry steam may be injected from a steam supply pipe or generated in a local packaged unit as shown in the photograph below. A disadvantage of using an existing steam supply is smells may be carried over into the air.

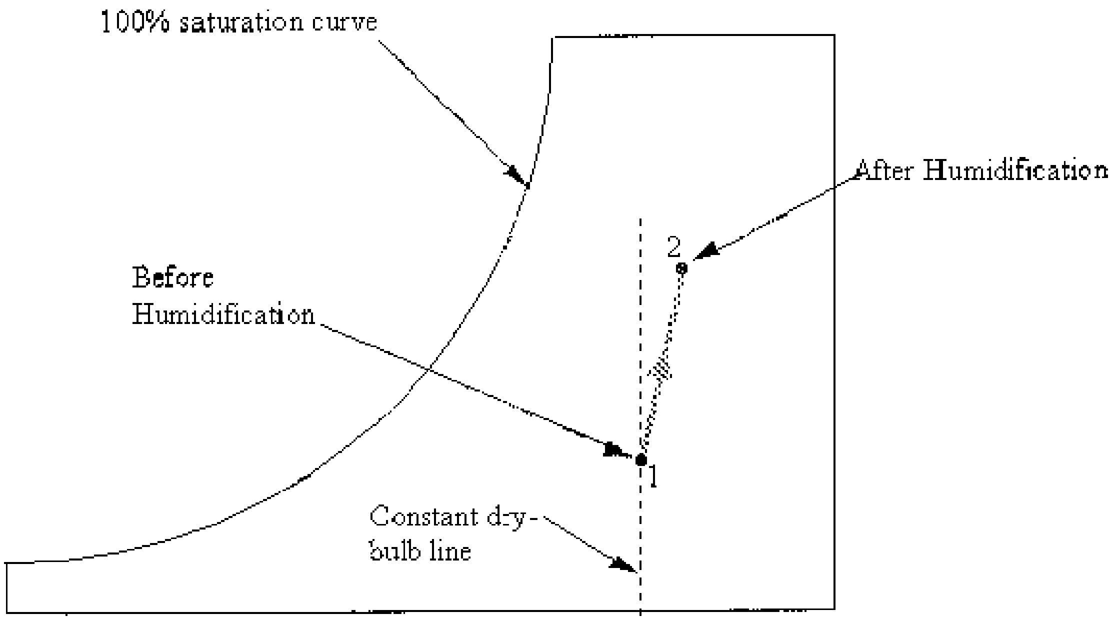


PACKAGED STEAM HUMIDIFIER

The steam package unit is situated close to the air duct and is sized to meet the maximum requirements; this is usually in winter in the U.K.

A steam pipe (sometimes hoses are used) passes from the packaged unit to the air duct and steam at 100°C is injected into the air stream via. a sparge pipe. The un-used steam is drained from the system via a condensate tundish and drain. It is important to layout the steam pipework so that any condensate will drain back to the unit.

The psychrometric process is shown below.



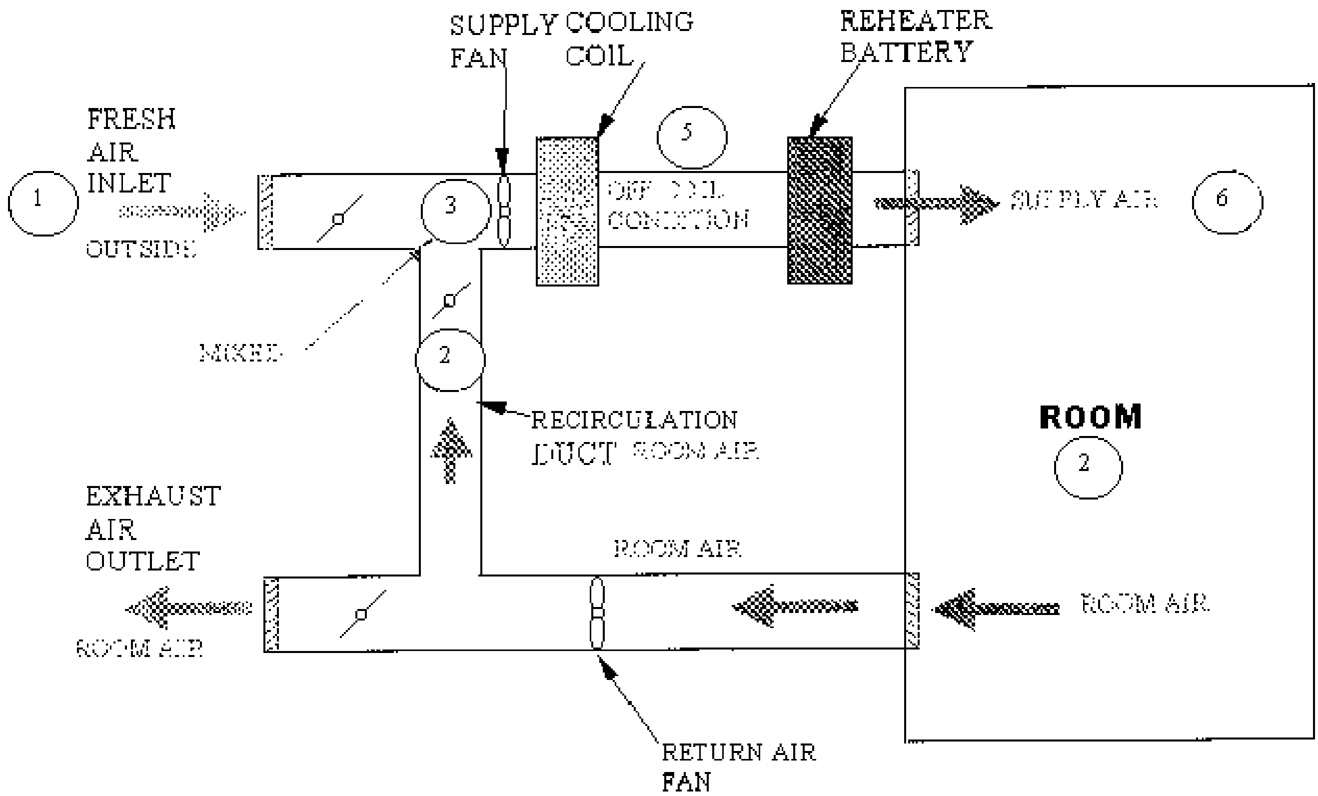
PSYCHROMETRIC CHART SHOWING HUMIDIFICATION

See Summer and Winter Cycles section for calculation of amount of moisture added at humidifier.

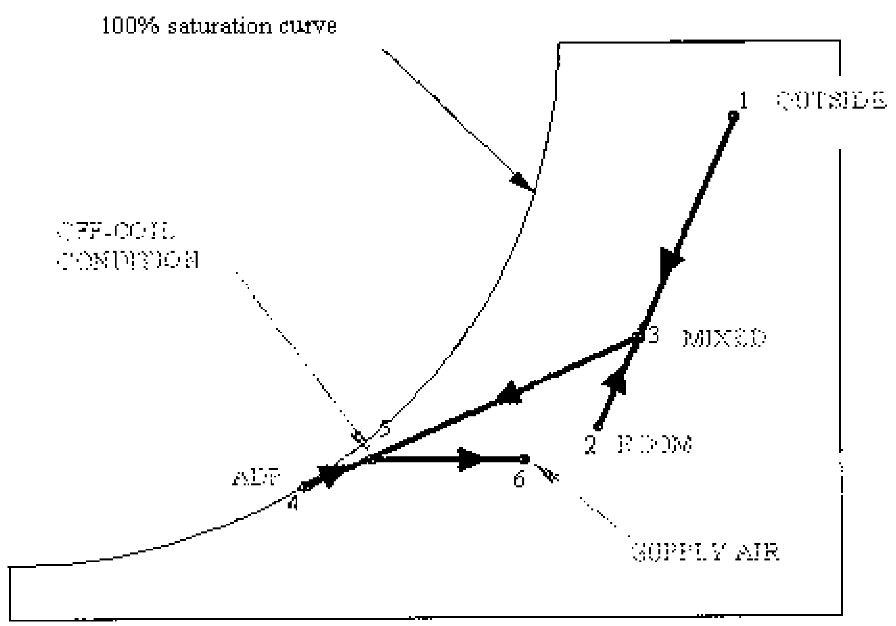
$$\text{Contact Factor} = \frac{\text{Distance 1 to 2 (mm)}}{\text{Distance 1 to 3 (mm)}}$$

TYPICAL AIR CONDITIONING PROCESSES

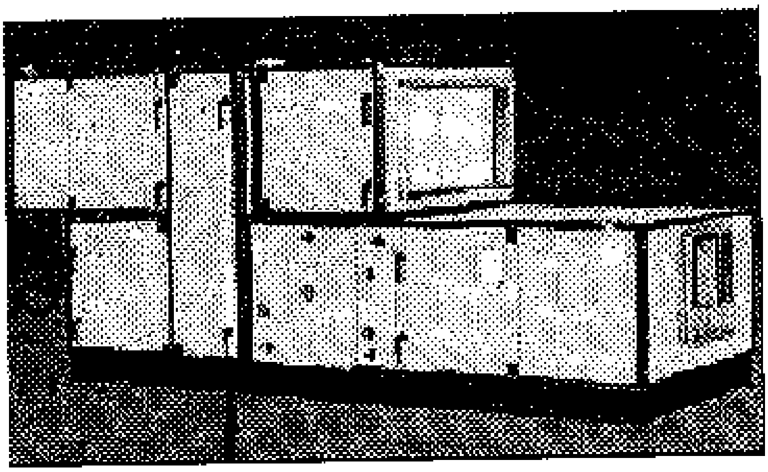
The schematic diagram below shows a typical plant system for summer air conditioning.



The psychrometric diagram below shows a typical summer cycle.

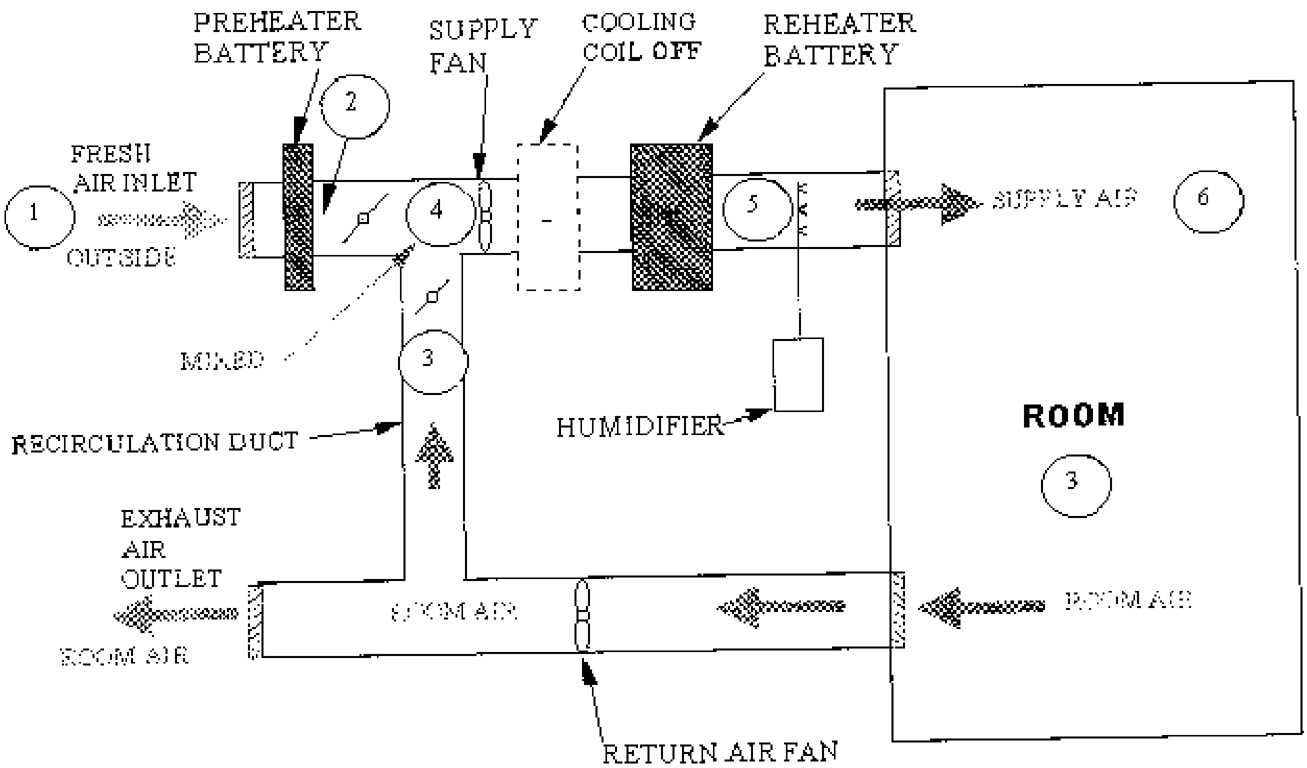


PSYCHROMETRIC CHART SHOWING TYPICAL SUMMER CYCLE

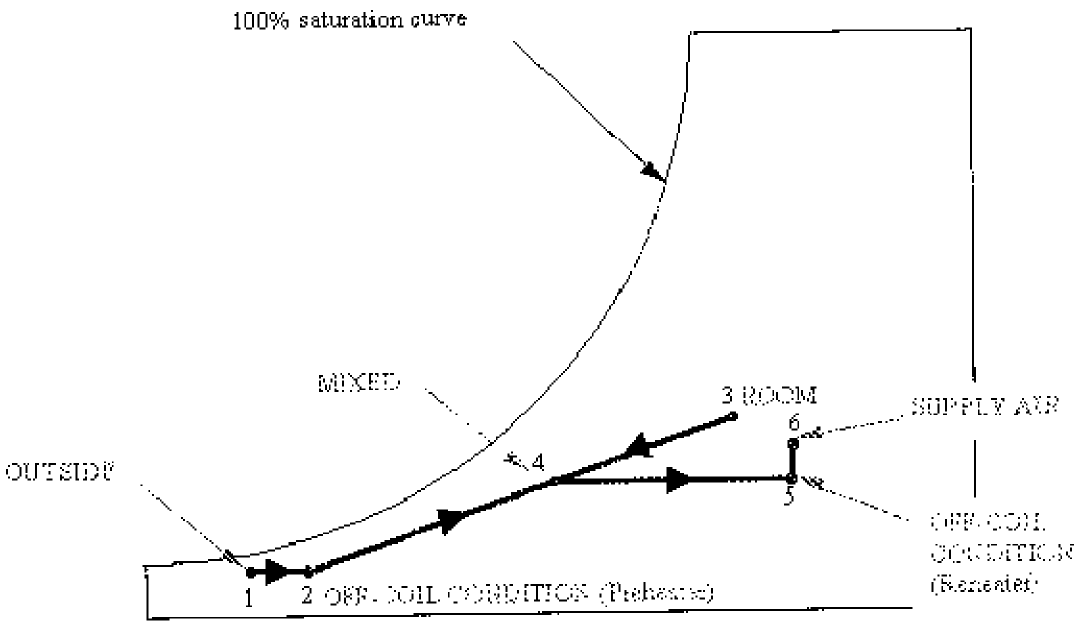


AIR HANDLING UNIT

The schematic diagram below shows a typical plant system for winter air conditioning.



The psychrometric diagram below shows a typical winter cycle.



PSYCHROMETRIC CHART SHOWING TYPICAL WINTER CYCLE

ANNOTATION

The state points on a psychrometric chart may be given numbers or symbols to identify them. If symbols are used the following system may be adopted:

Air State point	Letter
Outside	**O**
Room	**R**
Mixed	**M**
Apparatus Dew Point	**ADP**
Off cooling coil condition	**W**
Room Ratio Line	**RRL**
Preheater off coil condition	**P**
Upstream of Humidifier	**H**
Supply	**S**
Duct, fan gain allowance	**D**

ROOM RATIO

This is the ratio of sensible to total heat in the room for summer or winter.

The total heat gain (summer) or loss (winter) will be determined by adding the Latent and Sensible heat in a room or rooms, i.e.

(SUMMER) Total heat gain = Sensible heat gain + Latent heat gain

(WINTER) Total heat = Sensible heat loss + Latent heat gain

The room ratio is used on a psychrometric chart to determine the supply air state point.

A room ratio line is superimposed from the protractor on the psychrometric chart onto the main body of the chart by a line passing through the room state point R.

An example calculation is as follows:

Sensible heat gain = 9.0 kW

Latent heat gain = 2.25 kW

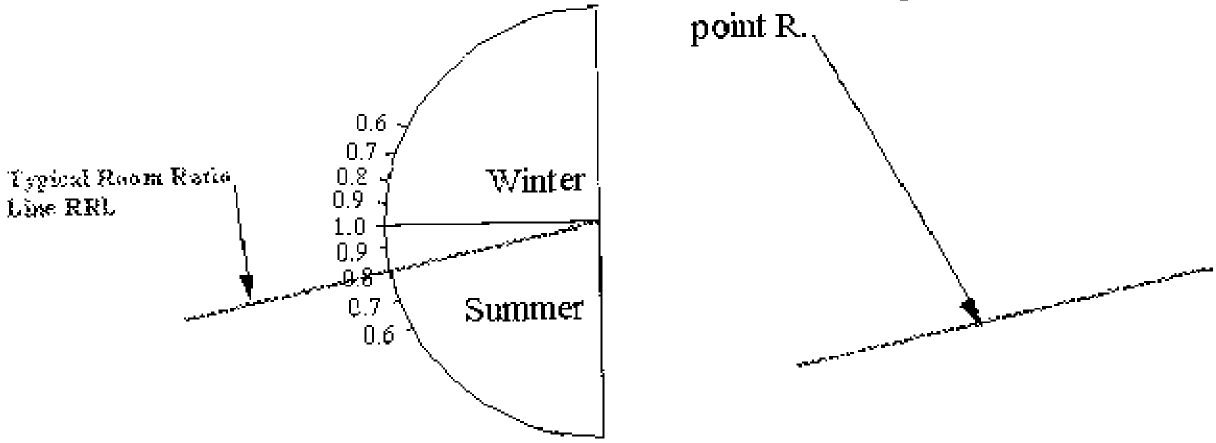
Total heat gain = 9.0 kW + 2.25 kW = 11.25 kW.

Room ratio = Sensible / Total heat

Room ratio = 9 / 11.25 = 0.8

The supply air state point must also be somewhere on this room ratio line to meet the room heat gain requirements i.e. the room ratio line always passes through points R and S.

The RRL at this slope or gradient is then superimposed onto the main body of a psychrometric chart with the aid of set squares. The line must pass thought state point R.



HEAT RATIO PROTRACTOR ON PSYCHROMETRIC CHART

Summer and Winter Cycles

1. SUMMER CYCLE PSYCHROMETRICS

1. Draw schematic diagram of air-conditioning plant.

2. Plot room condition R on psychrometric chart.

3. Plot outside condition O on psychrometric chart.

4. Join points O and R.

5. Find the mix point M by measuring the length of the line O-R and multiply this by the mixing ratio.
   If there is more recirculated air than outside air at the mix point, then point M will be closer to point R than point O.

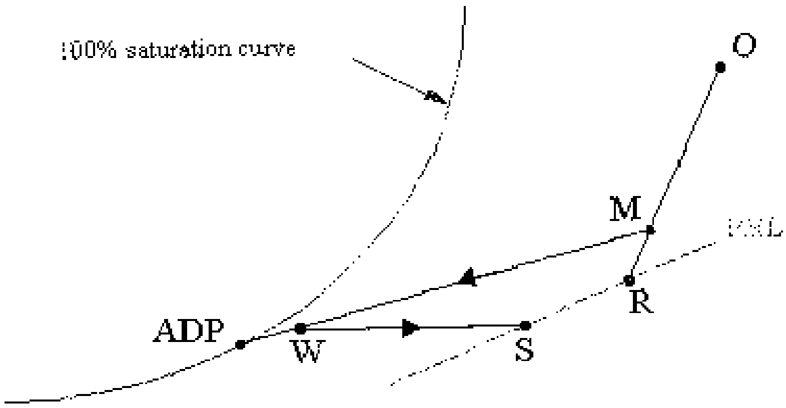
6. Find the room ratio.
   This is the sensible to total heat gain ratio.
   Plot this ratio on the protractor, bottom segment, on the psychrometric chart and transfer this line onto the chart so that it passes through point R.

7. Plot the Apparatus Dew Point ADP of the cooling coil.
   This is on the 100% saturation curve.
   The wet bulb and dry bulb temperatures at this point will be equal.

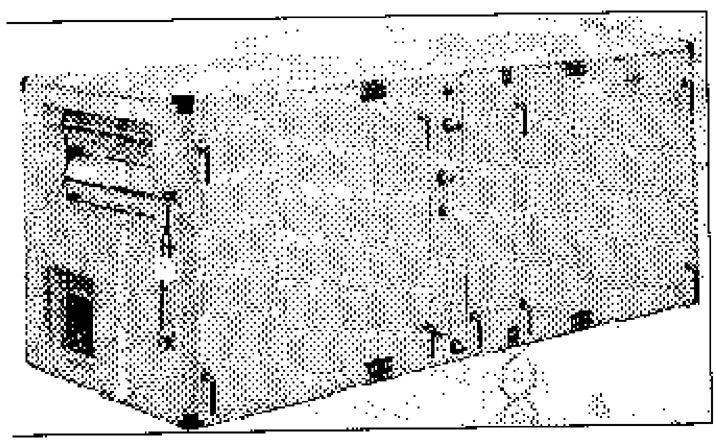
8. Join points M and ADP.

9. Find the off-coil condition W by measuring the length of the line M-ADP and multiply this by the cooling coil contact factor.
   Measure down from point M to the point W.
   The closer the contact factor is to unity, the closer point W will be to point ADP.

10. Plot the supply air condition S.
    The reheater process will be a horizontal line from point W to point S.
    Point S is on the room ratio line.



SUMMER CYCLE



TYPICAL AIR HANDLING UNIT

2. SUMMER CYCLE CALCULATIONS

2.1 MASS FLOW RATE

When the supply air temperature has been found from the psychrometric chart the the mass flow rate of air can be calculated from the following formula:

$$H_s = m_a \times Cp \quad (t_r - t_s)$$

where.

H_s	=	Sensible heat gain to room (kW)
m_a	=	mass flow rate of air (kg/s)
Cp	=	Specific heat capacity of humid air (approx.1.01 kJ/kg degC)
t_r	=	room temperature (°C)
t_s	=	supply air temperature (°C)

2.2 COOLING COIL OUTPUT

The cooling coil output is as follows:

$$H_{cooling\ coil} = m_a \quad (h_M - h_{ADP})$$

where:

$H_{cooling\ coil}$	=	Cooling coil output (kW)
m_a	=	mass flow rate of air (kg/s)
h_M	=	specific enthalpy at condition M (kJ/kg)
h_{ADP}	=	specific enthalpy at condition ADP (kJ/kg).

2.3 HEATER BATTERY OUTPUT

The heater battery or reheater output is as follows:

$$H_{heater\ battery} = m_a \quad (h_S - h_W)$$

where:

$H_{heater\ battery}$	=	Heater battery output (kW)
m_a	=	mass flow rate of air (kg/s)
h_S	=	specific enthalpy at condition M (kJ/kg)
h_W	=	specific enthalpy at condition W (kJ/kg)

Summer and Winter Cycles

3. WINTER CYCLE PSYCHROMETRICS

1. Draw schematic diagram of air-conditioning plant.

2. Plot room condition R on psychrometric chart.

3. Plot outside condition O on psychrometric chart.

4. Plot the after Preheater condition P if there is one in the system. The Preheater process will be a horizontal line from O to P and will be only a few degrees dry bulb if it acts as a frost coil.

5. Join points P and R. If there is no frost coil read O-R for P-R.

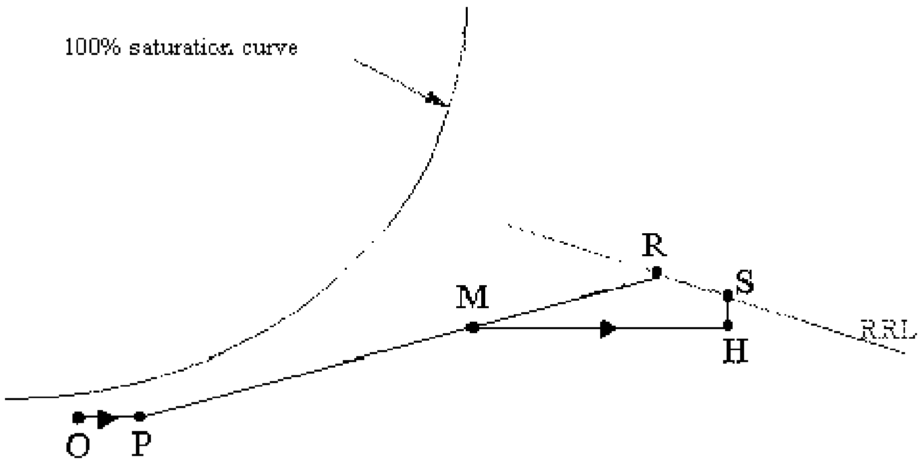
6. Find the mix point M by measuring the length of the line P-R and multiply this by the mixing ratio. If there is more recirculated air than outside air at the mix point, then poin M will be closer to point R than point P.

7. Find the room ratio. This is the sensible to total heat ratio. Neglect signs ie. the total heat for the room will be Sensible loss plus Later gain. Plot this ratio on the protractor, top segment, on the psychrometric chart an transfer this line onto the chart so that it passes through point R.

8. Find the supply air dry bulb temperature by calculation. The mass flow rate of air is the same as that for Summer for a Constar Volume system.

9. Plot the supply air condition S on the room ratio line.

10. Plot condition H on the psychrometric chart. This is vertically down from point S, and horizontally across from point M. This is because M-H is the reheater process and thus a horizontal line an H-S is the humidification process and is close to a vertical line if steam i used.



WINTER CYCLE

4. WINTER CYCLE CALCULATIONS

4.1 SUPPLY AIR DRY BULB TEMPERATURE

When the mass flow rate of air is calculated for the summer condition then the winter supply air dry bulb temperature can be calculated from the following formula

$$H_s = m_a \times Cp \ (t_s - t_r)$$

where:

H_s = Sensible heat gain to room (kW)
m_a = mass flow rate of air (kg/s)
Cp = Specific heat capacity of humid air (approx 1.01 kJ/kg degC)
t_r = room temperature (°C)
t_s = supply air temperature (°C)

4.2 PREHEATER BATTERY OUTPUT (or frost coil)

The preheater battery output is as follows:

$$H_{preheater\ battery} = m_{af} \ (h_P - h_O)$$

where:

$H_{preheater\ battery}$ = Preheater battery output (kW)
m_{af} = mass flow rate of fresh air (kg/s)
h_P = specific enthalpy at condition P (kJ/kg)
h_O = specific enthalpy at condition O (kJ/kg)

4.3 REHEATER BATTERY OUTPUT

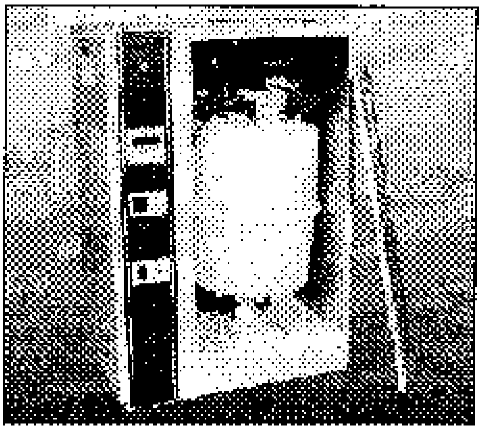
The reheater battery output is as follows:

$$H_{reheater\ battery} = m_a \ (h_H - h_M)$$

where:

$H_{reheater\ battery}$ = Reheater battery output (kW)
m_a = mass flow rate of supply air (kg/s)
h_H = specific enthalpy at condition H (kJ/kg)
h_M = specific enthalpy at condition M (kJ/kg)

4.4 HUMIDIFIER OUTPUT



The amount of moisture added to the air may be calculated from the following formula:

$$m_{moisture\ added} = m_a \ (m_{sS} - m_{sH})$$

where:

$m_{moisture\ added}$	=	The amount of moisture or added or steam flow rate (kg/s)
m_a	=	mass flow rate of air (kg/s)
m_{sS}	=	moisture content at condition S (kg/kg d.a.)
m_{sH}	=	moisture content at condition H (kg/kg d.a.)

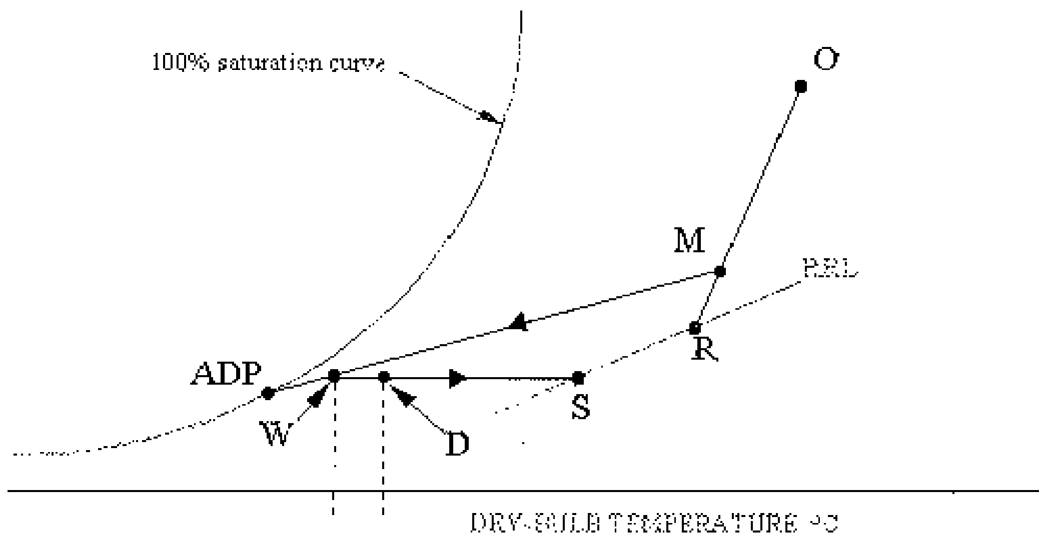
Summer and Winter Cycles

5. DUCT AND FAN GAINS

The air in a duct is slightly heated from the fan electric motor and heat is also transmitted through the duct wall from warm areas Into the air stream, for example;
a duct contains air at 15°C and passes through a roof space at 30°C in summer.
There will be heat transferred through the duct wall, which increases the air temperature slightly.

To allow for this in the summer psychrometric process an additional sensible heating state point D is added as shown below.

The air may be heated by several °C depending on the fan motor, length of duct and type of duct insulation used, if any. The distance from point W to point D may be typically 1°C to 3°C dry bulb temperature in the U.K.



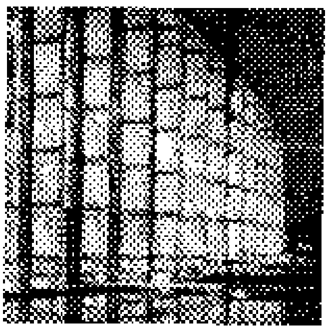
SUMMER CYCLE

Heat Gain

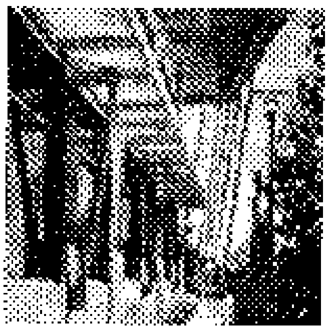
Introduction

Heat gains from the sun can lead to increases in internal temperatures beyond the limits of comfort.
This is usually about 24°C.
It is therefore necessary to determine the amount of solar radiation that is transmitted into buildings through; windows, walls, roof, floor and by admitting external air into the building.

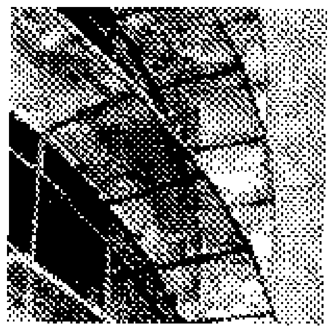
Several measures can be adopted to reduce solar radiation in buildings.
These are external and internal shading and by careful building design.



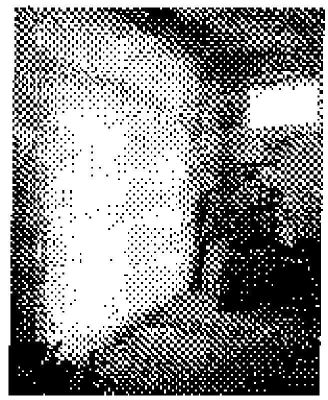
Vertical fin shading



Horizontal shading



Horizontal shading



Internal vertical blinds

Calculating Heat Gains

The load on an air-conditioning system can be divided into the following sections:

1. Sensible Transmission through glass.
2. Solar Gain through glass.
3. Internal Heat gains
4. Heat gain through walls.
5. Heat gain through roof.
6. Ventilation and/ or infiltration gains.

The heat gain through the glass windows is divided into two parts since there is a heat gain due to temperature difference between outside and inside and another gain due to solar radiation shining through windows.

The method adopted uses the priveous tabels.
It would be helpful to have these Tables close by, to complete the calculations.
An example of a heat gain claculation is given .
Heat gains through solid ground floors are minimal and can be neglected.

1.0 Sensible Transmission Through Glass

This is the Solar Gain due to differences between inside and outside temperatures.
In very warm countries this can be quite significant.
This gain only applies to materials of negligible thermal capacity i.e. glass.

$$Qg = Ag \cdot Ug\,(t_o - t_r) \qquad \text{........ eqn. 1}$$

Where:

Q_g	=	Sensible heat gain through glass (W)
A_g	=	Surface area of glass (m^2)
U_g	=	'U' value for glass ($W/m^2\ ^oC$)
t_o	=	outside air temperature (^oC).
t_r	=	room air temperature (^oC)

2.0 Solar Gain Through Windows

This gain is when the sun shines though windows.
The cooling loads per metre squared window area have been tabulated for various times, orientations, latitutes and building weights.

Heat load is found from;

$$Q_{sg} = SHG \cdot SC \cdot CLF \cdot A_g \qquad \text{........ eqn. 2}$$

where

Q_{sg}	=	Actual cooling load (W)
SHG	=	Soler heat gaincoefficient
SC	=	Shading cefficients

CLF - Cooling load factor
A_g = Area of glass (m^2)

3.0 Internal Heat Gains -

Internal gains can account for most heat gain in buildings in the iraq.
These gains are from occupants, lights, equipment and machinery, as detailed below.
OCCUPANTS - Sensible and latent heat gains can be obtained from CIBSE Guide A (2006) - Table 6.3.
Typical gains are shown below.

Conditions	Typical building	Sensible Heat Gain (Watts)	Latent Heat Gain (Watts)
Seated very light work	Offices, hotels, apartments	**70**	45
Moderate office work	Offices, hotels, apartments	**75**	55
Standing, light work; walking	Department store, retail store	**75**	55
Walking standing	Bank	**75**	70
Sedentary work	Restaurant	**80**	80
Light bench work	Factory	**80**	140
Athletics	Gymnasium	**210**	315

LIGHTING – Average power density from Tables 6.4.

ELECTRICAL EQUIPMENT - PC's and Monitors - Tables 6.7 and 6.8.
Laser Printers and Photocopiers - Tables 6.9 and 6.10
Electric Motors – Table 6.13 and 6.14.
Lift Motors – Table 6.15.
Cooking equipment - Table 6.17.

Heat load is found from;

$Q_{int.}$ = Heat from Occupants + Heat from Lighting + Heat from Electrical Equipment + Heat from Cooking

4.0 Heat Gain Through Walls

This is the unsteady-state heat flow through a wall due to the varying intensity of solar radiation on the outer surface.

4.1 Sol-Air Temperature

In the calculation of this heat flow use is made of the concept of **sol-air temperature**, which is defined as;

the value of the outside air temperature which would, in the absence of all radiation exchanges, give the same rate of heat flow into the outer surface of the wall as the actual combination of temperature difference and radiation exchanges.

SOL-AIR TEMP.

$$t_{eo} = t_a + \left(\frac{\alpha \cdot I \cdot \cos a \cdot \cos n + \alpha\, I_s}{h_{so}} \right) \quad \text{........ eqn. 4.1}$$

where

t_{eo} = sol-air temperature (°C)

t_a = outside air temperature (°C)

α = absorption coefficient of surface

I = intensity of direct solar radiation on a surface at right angles to the rays of the sun. (W/m^2)

a = solar altitude (degrees)

n = wall-solar azimuth angle (degrees)

I_s = intensity of scattered radiation normal to a surface (W/m^2)

h_{so} = external surface heat transfer coefficient ($W/m^2\,°C$)

The U.K. values of sol-air temperature are found from CIBSE Guide J (2002) Table 5.36 (London), Table 5.37 (Manchester) and Table 5.38 (Edinburgh).

4.2 Thermal Capacity

The heat flow through a wall is complicated by the presence of thermal capacity, so that some of the heat passing through it is stored, being released at a later time.

Thick heavy walls with a high thermal capacity will damp temperature swings considerably, whereas thin light walls with a small thermal capacity will have little damping effect, and fluctuations in outside surface temperature will be apparent almost immediately.

The thermal capacity will not affect the **daily mean solar gain** but will affect the solar gain at a particular time. The calculation is, therefore, again split into two components.

1. Mean gain through wall,

$$Q_0 = A \,.\, U \,.\, CLTD_c \qquad \text{........ eqn. 4.2}$$

where,

Q_0	=	heat gain through wall at time θ
A	=	area of wall (m^2)
U	=	overall thermal transmittance (W/m^2 ºC)
$CLTD_c$	=	Cooling load temperature difference correction

5.0 Heat Gain Through Roof

The heat gain through a roof uses the same equation as for a wall as shown below.

$$Q_{(\theta)\,Roof} = A \,.\, U \,.\, CLTD_c \qquad \text{........ eqn. 5}$$

6.0 Ventilation and/or Infiltration Gains

Heat load is found from;

$$Qsi = n \cdot V \ (t_o - t_r) \ / \ 3 \quad \text{....... eqn. 6}$$

where	Q_{si}	=	Sensible heat gain (W)
	n	=	number of air changes per hour (h^{-1})
	V	=	volume of room (m^3)
	t_o	=	outside air temperature (^oC)
	t_r	=	room air temperature (^oC)

Infiltration gains should be added to the room heat gains.
Recommended infiltration rates are 1/2 air change per hour for most air-conditioning cases or 1/4 air change per hour for double glazing or if special measures have been taken to prevent infiltration.

Ventilation or fresh air supply loads can be added to either the room <u>or</u> central plant loads but should only be accounted for once.

Total Room Load From Heat Gains

$$\mathbf{Q_{total} = Q_g + Q_{sg} + Q_{int.} + Q_{\theta+\phi\,Wall} + Q_{\theta+\phi\,Roof} + Q_{si}}$$

Q_{total}	=	$A_g \cdot U_g (t_o - t_r)$	1. Sensible Glass
	+	$SHG.SC.CLF.A_g$	2. Solar Glass.
	+	$Q_{int.}$	3. Internal
	+	$A.U .CLTD_C$	4. Walls
	+	$A.U . CLTD_C$	5. Roof
	+	$n \cdot V \ (t_o - t_r) \ / \ 3$	6. Ventilation

.... eqn. 7

In the majority of cases, by far the greatest external fluctuating component is the solar heat gain through the **windows.**
Therefore, it will be this gain which determines when the total heat gain to the room is a **maximum.**

Heat gains may be calculated and displayed in table form as shown below.

Heat Gain from	**Watts**	**%**
1. Sensible transmission through glass		
2. Solar gain through glass		
3. Internal		
4. External walls		
5. Roof		
6. Ventilation		
Total		100%
Heat gain per m^2 floor area =		
Heat gain per m^3 space =		

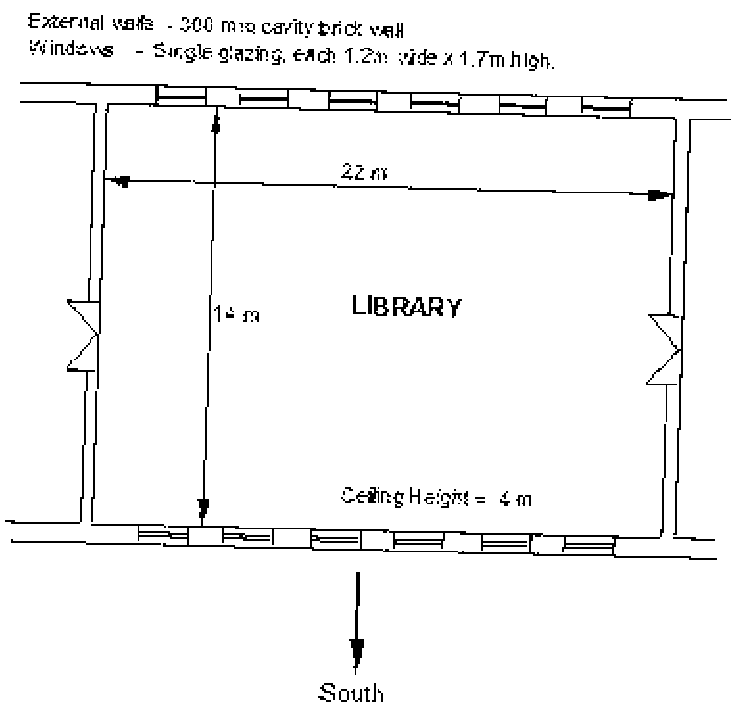
EXAMPLE 1

The room shown below is to be maintained at a constant environmental temperature of 23°C for a plant operation of 12 hours per day.
The room is on the intermediate floor of an Library located in London latitute 51.7°N. The internal construction is lightweight demountable partitions, lightweight slab floors and suspended acoustically treated ceilings.

Calculate the maximum sensible cooling load in the room in July
The outside air temperature (t_o) may be found from CIBSE guide Table A8.3. for July 23rd. The maximum value ocurrs at 15.00 hrs. and is 24.5°C.

DATA:

Occupants	=	100
Infiltration	=	0.5 air changes per hour
Building classification	=	lightweight.
External wall 'U' value	=	0.45 W/m^{2}°C, internal insulation.
External wall colour	=	light.
External wall decrement factor f	=	0.65
Glass type & 'U' value	=	clear 6mm, single, U = 2.80 W/m^{2}°C
Window blinds	=	light slatted blind – open.
Lighting	=	30 Watts / m^2 floor area
Heat gain from machinery and equipment	=	4000 Watts



NOTE: It should be noted that this total heat gain is used to size central plant items such as Chillers, Condensers and Cooling Towers.
Cooling coils are sized usually with a pschrometric chart.

Answer

Areas:

Area of window = $1.2 \times 1.7 = 2.04\ m^2$.
Total area of glass = 2.04×12No. windows = $24.48\ m^2$.
Area of glass facing South = $12.24\ m^2$.

Area of wall facing South = 22.0 m x 4.0 m high = $88\ m^2$ less glass = $88 - 12.24 = 75.76\ m^2$.

Floor area = 22×14 = $308 m^2$.
Room volume = 308×4 = $1232\ m^3$.

Gains:

1. Sensible transmission through glass $Q_g = A_g\, U_g\, (t_o - t_r)$

Q_g = 239.9 Watts

2. Solar Gain through glass $Q_{sg} = F_c\ F_s\ q_{sg}\ A_g$

(Maximum is at 12.00 hrs)

Q_{sg} = 3266.9 Watts

3. Internal

$Q_{int.}$ = 23,240.0 Watts

4. External wall $Q_{\theta+\phi Wall} = A\ U\ CLTD_C$

$Q_{\theta+\phi Wall}$ = 350.3 Watts

5. Roof $Q_{\theta+\phi Roof}$ = Nil for intermediate floor.

6. Ventilation $Q_{si} = n \; V \; (t_o - t_r) \; / \; 3$

Q_{si} = 718.0 Watts

7. $Q_{total} = Q_g + Q_{sg} + Q_{int.} + Q_{\theta+\phi Wall} + Q_{\theta+\phi Roof} + Q_{si}$

$Q_{total} = 239.9 - 2{,}806.9 + 23{,}240.0 + 350.3 + 0 + 718.0$

Q_{total} = 27,354.1 Watts

The results are shown in the table below.

Heat Gain from	Watts	%
1. Sensible transmission through glass	239.9	0.9
2. Solar gain through glass	2,806.9	10.3
3. Internal	23,240.0	84.9
4. External walls	350.3	1.3
5. Roof	0	0
6. Ventilation	718.0	2.6
Total	27,354.1	100%
Heat gain per m^2 floor area = 88.8 W/m^2		
Heat gain per m^3 space = 22.2 W/m^3		

EXAMPLE 2

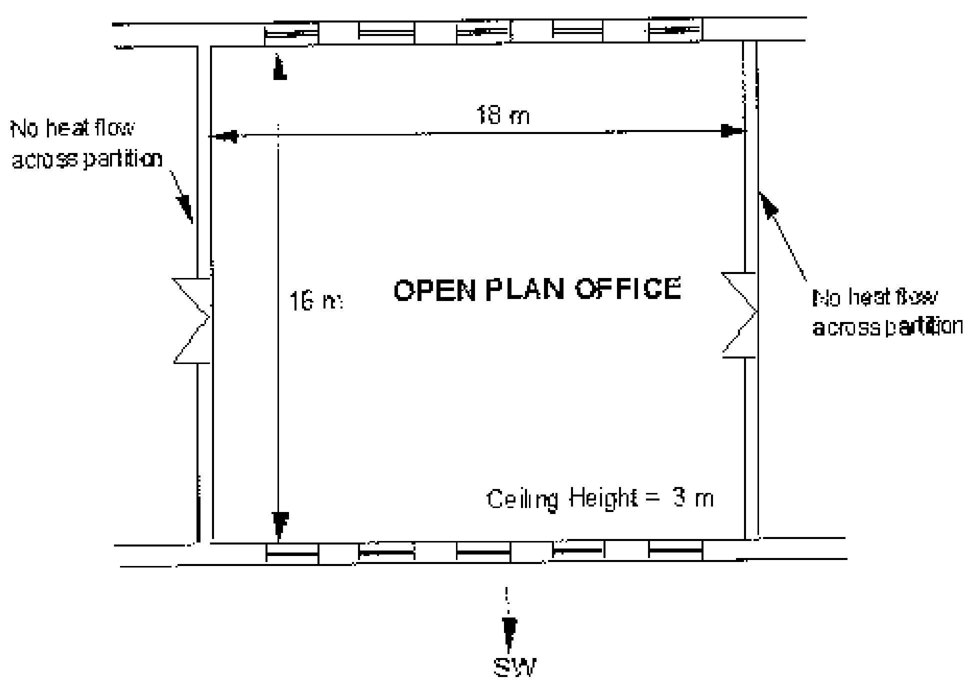
The room shown below is to be maintained at a constant environmental temperature of 2[illegible]°C for a plant operation of 12 hours per day.
The room is on the intermediate floor of an Office Block located in London.
The internal construction is lightweight partitions, concrete hollow slab floors and suspended ceilings.

Calculate the maximum sensible cooling load in the room in July.

DATA:

Occupants = 80
Lighting = 35 Watts / m2 floor area
Infiltration = 1.0 air changes per hour
Outside air temperature (t_o) = 26°C.
Building classification = lightweight.
External wall surface texture = dark.
External wall thickness = 300mm, internal insulation, decrement factor is 0.27.
Light slatted blinds = Assume open
Heat gain from machinery and equipment = 3000 Watts

External walls - 'U' value = 0.35 W/m^2°C
Windows - Single glazing, each 1.2m wide x 1.7m high
'U' value = 5.3 W/m^2°C



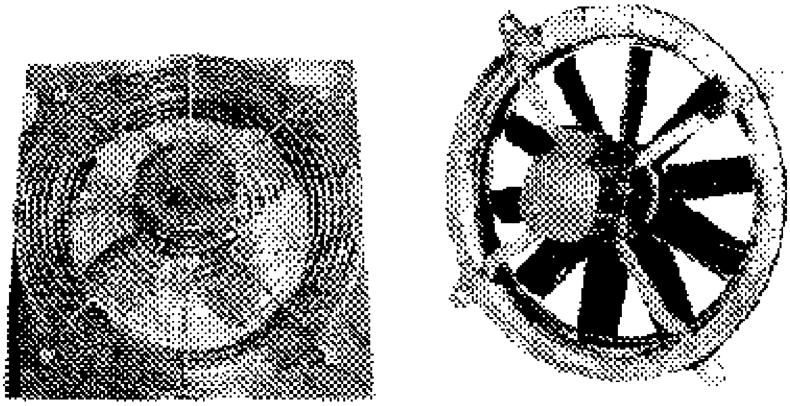
Answer

Heat Gain from	Watts	%
1. Sensible transmission through glass		
2. Solar gain through glass		
3. Internal		
4. External walls		
5. Roof	0	0
6. Ventilation		
Total		100%
Heat gain per m^2 floor area =		
Heat gain per m^3 space =		

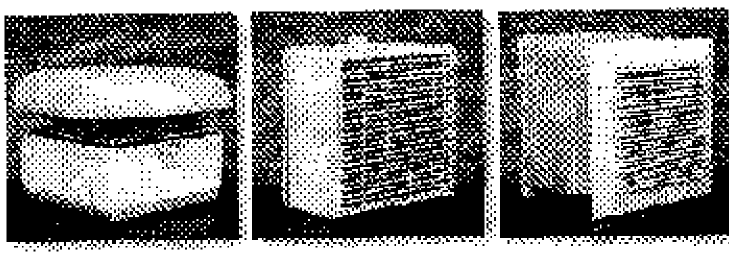
Types of Fans

There are several types of fan to choose from in ventilation.
These are:

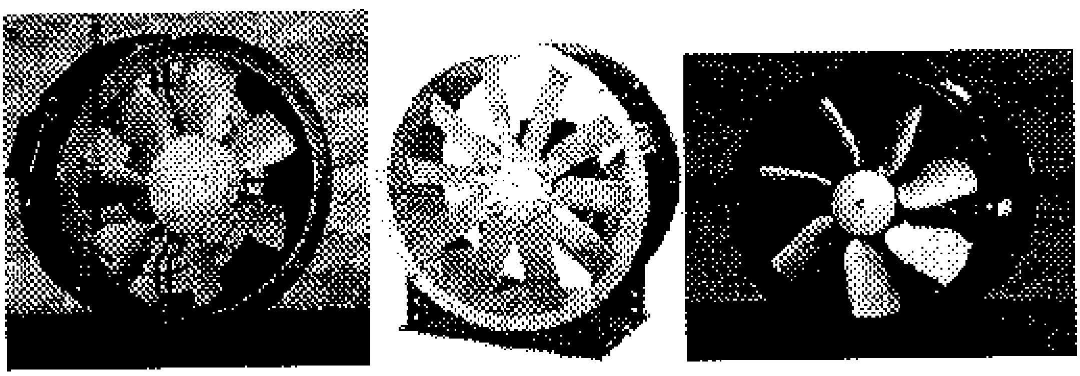
1. **Propeller**
2. **Axial flow**
3. **Centrifugal**
4. **Mixed flow**



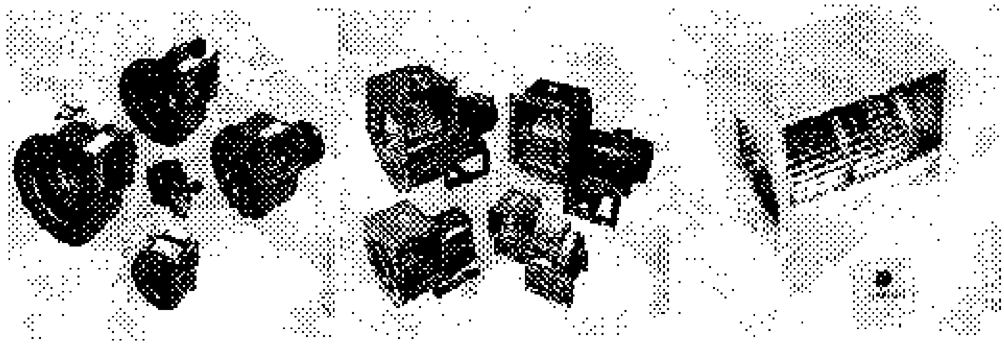
PROPELLER FANS

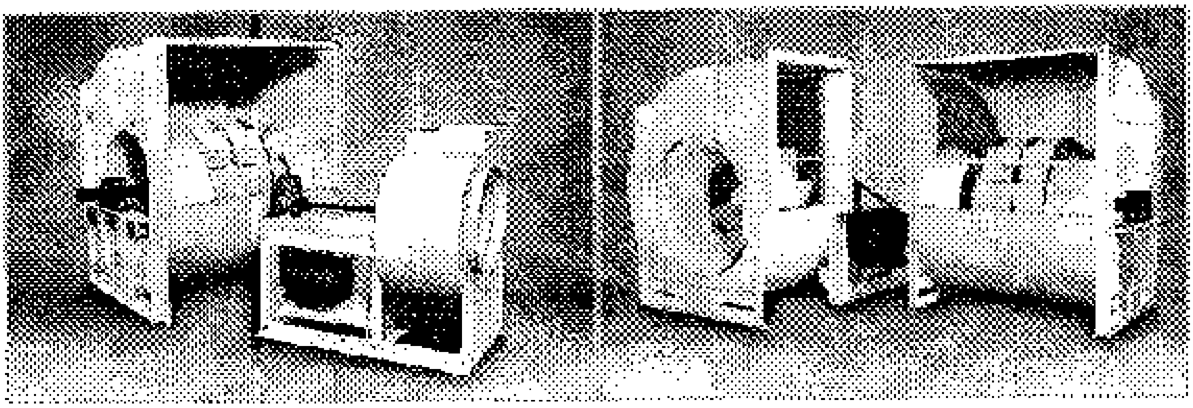


SMALL COMMERCIAL ROOF EXTRACT, WINDOW AND WALL FANS

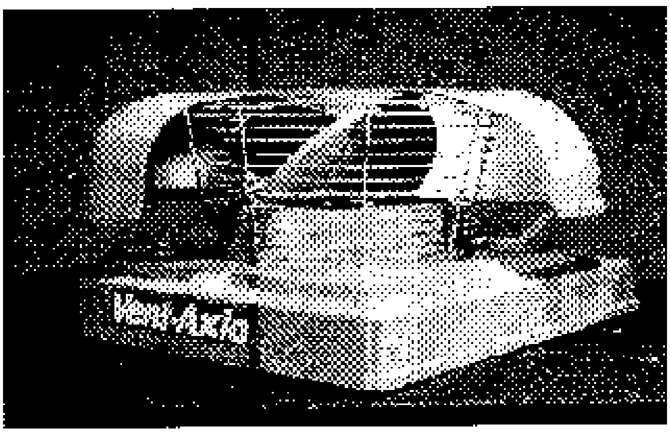


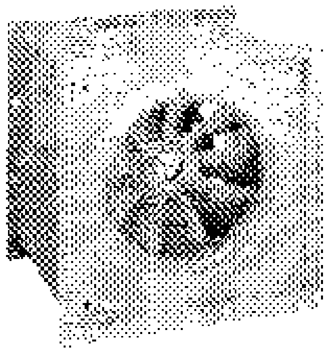
AXIAL FLOW FANS





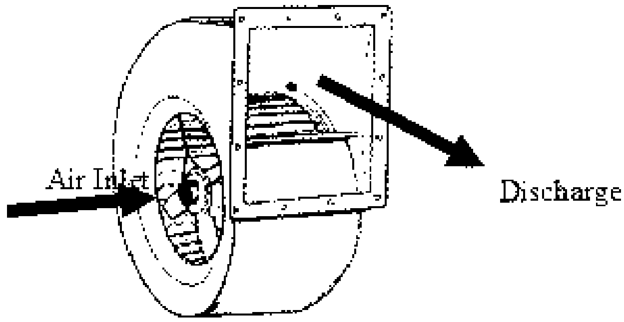
CENTRIFUGAL FANS



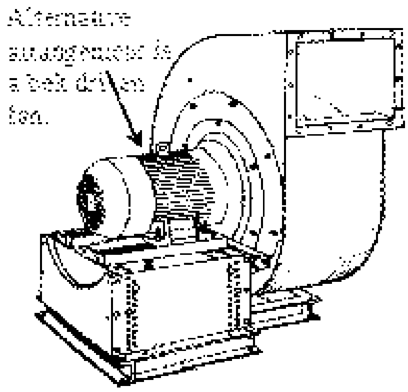


MIXED FLOW FANS – ROOF UNIT AND BOXED FAN

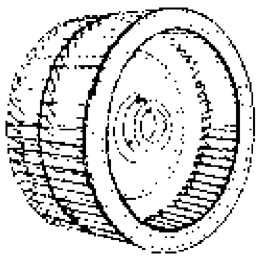
Centrifugal Fans



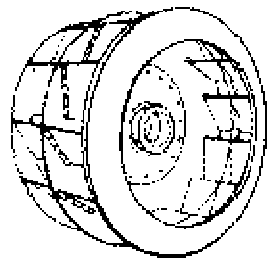
CENTRIFUGAL FAN IMPELLER
WITH CASING



CENTRIFUGAL FAN WITH CLOSE
COUPLED ELECTRIC MOTOR

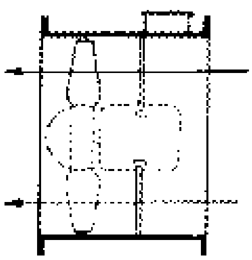


FORWARD CURVED BLADES FOR
CENTRIFUGAL FAN



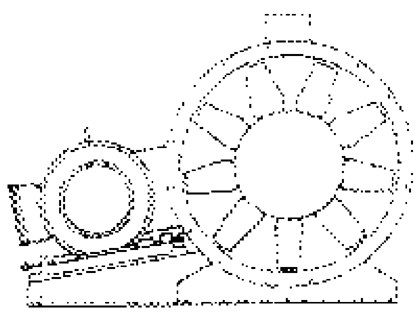
BACKWARD CURVED BLADES
FOR CENTRIFUGAL FAN

Axial Flow Fans



AXIAL FLOW
FAN

AEROFOIL BLADE OF
AN AXIAL FLOW FAN



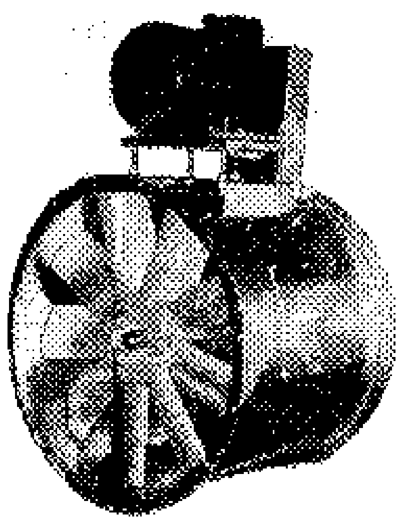
BELT-DRIVEN AXIAL
FLOW FAN

1. Propeller Fan

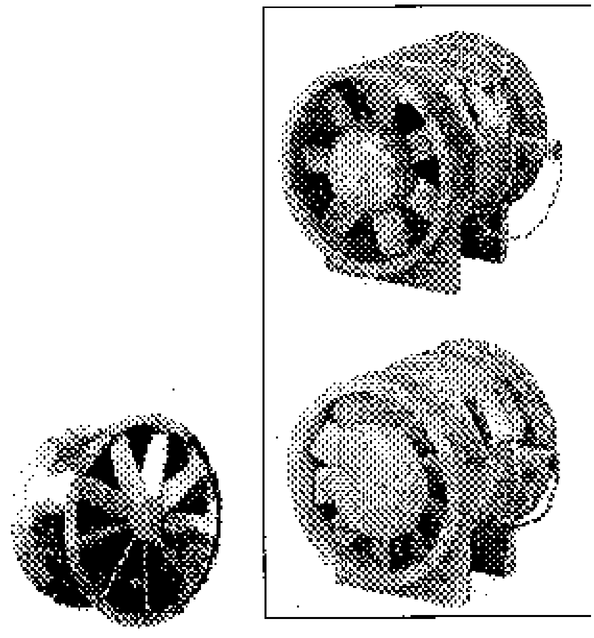
Used in situations where there is minimal resistance to air flow.
Typical outputs are; up to **4 m^3/s** and up to **250 Pa pressure**.
Fan efficiency is low at about **40%**.
Suitable for wall, window and roof fans where the intake and discharge are free from obstacles.
Can move large volumes of air.
Low installation cost.

2. Axial Flow Fan

High volume flow rate is possible with this type of fan with high efficiency, about **60% to 65%**.
Typical outputs are; up to **20 m^3/s** and up to **700 Pa pressure**.
The fan is cased in a simple enclosure with the motor housed internally or externally.



Axial Flow Fan with External Motor

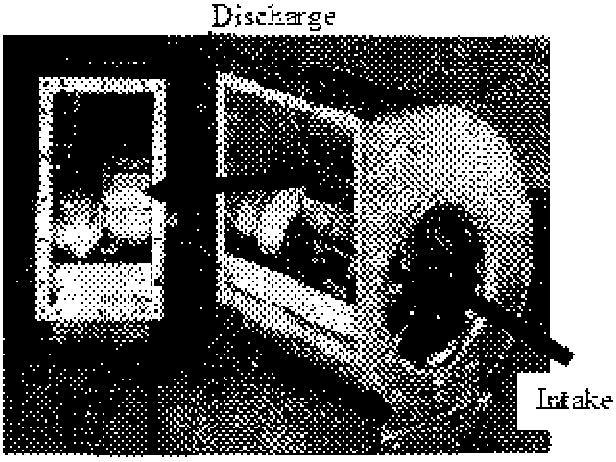


Axial Flow Fans with Internal Motor

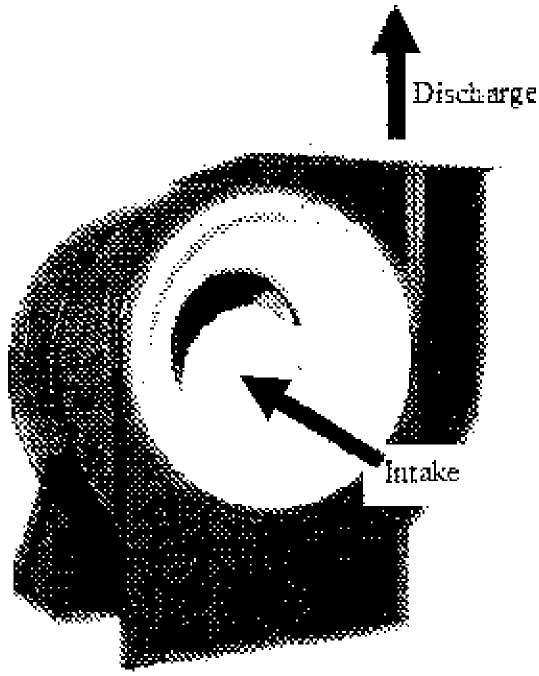
Aerofoil blades can be used to increase efficiency.
Adjustable pitch blades can be used for greater flexibility.
Ductwork can be simply connected to the flange at either end of the fan.

3. Centrifugal Fan

High pressure air flow is possible with this type of fan.
Used in air handling units and other situations to overcome high resistance to air flow.
The impeller is made of thin blades which are either forward or backward curved.
The air changes direction by 90 degrees in a centrifugal fan so more space is required.



Small Centrifugal Fans



Large Centrifugal Fan

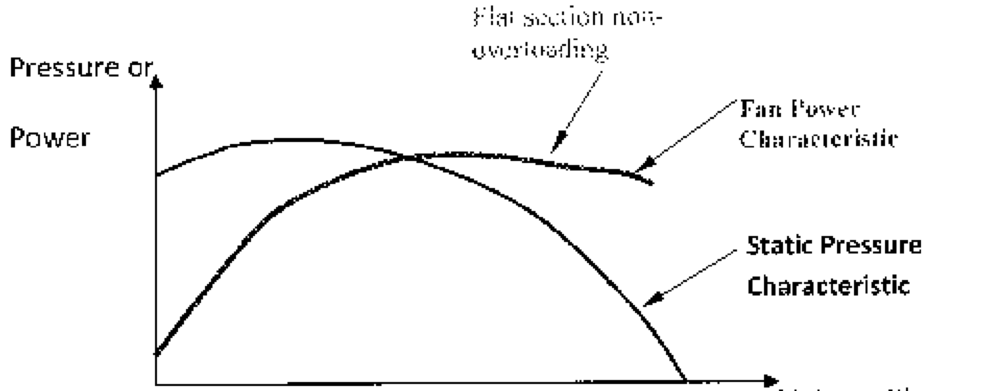
Usually the motor is placed external to the casing and a vee belt and pulley drive is commonly used.

Centrifugal Blades

Centrifugal curved fan blades generally have **higher efficiencies** than if a plain flat blade is used.
The efficiency of a fan with forward curved blades is about **50% to 60%.**
The forward curve has a scoop effect on the air thus a higher volume may be handled.

FORWARD CURVED BLADE

Backward curved blades offer even better efficiency, **70% to 75%.**
This improves airflow through the blade and reduces shock and eddy losses.
High pressures can be developed with backward curved blades.
Even further improvements may be made by using an aerofoil section blade in which case the efficiency may be **80% to 85%.**
Another feature of backward curved blades is their non-overloading characteristic.

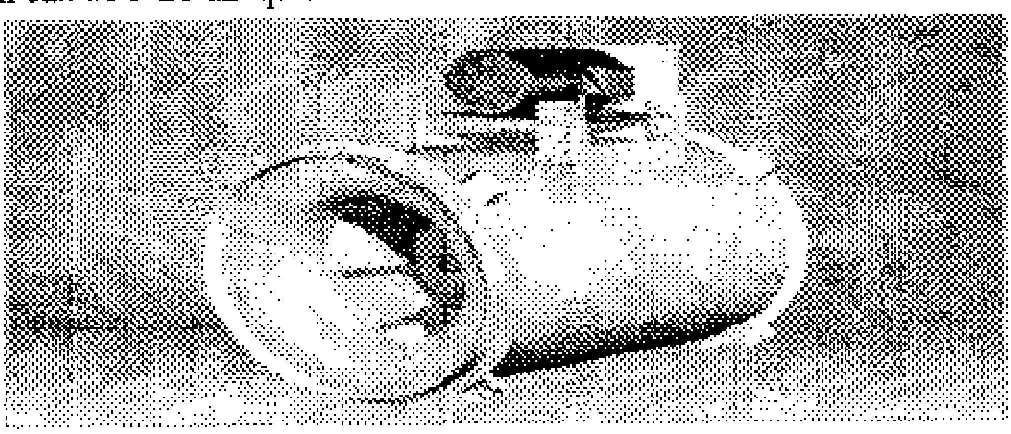


FAN CHARACTERISTIC CURVES

A disadvantage is the high blade tip speed, necessary to obtain a comparable rate of discharge to forward curved blades, makes the fan noisy.

4. Mixed Flow Fan

Mixed Flow fans can be used for return air, supply, or general ventilation applications where low sound is critical. As compared to similarly sized axial fans, a mixed flow fan can be 5-20 dB quieter.

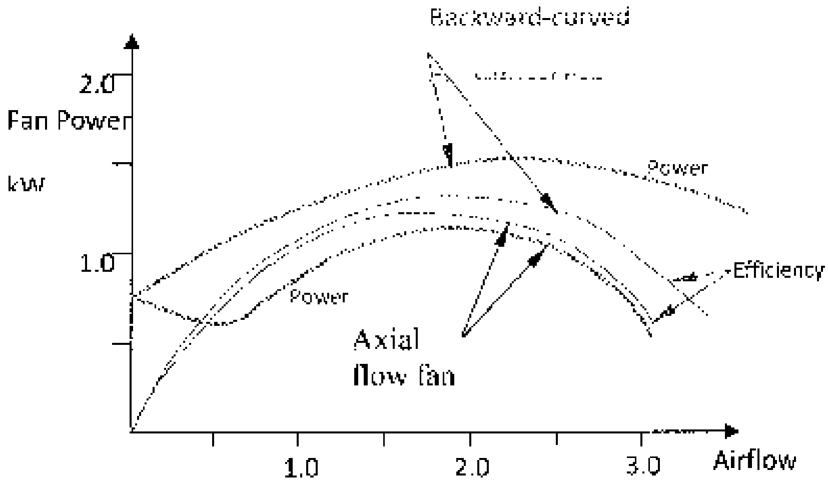


MIXED FLOW FAN

Characteristics of Axial Flow and Centrifugal Fans

Axial Flow Fans

1. Axial flow and backward-curved centrifugal fans have similar characteristics as shown below.



TYPICAL FAN CHARACTERISTIC CURVES

2. The axial flow fan is very convenient from an installation point of view, it can be directly duct mounted even in restrictive areas but they tend to be noisy. This is because they run at a higher speed compared to a centrifugal fan.

3. Like the Backward-bladed centrifugal fan, the Axial flow fan has a self-limiting power curve as shown above.

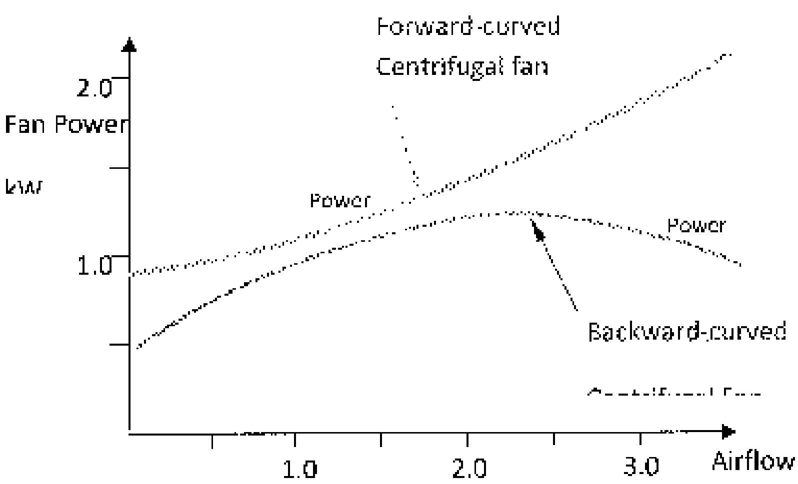
Centrifugal Fans

4. The backward curved centrifugal fan runs at a higher speed than the forward curved fan for the same output.

5. A forward-curved centrifugal fan may be liable to overloading because the power rises as the volume increases. An example of this in practice is if the main dampers are left wide open when the fan is first started up, too much air

will be handled and the excessive power absorbed will overload the driving motor.

6. The backward curved fan is less liable to over-loading than the forward-curved centrifugal fan and it is also able to deliver a relatively constant amount of air as the system resistance varies. The power of a backward curved fan reaches a peak and then begins to fall, this is called the self-limiting characteristic. This is shown below.



CENTRIFUGAL FAN CHARACTERISTICS

7. A backward-curved centrifugal fan must run at higher speed to deliver the same amount of air as a forward-curved fan because of the shape of the impeller blades and the direction of rotation.

8. The backward-bladed fan is used in high velocity systems where high pressures are required and is often made with aerofoil blades to increase efficiency.

9. Up to about 750 N/m^2 fan pressure, the forward-curved centrifugal fan tends to be quieter and cheaper. Above this value of pressure backward-curved fans take over

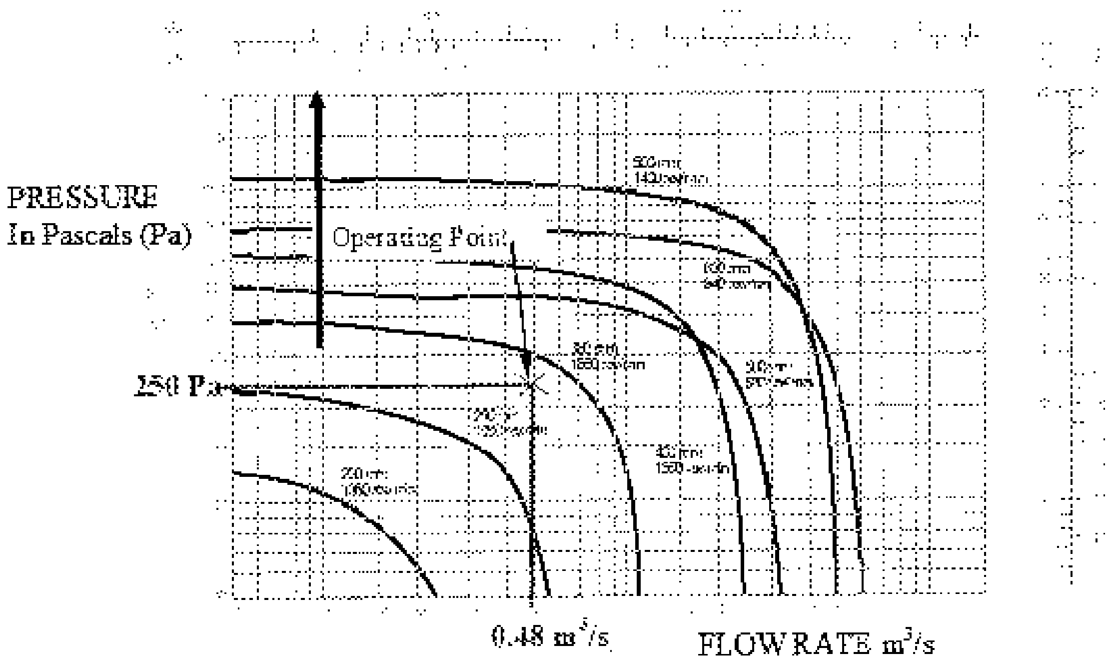
Choosing a Fan

To choose a suitable fan one must look at the performance curves.
Performance curves are found in fan catalogues.

These curves show the pressure developed by a fan at a given flow rate.
The pressure to be developed by the fan is found from duct sizing data (See DUCT SIZING section) and the flow rate is found from design data (See VENTILATION DESIGN section).
The **operating point** of the system is marked as a point on the curve.

Example 1

The example below shows a **system operating point** of 250 Pascals (Pa) pressure and 0.48 (m^3/s) flow rate.

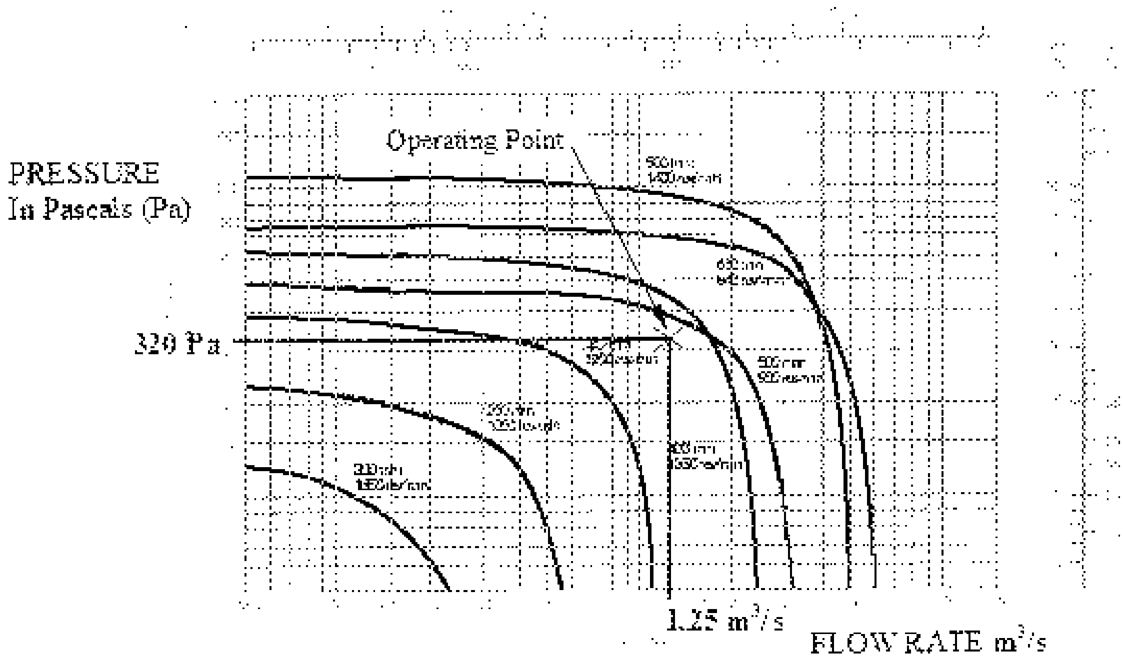


CENTRIFUGAL FAN PERFORMANCE CURVES

Go to the curve above the operating point, this is the fan curve for the appropriate fan.
The fan size is chosen as a **250mm-diameter** fan (1350 r.p.m. speed).

Example 2

The example below shows a **system operating point** of 320 Pascals (Pa) pressure and 1.25 (m^3/s) flow rate.



CENTRIFUGAL FAN PERFORMANCE CURVES

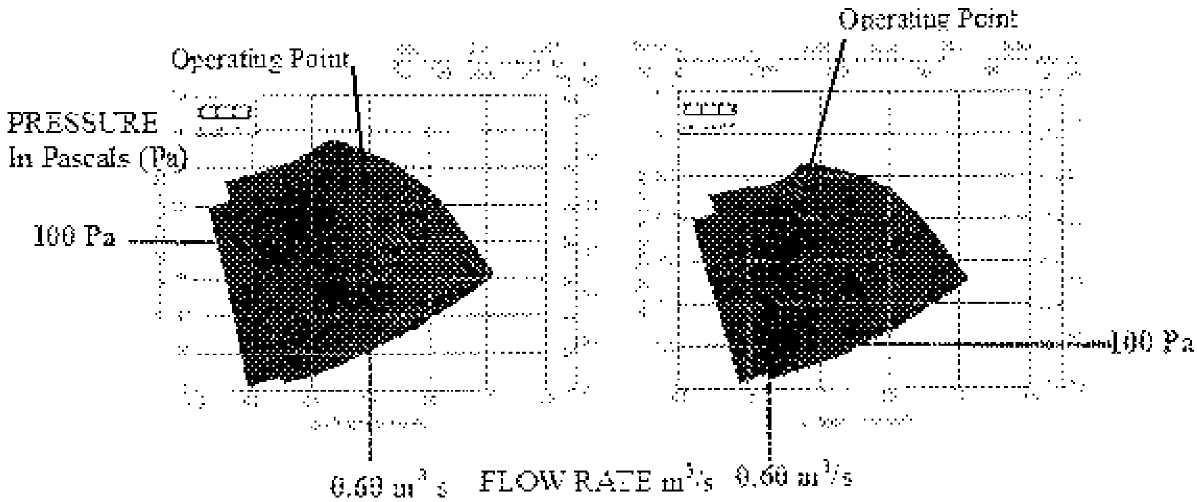
The fan performance curve for a **400mm-diameter** fan will be suitable for the requirements for this example since the curve is above the operating point.

The fan size is chosen as a **400mm-diameter** fan (1350 **r.p.**m. speed).

Example 3

An axial flow fan is required for a ventilation system for a **Workshop.**
Four fans are represented below in the four curves – 2 green and 2 red curves.
The left-hand diagram shows fans with 4-pole electric motors, and the right hand diagram shows fans with 2-pole electric motors.

Four pole electric motors are slower than two pole motors, in this example 4-pole is at 1420 r.p.m. and 2-pole is at 2840 r.p.m.
The **system operating point** requirements are 100 Pascals (Pa) pressure and 0.60 (m^3/s) flow rate.



Axial Flow Fan
RED CURVE 310mm Diameter
GREEN CURVE 350mm Diameter
Electric Motor – 4 pole windings

Axial Flow Fan
RED CURVE 310mm Diameter
GREEN CURVE 350mm Diameter
Electric Motor – 2 pole windings

The fan size is chosen as a **350mm-diameter** fan (1420 **r.p.**m. speed).
The electric motor for this fan has a 4-pole winding and will run at 1420 r.p.m. which will be slower than a 2-pole motor and therefore quieter

Fan Laws

The Fan Laws are as follows:

No.1 Speed / Volume

$$\frac{N_1}{N_2} = \frac{Q_1}{Q_2}$$

Where;

N = Fan speed (rev. per minute or r.p.m.)

Q = Volume flow rate of air (m^3/s)

This means that fan speed and volume flow rate of air are directly proportional.

No.2 Speed / Pressure

$$\left(\frac{N_1}{N_2}\right)^2 = \frac{p_1}{p_2}$$

Where:

N = Fan speed (rev. per minute or r.p.m.)

p = Fan pressure (N/m^2)

This means that as the fan speed is doubled, for example, the pressure developed is raised by a factor of 4.

No.3 Speed / Power

$$\left(\frac{N_1}{N_2}\right)^3 = \frac{P_1}{P_2}$$

Where;

N = Fan speed (rev. per minute or r.p.m.)

P = Fan power (Watts)

This means that as the fan speed is doubled, for example, the power required to drive the fan is raised by a factor of 8.

The above **three laws** may be written differently to aid calculations, as follows;

No.1 Speed / Volume

$$Q_2 = Q_1 \left(\frac{N_2}{N_1}\right) \quad \text{or} \quad N_2 = N_1 \left(\frac{Q_2}{Q_1}\right)$$

No.2 Speed / Pressure

$$p_2 = p_1 \left(\frac{N_2}{N_1}\right)^2 \quad \text{or} \quad N_2 = N_1 \left(\frac{p_2}{p_1}\right)^{1/2}$$

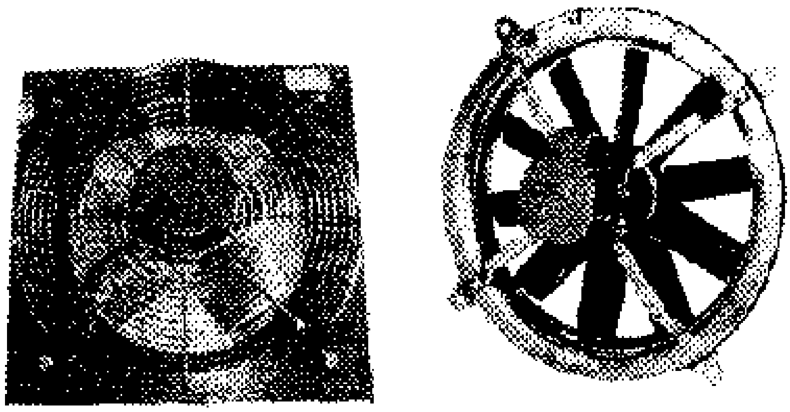
No.3 Speed / Power

$$P_2 = P_1 \left(\frac{N_2}{N_1}\right)^3 \quad \text{or} \quad N_2 = N_1 \left(\frac{P_2}{P_1}\right)^{1/3}$$

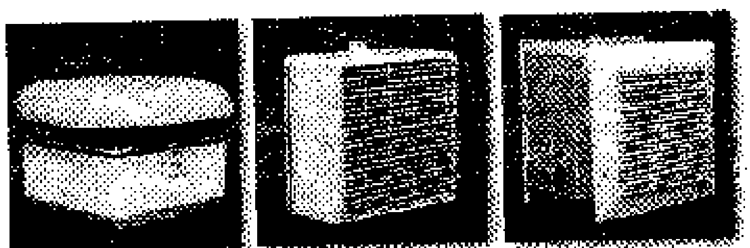
Types of Fans

There are several types of fan to choose from in ventilation.
These are:

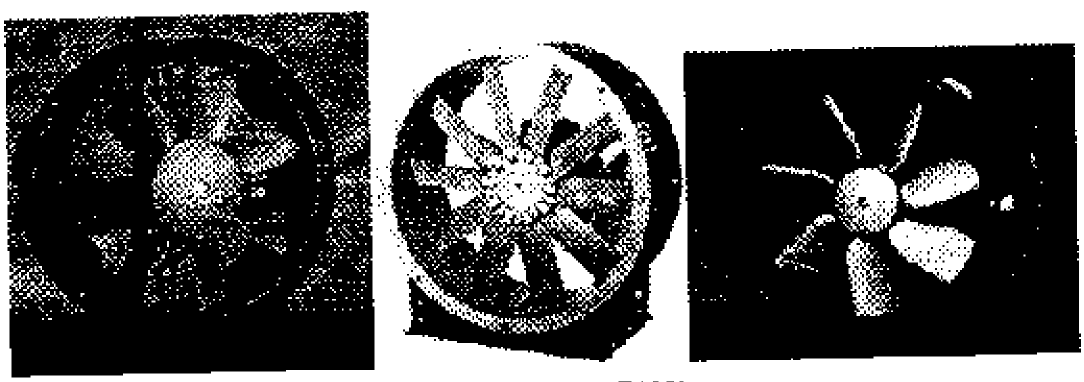
1. **Propeller**
2. **Axial flow**
3. **Centrifugal**
4. **Mixed flow**



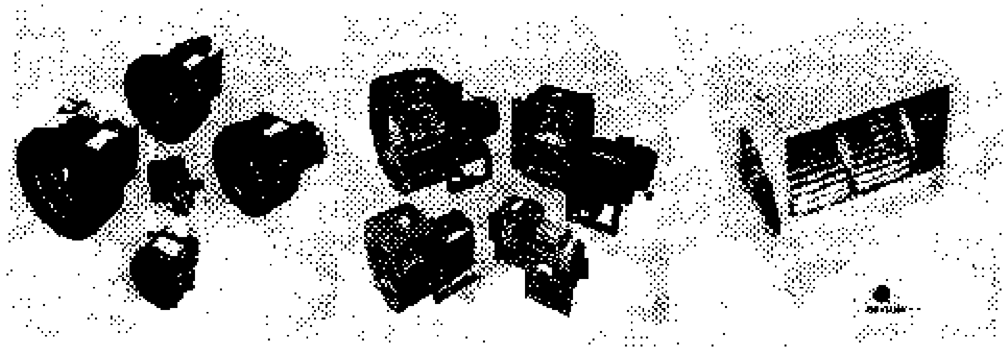
PROPELLER FANS

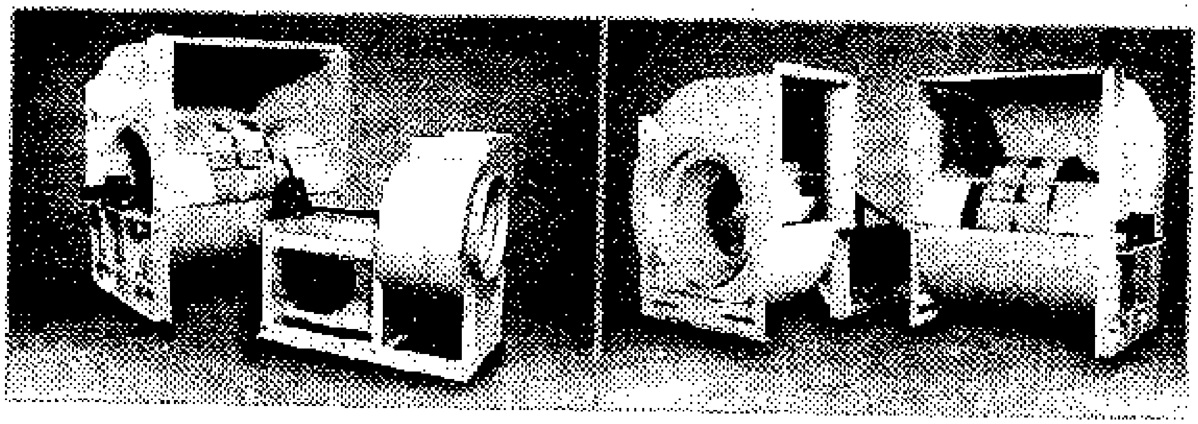


SMALL COMMERCIAL ROOF EXTRACT, WINDOW AND WALL FANS

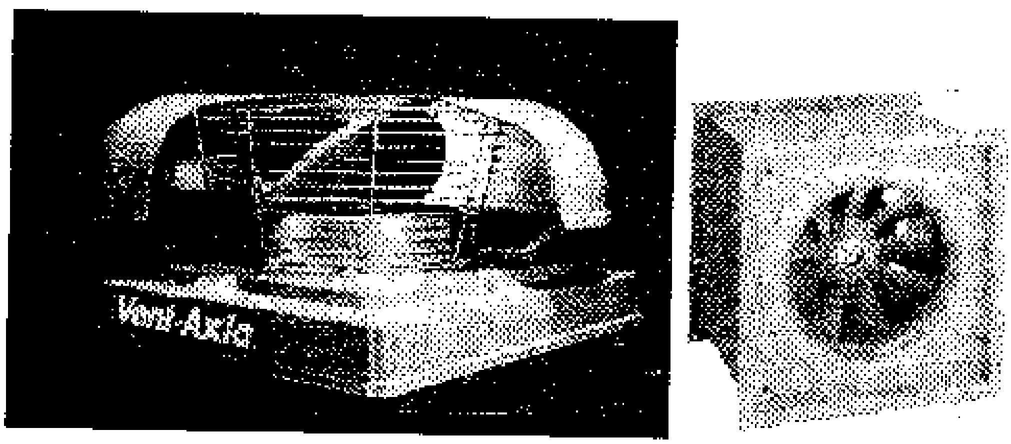


AXIAL FLOW FANS



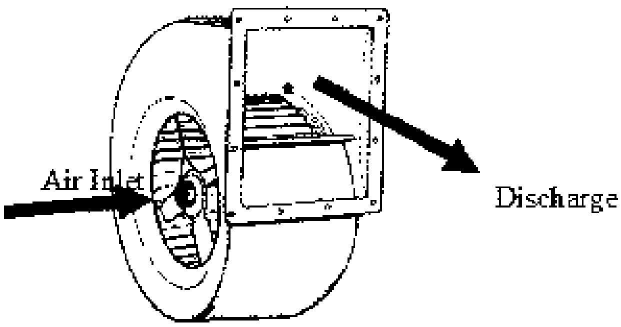


CENTRIFUGAL FANS

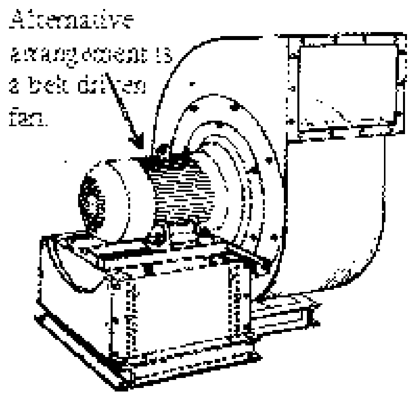


MIXED FLOW FANS – ROOF UNIT AND BOXED FAN

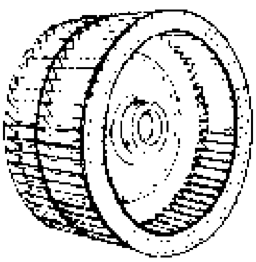
Centrifugal Fans



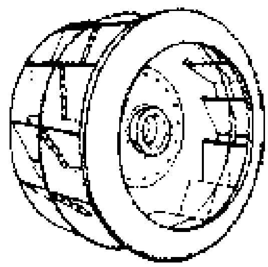
CENTRIFUGAL FAN IMPELLER
WITH CASING



CENTRIFUGAL FAN WITH CLOSE
COUPLED ELECTRIC MOTOR

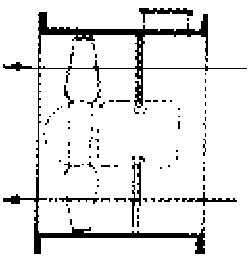


FORWARD CURVED BLADES FOR
CENTRIFUGAL FAN



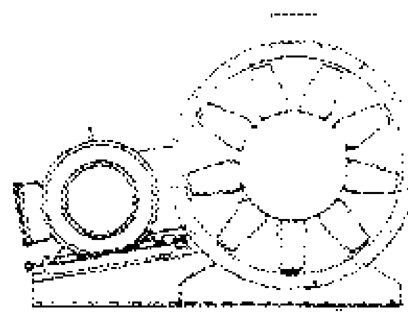
BACKWARD CURVED BLADES
FOR CENTRIFUGAL FAN

Axial Flow Fans



AXIAL FLOW
FAN

AEROFOIL BLADE OF
AN AXIAL FLOW FAN



BELT-DRIVEN AXIAL
FLOW FAN

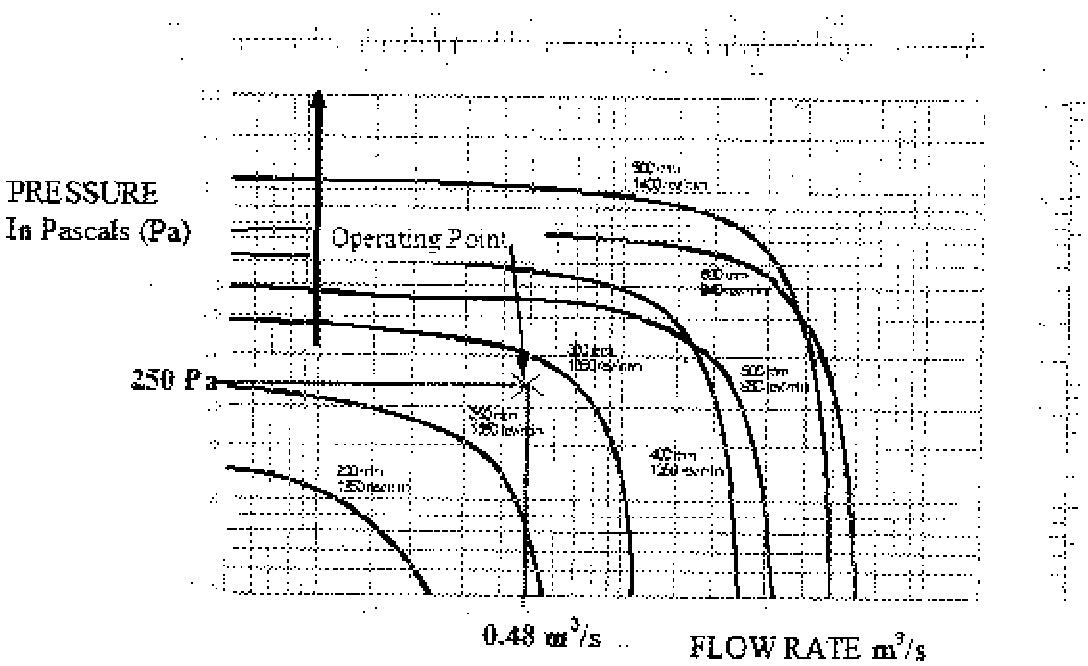
Choosing a Fan

To choose a suitable fan one must look at the performance curves. Performance curves are found in fan catalogues.

These curves show the pressure developed by a fan at a given flow rate.
The pressure to be developed by the fan is found from duct sizing data (See DUCT SIZING section) and the flow rate is found from design data (See VENTILATION DESIGN section).
The **operating point** of the system is marked as a point on the curve.

Example 1

The example below shows a **system operating point** of 250 Pascals (Pa) pressure and 0.48 (m^3/s) flow rate.

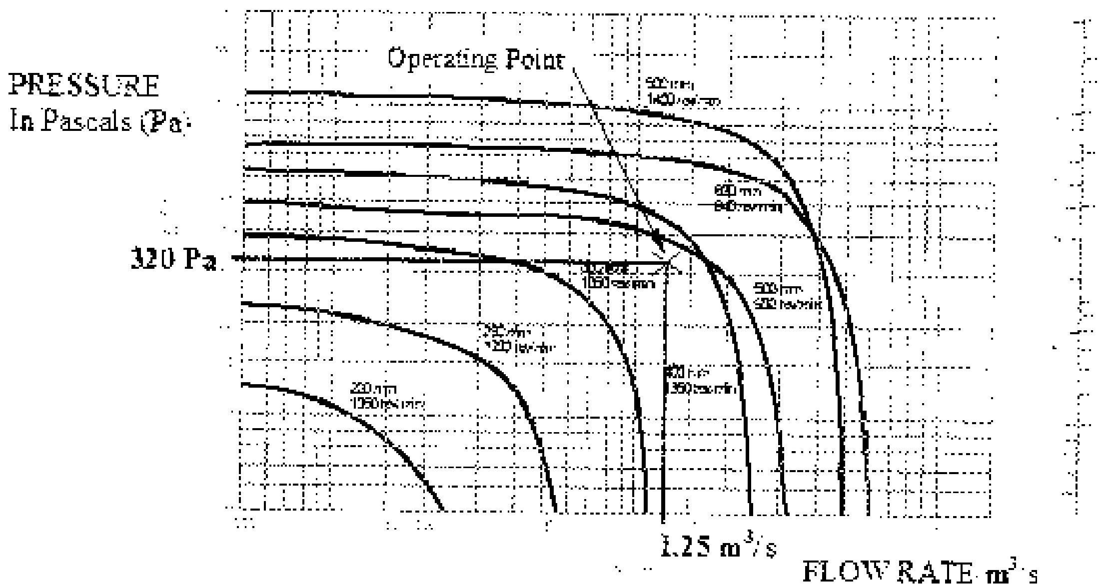


CENTRIFUGAL FAN PERFORMANCE CURVES

Go to the curve above the operating point, this is the fan curve for the appropriate fan.
The fan size is chosen as a **250mm-diameter** fan (1350 r.p.m. speed).

Example 2

The example below shows a **system operating point** of 320 Pascals (Pa) pressure and 1.25 (m^3/s) flow rate.

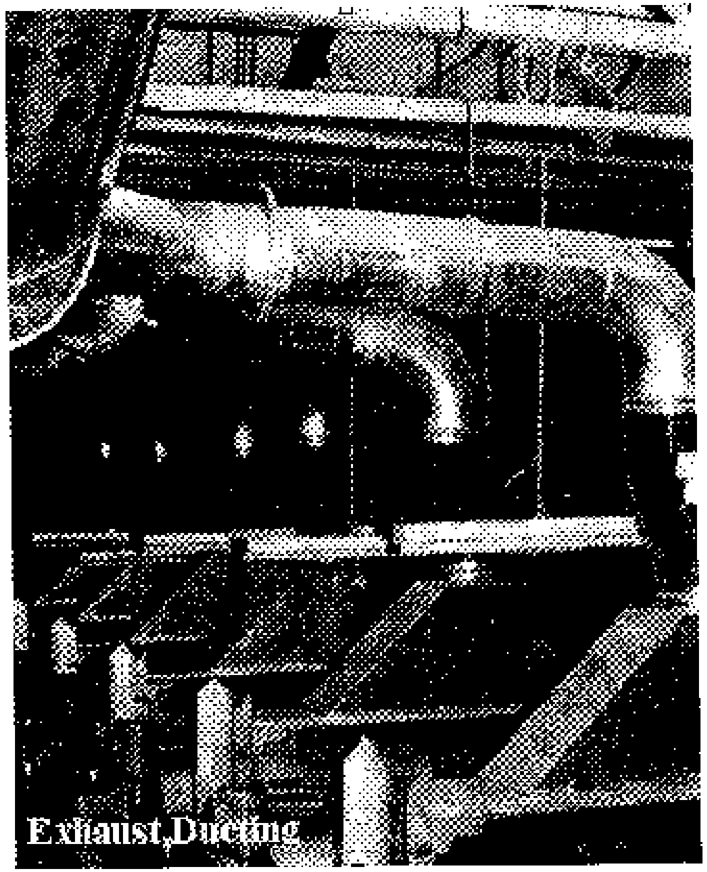


CENTRIFUGAL FAN PERFORMANCE CURVES

The fan performance curve for a **400mm-diameter** fan will be suitable for the requirements for this example since the curve is above the operating point.

The fan size is chosen as a **400mm-diameter** fan (1350 r.p.m. speed).

Duct Sizing - Introduction



For conventional **low velocity** ductwork the sizing method most used is by **constant pressure**, that is, the average pressure or resistance to flow per unit length is kept at a constant figure.

The duct sizing chart (Figure 1 below) shows the various pressure drops against air quantity or volume and duct diamete

By selecting a *certain pressure drop*, the required duct diameter can be selected for any given air volume.

When using Figure 1 any resistance per unit length can be selected

Some designers use values as shown below.

1. **Quiet** - Pressure drop 0.4 Pa/m.

2. **Commercial** - Pressure drop 0.6 Pa/m.

3. **Industrial** - Pressure drop 0.8 Pa/m.

However, we will use a pressure drop of **1.0 Pa/m** for examples of duct sizing, always bearing in mind that the designer may wish to use alternative values as listed above.

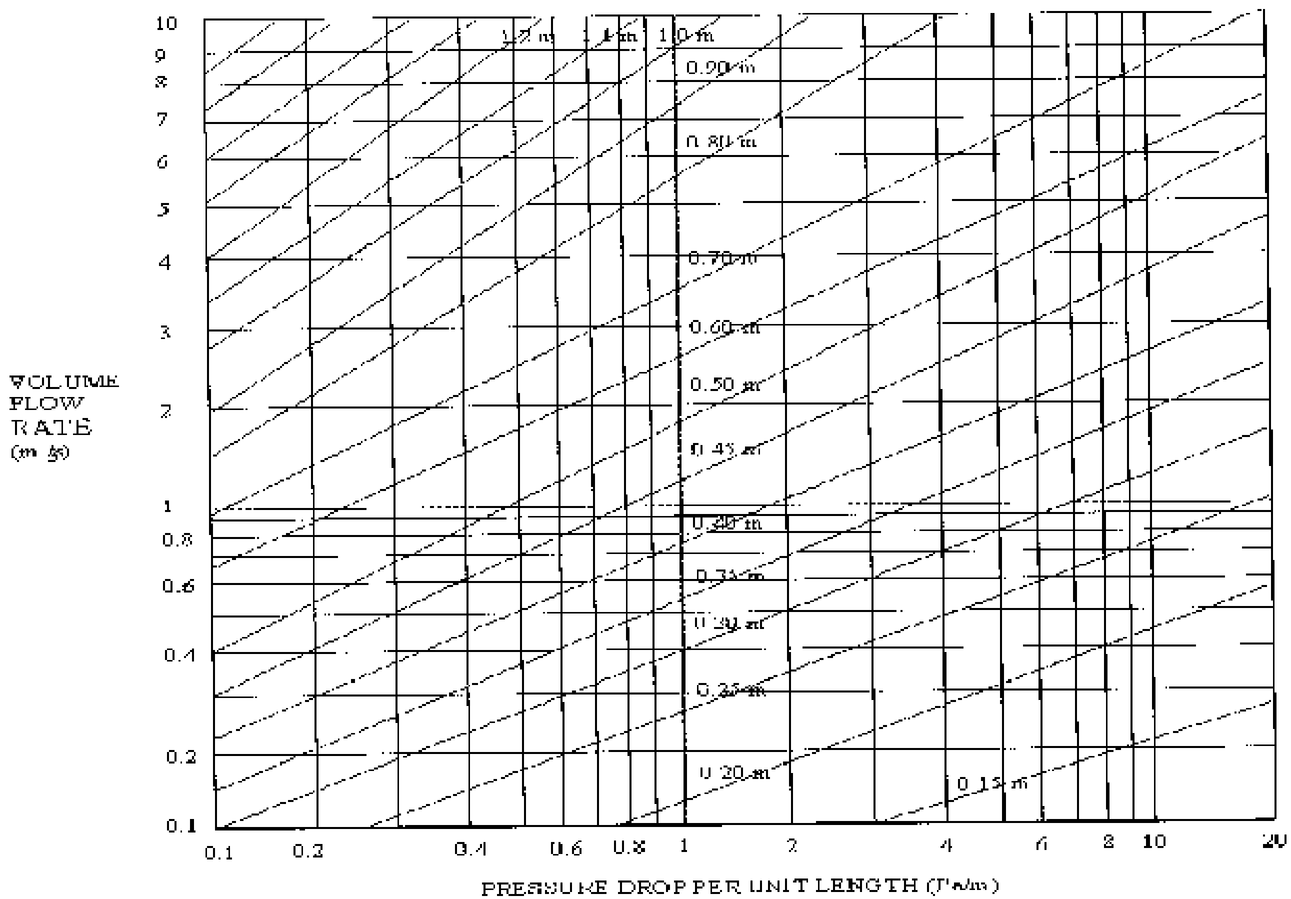


FIGURE 1

FLOW OF AIR IN CIRCULAR DUCTS

Basic Duct Sizing

Consult Figure 1 to determine the size of duct if the volume flow rate of air is known.

Example 1

Size the ventilation ductwork in an extract system, which removes 0.8 m^3/s from a kitchen.
Use a duct pressure drop per metre of 1.0 Pa/m.
The ductwork should be square.

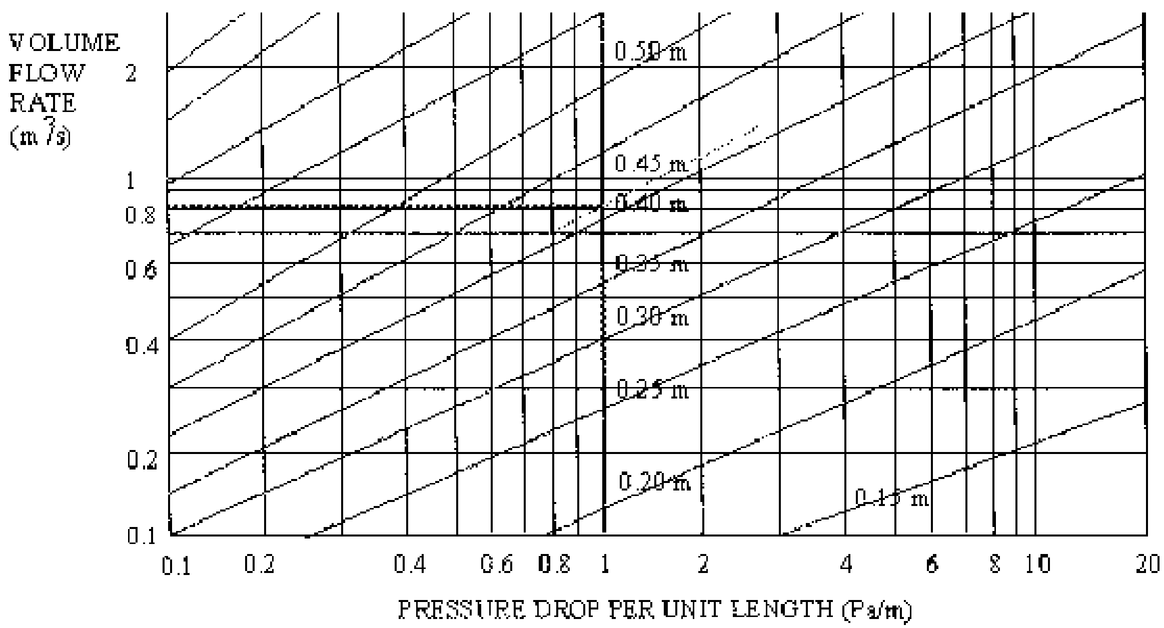


FIGURE 1
FLOW OF AIR IN CIRCULAR DUCTS

From figure 1 above the volume flow rate is indicated with a green line.

The corresponding duct diameter can be read from the diagram.

It is between two of the blue lines, which represent 0.40, and 0.45 metres diameter.

Careful examination of Figure 1 will reveal that the design point is closer to 0.40 metres than 0.45 metres diameter.

The diameter is about 0.41 metres.

This is **410 mm** diameter.

Convert this to an appropriate square size.

Area (Circle) = $\pi \times r^2$

Cross Sectional Area of Duct (CSA) = $\pi \times 0.205^2$ = $0.132\ m^2$

For square duct of same CSA, one side = $(0.132)^{0.5}$ = 0.363 m

= 363 mm

The next standard size of galvanised sheet metal ductwork would be **400mm x 400mm.**

Duct Sizing Using Equal Pressure Drop Method

The following is a step by step approach to duct sizing by keeping the pressure drop the same in straight lengths.

1. Choose a rate of pressure drop and keep this constant for the whole system e.g. **1.0 Pa per metre run.**

2. Size ductwork using Figure 1 (Duct Sizing Chart) if the volume flow rate of air is known.

   This will give the duct diameter.

3. Determine the equivalent size of **rectangular** duct if required by calculation or by using CIBSE guide Table **C4.30.**

4. Calculate the actual air velocity from:

Air velocity (m/s) = volume flow rate (m^3/s) / CSA = Cross sectional area of duct (m2)

Fittings Pressure Loss

5. Determine the velocity pressure factors (ζ zeta) for the fitting(s) in each section of ductwork from CIBSE Guide Table **C4.35.**

6. Determine the **velocity pressure (V.P.)** by calculation or by using CIBSE Guide Table **C4.33.**

   The actual air velocity will be that obtained from section 4 above.

$$\textbf{V.P.} = 0.5 \times \rho \times v^2$$

Where:
- V.P. = Velocity pressure (Pa)
- ρ = Density of air (1.2 kg/m^3)
- v = Air velocity (m/s)

7. Multiply **ζ factors x V.P.** to give total pressure loss for fittings.

Pressure loss for fittings (Pa) = ζ factors x V.P.

Where:
- V.P. = Velocity pressure (Pa)
- ζ factor = Pressure loss factor for a fitting from CIBSE guide Table C4.35.

Total Pressure Drop in Section

8. **Pressure loss in straight duct (Pa) = Rate of pressure drop (1.0 Pa per metre run) x length of section (m).**

9. **Total Pressure drop in Section (Pa) = Pressure loss for fittings (Pa) + Pressure loss in straight duct (Pa)**

Pressure Loss in Fittings

Zeta (ζ) factors are to be used with the **Velocity Pressure** to find fittings resistances.

Zeta (ζ) factors are pressure loss factors.

These allow for the resistance of fittings in ductwork systems which can be quite significant compared to straight runs of duct.

Pressure Loss (or resistance) (Pa) = zeta factor (ζ) x Velocity Pressure (Pa)

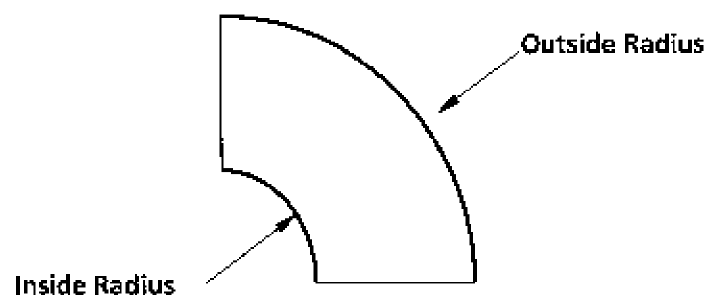
and Velocity Pressure (Pa) V.P. $= 0.5 \times \rho \times v^2$

Where:
- V.P. = Velocity pressure (Pa)
- ρ = Density of air (1.2 kg/m^3)
- v = Actual air velocity (m/s)

Remember, for rectangular ductwork, the velocity used to find Velocity Pressure should be the ***actual*** velocity related to a rectangular duct with a Cross Sectional Area.

Examples of Zeta Factors

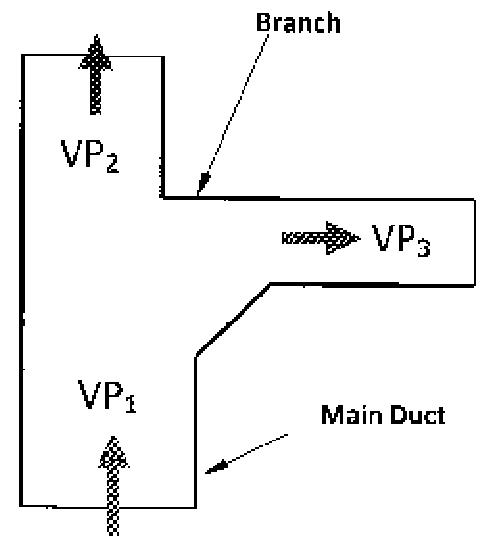
1. Bend – mitred and radiused on both inside and outside.



Zeta factor = 0.67

Straight Thro' 1 to 2		To Branch 1 to 3	
Velocity pressure ratio	**Zeta factor (ζ)**	**Velocity pressure ratio**	**Zeta factor (ζ)**
VP_2/VP_1		VP_3/VP_1	
0.6	0.44	0.6	1.60
0.8	0.09	0.8	0.78
1.0	0.04	1.0	0.55
1.2	0.02	1.2	0.45

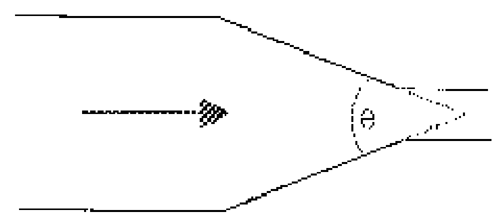
2. Rectangular Duct Branch - read table below.



Decide which duct is to be sized, No2 or No.3.

3. Tapered Reduction – Read Table Below.

Included angle θ	Zeta factor	
	Taper both sides	Taper one side
30^0	0.02	0.07
45^0	0.04	0.20
60^0	0.07	0.40



TAPER BOTH SIDES

TAPER ONE SIDE

Duct Sizing Table

The duct sizing table shown below is an aid to duct sizing. The explanation for use is given in the table below.

A blank table is included in this section at the end.

Duct Sizing Table

1	2	3	4	5	6	7	8	9	10	11	12
								Pressure Loss			
Section	**Length (m)**	**Flow Rate (m^3/s)**	**Pressure drop per metre (Pa/m)**	**Duct Size (mm)**	**Velocity (m/s)**	**Velocity Pressure (Pa)**	**Fittings pressure loss factor or ζ (zeta) factor**	**Fittings (Pa)**	**Straight Duct (Pa)**	**Total Pressure Loss (Pa)**	**Cumulative Pressure Loss (Pa)**
Divide System into sections	Measure lengths of each section from drawing	From Drawing or from grille outputs	Normally 1.0	From FIG 1 For circular Ducts. Convert to Rectangular if necessary	Use formula: Velocity – Volume flow rate / Cross sectional area (CSA)	From Calculation or Table C4.33 V.P = 0.5 x ρ x v^2 Where ρ = air density 1.2 m^3/kg, v = air velocity from col.6.	From (EXAMPLES OF ζ zeta FACTORS) or Table C4.35. e.g. Radiused Bend ζ = 0.67 Contraction on one side (b) with θ = 60^0 gives ζ = 0.4 If more than one fitting..	Velocity pressure from column 7 x total fittings factor from column 8.	Multiply Columns 2 by 4	Add Columns 9 and 10.	Add new Pressure loss for each section to give total at end of system.

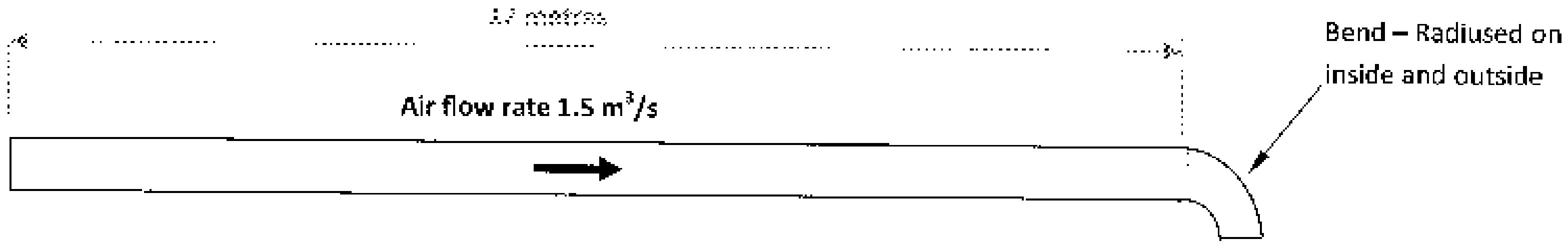
Blan
k
tabl
e
belo
w

Duct Sizing Table

1	2	3	4	5	6	7	8	9	10	11	12
Section	Length (m)	Flow Rate (m^3/s)	Pressure drop per metre (Pa/m)	Duct Size (mm)	Velocity (m/s)	Velocity Pressure (Pa)	Fittings pressure loss factor or ζ (zeta) factor	Pressure Loss		Total Pressure Loss (Pa)	Cumulative Pressure Loss (Pa)
								Fittings (Pa)	Straight Duct (Pa)		

Example 2

Size the section of ductwork shown below and give the total pressure drop for the system.



NOTES:

1. Keep one side 300 mm high.
2. Ductwork to be rectangular galvanised steel.
3. There are no additional pressure losses.
4. Neglect entry or exit losses from the section.

Duct Sizing Table

1	2	3	4	5	6	7	8	9	10	11	12
Section	Length (m)	Flow Rate (m^3/s)	Pressure drop per metre (Pa/m)	Duct Size (mm)	Velocity (m/s)	Velocity Pressure (Pa)	Fittings pressure loss factor or ζ (zeta) factor	Pressure Loss Fittings (Pa)	Pressure Loss Straight Duct (Pa)	Total Pressure Loss (Pa)	Cumulative Pressure Loss (Pa)
A	12	**1.5**	1.0	0.51 metres dia. CSA = 0.204 m^2 Rectangular Width = 0.204 / 0.3 = 0.681 m say duct size is; 700mm x 300 mm high	Vel = vol/CSA = 1.5 / 0.7 x 0.3 = 7.14 m/s	VP = 0.5 x 1.2 x 7.14^2 = 30.6 Pa	1 Bend. ζ factor = 0.67 from (EXAMPLES OF ζ zeta FACTORS) TOTAL ζ factor = 0.67	30.6 x 0.67 = 20.5	12 x 1.0 = 12.0	20.5 + 12.0 = 32.5	**= 32.5**

Some Duct Sizing Aids

1. Divide the system into sections.

2. A section is from one branch to another or in parts of the system with a steady volume flow rate.

3. Size the index circuit first, that is the circuit with the highest resistance to air flow.
   There is only one circuit in the above scheme so the index circuit includes sections A,B,C,D. Normally the index circuit is the longest circuit, but not always so check if necessary.

4. Branches should be included in the downstream section, for example the first branch in the above system should be allowed for in section B resistance calculations.

   This means that the Zeta factor for the branch is multiplied by the correct velocity pressure, that is the smaller velocity pressure as indicated in CIBSE Table. C4.35. see EXAMPLES OF Zeta FACTORS

5. Contractions should be included in the downstream section rather than the upstream section for the same reason as in part 4.

Example 3

Size the index run for the ductwork system shown below and give the total pressure drop for the system.

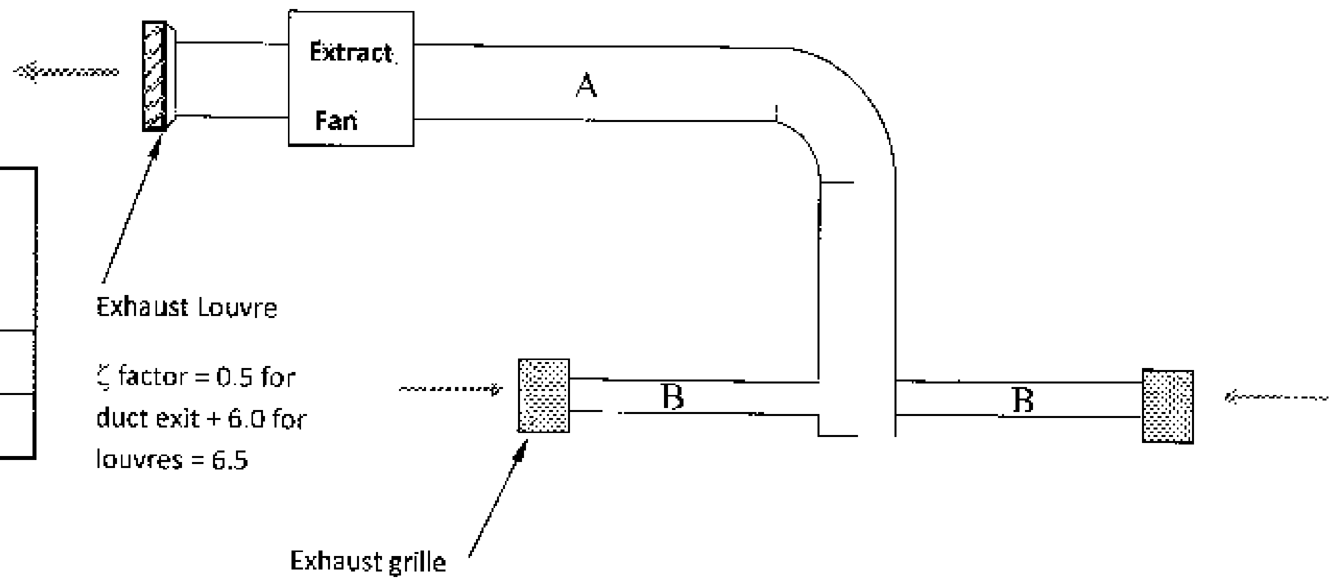
Size the fan for the system.

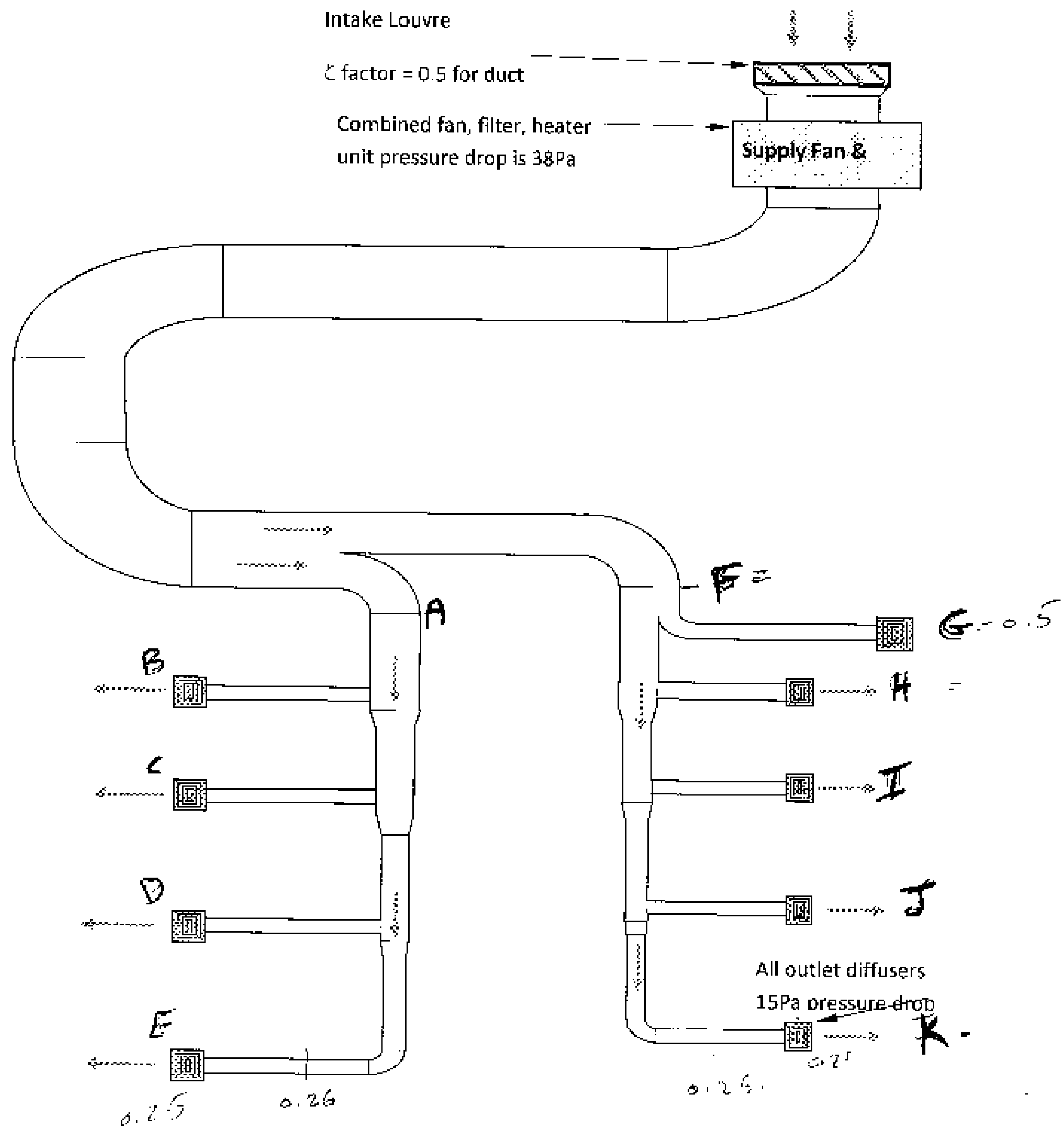
The building is a Garage and general high level extract ventilation is required.

The index run includes sections A,B.

NOTES:
1. Keep one side 200 mm high.
2. Ductwork to be rectangular galvanised steel.
3. There are no additional pressure losses.
4. Tapered reductions are at 30 degrees.

Section	Air Flow Rate (m^3/s)	Length (m)
A	1.00	10
B	0.50	6





Intake Louvre
ζ factor = 0.5 for duct
Combined fan, filter, heater unit pressure drop is 38Pa
Supply Fan &
A
B
C
D
E
F
G - 0.5
H
I
J
K
All outlet diffusers 15Pa pressure drop
0.25
0.26

DUCT SIZING TABLE

SJF 1999

1	2	3	4	5	6	7	8	9	10	11	12
Section	*Length* (m)	Flow Rate (m^3/s)	Pressure drop per metre (Pa/m)	Duct Size (mm)	Velocity (m/s)	Velocity *Pressure* (Pa)	Fittings ζ factor	Pressure Loss Fittings (Pa)	Pressure Loss Straight Duct (Pa)	Total Pressure Loss (Pa)	Cumulative Pressure Loss (Pa)
F	7.0	0.25	1.0	0.26m dia. Width x Height 200 x 300	4.17 or 4.20	10.58	1No. bend @ 0.67 = 0.67	1No. diffuser – 15Pa 10.58 x 0.67 = 7.1Pa **Total** = 22.1Pa	7.0 x 1.0 = 7.0	29.1	29.1

E	5.0	0.50	1.0	0.345m dia. 350 x 300	4.76 or 4.80	13.82	1No. reducer @ 0.04 = 0.04 1No.rectangular duct branch Ratio V_2/V_1 = 10.58/13.82 = 0.77 gives ζ factor = 0.14 Total ζ factor = 0.04 + 0.14 = 0.18	10.58 x 0.18 = 1.9 Pa	5.0 x 1.0 = 5.0	6.9	36.0
D	4.0	0.75	1.0	0.40m dia. 350 x 400	5.36 or 5.40	17.50	1No. reducer @ 0.04 = 0.04 1No.rectangular duct branch Ratio V_2/V_1 = 13.82 / 17.5 = 0.79 gives ζ factor = 0.09 Total ζ factor = 0.04 + 0.09 = 0.13	13.82 x 0.13 = 1.8 Pa	4.0 x 1.0 = 4.0	5.8	41.8

C	4.0	1.00	1.0	0.445m dia. 350 x 500	5.71 or 5.70	19.49	1No. reducer @ 0.04 = 0.04 1No.rectangular duct branch Ratio V_2/V_1 = 17.5 / 19.49 = 0.9 gives ζ factor = 0.065 Total ζ factor = 0.04 + 0.065 = 0.105	17.50 x 0.105 = 1.8 Pa	4.0 x 1.0 = 4.0	5.8	47.6
B	3.0	1.25	1.0	0.48m dia. 400 x 500	6.25 or 6.30	23.81					

[illegible]	[illegible]	[illegible]	[illegible]	[illegible]m dia. 350 x 400	[illegible] or 5.40	17.50	1No. [illegible] 1No.rectangular duct branch Ratio V_2/V_1 = 13.82 / 17.5 = 0.79 gives ζ factor = 0.09 Total ζ factor = 0.04 + 0.09 = 0.13	[illegible] = 1.8 Pa	4.0 x 1.0 = 4.0	5.8	42.8
G	4.0	1.00	1.0	0.445m dia. 350 x 500	5.71 or 5.70	19.49	1No. reducer @ 0.04 = 0.04 1No.rectangular duct branch Ratio V_2/V_1 = 17.5 / 19.49 = 0.9 gives ζ factor = 0.065 Total ζ factor = 0.04 + 0.065 = 0.105	17.50 x 0.105 = 1.8 Pa	4.0 x 1.0 = 4.0	5.8	48.6
K	5.0	0.25	1.0	0.26m dia. Width x Height 200 x 300	4.17 or 4.20	10.58	1No.rectangular duct branch Ratio V_3/V_2 = 10.58 / 19.49 = 0.54 gives ζ factor = 1.6	1No. diffuser – 15Pa 1No. branch = 1.6 x 10.58 = 16.9 Pa Total = 31.9 Pa	5.0 x 1.0 = 5.0	36.9	

EXAMPLE 4

Size the index run for the ductwork system shown below and give the total pressure drop for the system. Size the fan for the system.
The building is an office.
The index run includes sections A, B, C, D, E, and F.

NOTES:

1. No ducts to be higher than 500 mm to fit into suspended ceiling.
2. Ductwork to be rectangular galvanised steel.
3. There are no additional pressure losses other than shown on the drawing.
4. Tapered reductions are at 45degrees.
5. Pressure drop through each diffuser is 15Pa.
6. Sections G, H, I, J, and K have been completed in the table.

Section	Air Flow Rate (m^3/s)	Length (m)
A	2.25	3
B	1.25	3
C	1.00	4
D	0.75	4
E	0.50	5
F	0.25	7
G	1.00	4
H	0.75	4
I	0.50	5
J	0.25	8
K	0.25	5

The drawing below does not show details such as flexible connections to the fan, circular to rectangular transformation sections at the fan, dampers for volume control, and plenum boxes at diffusers, these are often incorporated into ventilation design.

DUCT SIZING TABLE

SIF 1999

1	2	3	4	5	6	7	8	9	10	11	12
Section	*Length* (m)	Flow Rate (m^3/s)	Pressure dopper metre (Pa/m)	Duct Size (mm)	Velocity (m/s)	Velocity *Pressure* (Pa)	Fittings ζ factor	Pressure Loss Fittings (Pa)	 Straight Duct (Pa)	Total Pressure Loss (Pa)	Cumulative Pressure Loss (Pa)
j	8.0	0.25	1.0	0.26m dia. Width x Height 200 x 300	4.17 or 4.20	10.58	1No. bend @ 0.67 = 0.67	1No. diffuser – 15Pa 10.58 x 0.76 = 7.1Pa Total = 22.1Pa	8.0 x 1.0 = 8.0	30.1	30.1
I	5.0	0.50	1.0	0.345m dia. 350 x 300	4.76 or 4.80	13.82	1No. reducer @ 0.04 = 0.04 1No.rectangular duct branch Ratio V_2/V_1 = 10.58/13.82 = 0.77 gives ζ factor = 0.14 Total ζ factor = 0.04 + 0.14 = 0.18	10.58 x 0.18 = 1.9 Pa	5.0 x 1.0 = 5.0	6.9	37.0