This can be rearranged to find the unknown surface t_s °C:

$$F_1 U\left(\frac{1}{U} - R_{si}\right) = \frac{t_s - t_{ao}}{t_c - t_{ao}}$$

$$t_s - t_{ao} = (t_c - t_{ao}) \times F_1 U \times \left(\frac{1}{U} - R_{si}\right)$$

$$t_s = t_{ao} + (t_c - t_{ao}) \times F_1 U \times \left(\frac{1}{U} - R_{si}\right)$$

$$t_s = t_{ao} + (t_c - t_{ao}) \times F_1 \times (1 - U \times R_{si})\,°\mathrm{C}$$

> EXAMPLE 5.2
>
> Calculate the internal surface temperature of a wall that has a thermal transmittance U of 0.96 W/m² K, an internal surface film resistance R_{si} of 0.12 m² K/W, and an outdoor air temperature of −3 °C when the room dry resultant temperature is 21 °C. The overall room factor F_1 is 1.02.

$$t_s = t_{ao} + (t_c - t_{ao}) \times F_1(1 - UR_{si})\,°\mathrm{C}$$

$$= -3 + (21 - -3) \times 1.02 \times (1 - 0.96 \times 0.12)\,°\mathrm{C}$$

$$= 18.7\,°\mathrm{C}$$

> EXAMPLE 5.3
>
> Find the thermal transmittance that is required to avoid surface condensation on the indoor surface of a brick and block cavity wall. A minimum internal surface temperature of 13 °C is to be provided when the outdoor air remains steady at −3 °C. The surface film thermal resistance of the wall is 0.12 m² K/W, the air temperature factor F_1 is 0.97, and the room will be maintained at a resultant temperature of 16 °C.

Using the equation for surface temperature:

$$t_s = t_{ao} + (t_c - t_{ao}) \times F_1 \times (1 - UR_{si})\,°\mathrm{C}$$

Rearrange to find the U value:

$$\frac{t_s - t_{ao}}{t_c - t_{ao}} \times \frac{1}{F_1} = 1 - UR_{si}$$

$$UR_{si} = 1 - \frac{t_s - t_{ao}}{t_c - t_{ao}} \times \frac{1}{F_1}$$

$$U = \frac{1}{R_{si}} \times \left(1 - \frac{t_s - t_{ao}}{t_c - t_{ao}} \times \frac{1}{F_1}\right)$$

Insert the known data and calculate the U value that will satisfy the criteria:

$$U = \frac{1}{0.12} \times \left(1 - \frac{13 - -3}{16 - -3} \times \frac{1}{0.97}\right) \frac{\mathrm{W}}{\mathrm{m^2 K}}$$

$$= 1.1\ \mathrm{W/m^2\,K}$$

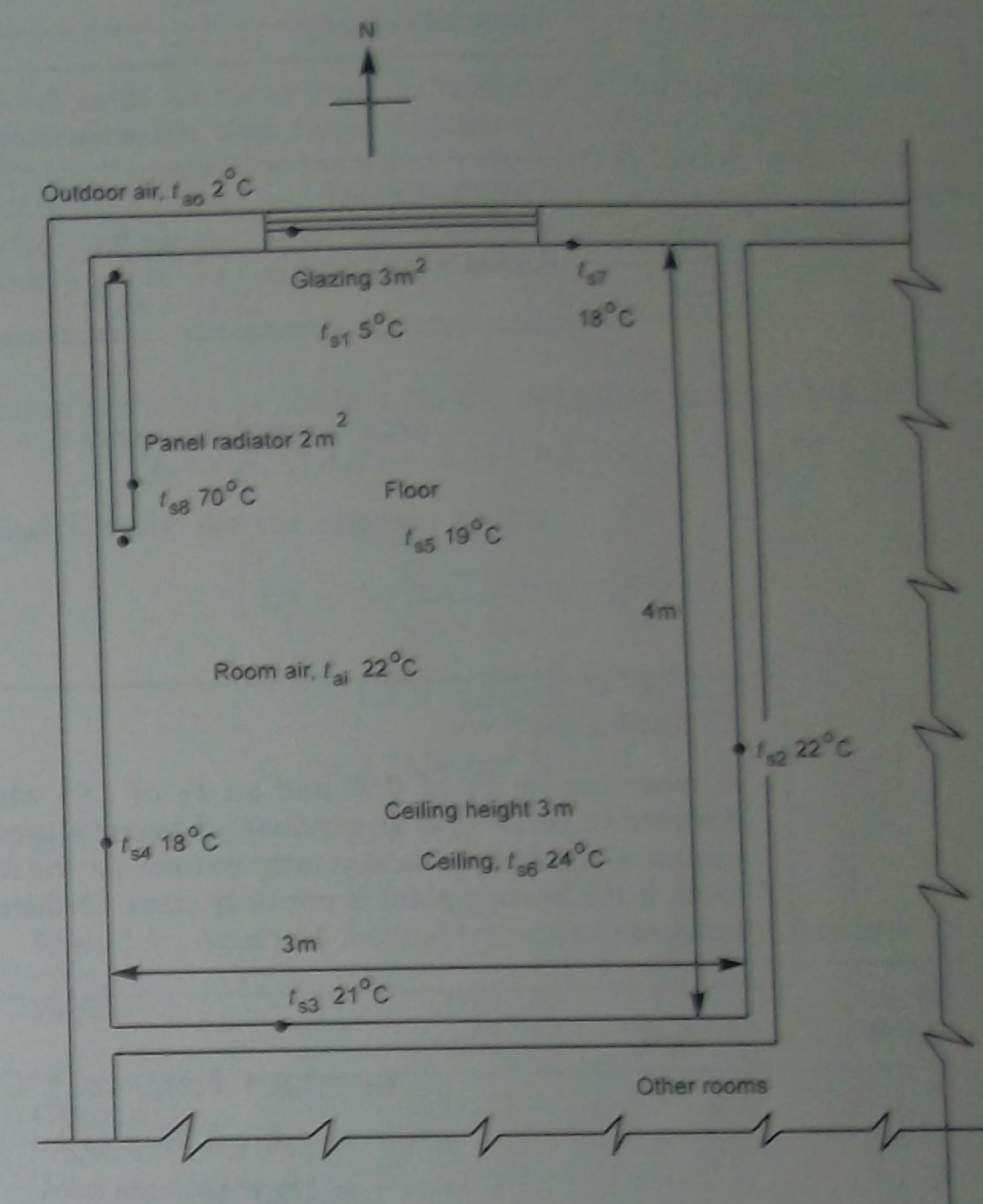


Fig. 5.2 Data for Example 5.4 (mean surface temperature) and Example 5.6 (heat loss).

> EXAMPLE 5.4
>
> Calculate the area weighted mean surface temperature of the room shown in Fig. 5.2.

$$\text{Area weighted mean } t_s = \frac{\Sigma(At_s)}{\Sigma(A)}\,°\mathrm{C}$$

The data are presented in Table 5.2.

Table 5.2 Mean surface temperature data for Example 5.4

Surface	*Area, A* (m²)	t_s (°C)	$(A \times t_s)$ (m² °C)
Glazing surface 1	3	5	15
Wall surface 2	12	22	264
Wall surface 3	9	21	189
Wall surface 4	10	18	180
Floor surface 5	12	19	228

This can be rearranged to find the unknown surface t_s °C:

$$F_1 U\left(\frac{1}{U} - R_{si}\right) = \frac{t_s - t_{ao}}{t_c - t_{ao}}$$

$$t_s - t_{ao} = (t_c - t_{ao}) \times F_1 U \times \left(\frac{1}{U} - R_{si}\right)$$

$$t_s = t_{ao} + (t_c - t_{ao}) \times F_1 U \times \left(\frac{1}{U} - R_{si}\right)$$

$$t_s = t_{ao} + (t_c - t_{ao}) \times F_1 \times (1 - U \times R_{si})\,°\mathrm{C}$$

> EXAMPLE 5.2
>
> Calculate the internal surface temperature of a wall that has a thermal transmi[ttance] U of 0.96 W/m²K, an internal surface film resistance R_{si} of 0.12 m² K/W, a[n] outdoor air temperature of −3 °C when the room dry resultant temperature is 2[1 °C.] The overall room factor F_1 is 1.02.

$$t_s = t_{ao} + (t_c - t_{ao}) \times F_1(1 - UR_{si})\,°\mathrm{C}$$
$$= -3 + (21 - -3) \times 1.02 \times (1 - 0.96 \times 0.12)\,°\mathrm{C}$$
$$= 18.7\,°\mathrm{C}$$

> EXAMPLE 5.3
>
> Find the thermal transmittance that is required to avoid surface condensation o[n the] indoor surface of a brick and block cavity wall. A minimum internal surface te[m]perature of 13 °C is to be provided when the outdoor air remains steady at −3 [°C.] The surface film thermal resistance of the wall is 0.12 m² K/W, the air temper[ature] factor F_1 is 0.97, and the room will be maintained at a resultant temperature of 1[6 °C.]

Using the equation for surface temperature:

$$t_s = t_{ao} + (t_c - t_{ao}) \times F_1 \times (1 - UR_{si})\,°\mathrm{C}$$

Rearrange to find the U value:

$$\frac{t_s - t_{ao}}{t_c - t_{ao}} \times \frac{1}{F_1} = 1 - UR_{si}$$

$$UR_{si} = 1 - \frac{t_s - t_{ao}}{t_c - t_{ao}} \times \frac{1}{F_1}$$

$$U = \frac{1}{R_{si}} \times \left(1 - \frac{t_s - t_{ao}}{t_c - t_{ao}} \times \frac{1}{F_1}\right)$$

Insert the known data and calculate the U value that will satisfy the criteria

$$U = \frac{1}{0.12} \times \left(1 - \frac{13 - -3}{16 - -3} \times \frac{1}{0.97}\right) \frac{\mathrm{W}}{\mathrm{m}^2\mathrm{K}}$$

$= 1.1$ W/m²K

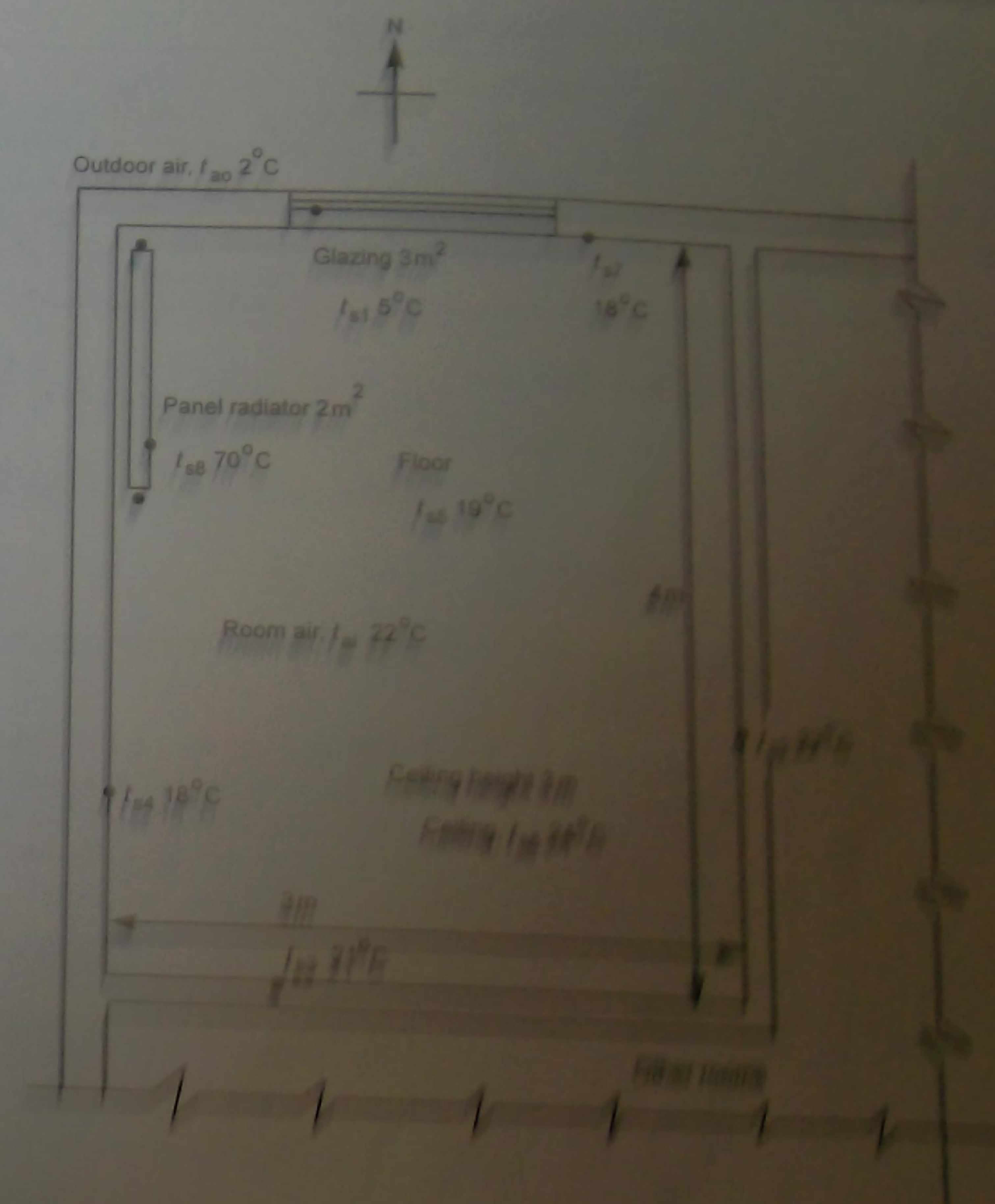


Fig. 5.3 Data for Example 5.4 (mean surface temperature) and Example 5.6 (heat loss).

> EXAMPLE 5.4
>
> Calculate the area weighted mean surface temperature of the room shown in Fig. 5.3.

$$\text{Area weighted mean } t_s = \frac{\Sigma(At_s)}{\Sigma(A)}\,°\mathrm{C}$$

The data are presented in Table 5.3.

Table 5.3 Mean surface temperature data for Example 5.4

Surface	Area, A (m²)	t_s (°C)	$(A \times t_s)$ (m² °C)
Glazing surface 1	3	5	15
Wall surface 2	[illegible]	[illegible]	[illegible]
Wall surface 3	[illegible]	[illegible]	[illegible]
Wall surface 4	[illegible]	[illegible]	[illegible]

Table 5.2 (*contd.*)

Surface	Area, A (m^2)	t_s (°C)	$(A \times t_s)$ (m^2 °C)
Ceiling surface 6	12	24	288
Wall surface 7	6	18	108
Radiator surface 8	2	70	140
	$\Sigma(A) = 66$		$\Sigma(At_s) = 1412$

$$\text{area-weighted mean } t_s = \frac{\Sigma(At_s)}{\Sigma(A)}\,°\text{C}$$
$$= \frac{1412}{66}\,°\text{C}$$
$$= 21.4\,°\text{C}$$

EXAMPLE 5.5

A room has an F_1 of 0.98 and an F_2 of 1.05 when the resultant temperature is designed to be 19 °C at an outdoor air temperature of −4°C. Calculate the environmental, air and mean radiant temperatures for the room that will be produced in the room if the heating plant is correctly sized. Deduce what type of heating system is being used.

$$t_{ei} = t_{ao} + F_1(t_c - t_{ao})\,°\text{C}$$
$$= -4 + 0.98 \times (19 - -4)\,°\text{C}$$
$$= 18.5\,°\text{C}$$
$$t_{ai} = t_{ao} + F_2(t_c - t_{ao})\,°\text{C}$$
$$= -4 + 1.05 \times (19 - -4)\,°\text{C}$$
$$= 20.2\,°\text{C}$$
$$t_r = 2t_c - t_{ai}\,°\text{C}$$
$$= 2 \times 19 - 20.2\,°\text{C}$$
$$= 17.8\,°\text{C}$$

The mean radiant temperature is a little lower than the resultant and environmental temperatures, so a minor proportion of the plant heat emission is by radiation. The room air temperature is greater than the resultant temperature so there is a considerable amount of convective heat output. It is likely that the heat emitter is a panel or column radiator.

EXAMPLE 5.6

Calculate the heating plant output that is required for the office shown in Fig. 5.2. The room is to be maintained at a resultant temperature of 22 °C when the outdoor air is at 2 °C by a ducted air heating system. The rate of infiltration of outdoor air is one air change per hour. There is an identical room below. The adjacent rooms are at a resultant temperature of 18 °C. The thermal transmittances are 5.7 W/m^2 K for the glazing, 0.6 W/m^2 K for the external walls, 0.35 W/m^2 for the flat roof, and 1.2 W/m^2 for the interior walls and floor. Calculate F_u, F_v, F_1, F_2, $\Sigma(AU)/\Sigma(A)$ and $(NV)/[3\Sigma(A)]$. Compare the calculated values for F_1 and F_2 with those in Tables A9.1–A9.7 in the *CIBSE Guide A* (CIBSE, 1986a). Calculate the environmental, air, mean surface and mean radiant temperatures and the plant heat output per m^2 of floor area and per m^3 of volume for the room.

The solution is shown as room 1 on the data disk in the file A:HLOSS.WK1.

$$Q_p = [F_1\Sigma(AU) + 0.33NVF_2] \times (t_c - t_{ao})\ \text{W}$$

The modified U value for the internal wall is

$$U_2 = U_1 \frac{t_{c1} - t_{c2}}{t_{c1} - t_{ao}}\ \text{W/m}^2\,\text{K}$$
$$= 1.2 \times \frac{22 - 18}{22 - 2}\ \text{W/m}^2\,\text{K}$$
$$= 0.24\ \text{W/m}^2\,\text{K}$$

Find $\Sigma(A)$ and $\Sigma(AU)$ from Table 5.3.

Table 5.3 Area and thermal transmittance data for Example 5.6

Surface	Area, A (m^2)	U (W/m^2 K)	$A \times U$ (W/K)
Glazing	3	5.7	17.1
North external wall	6	0.6	3.6
West external wall	12	0.6	7.2
South internal wall	9	0.24	2.16
East internal wall	12	0.24	2.88
Floor (no heat loss)	12	0	0
Flat roof	12	0.35	4.2
	$\Sigma(A) = 66$		$\Sigma(AU) = 37.14$

$$F_u = \frac{18\Sigma(A)}{\Sigma(AU) + 18\Sigma(A)}$$
$$= \frac{18 \times 66}{37.14 + 18 \times 66}$$
$$= 0.9697$$
$$F_v = \frac{6\Sigma(A)}{1012 \times 1.1906Q + 6\Sigma(A)}$$
$$Q = NV/3600\ \text{m}^3/\text{s}$$
$$= 1 \times 4 \times 3 \times 3/3600\ \text{m}^3/\text{s}$$
$$= 0.01\ \text{m}^3/\text{s}$$
$$F_v = \frac{6 \times 66}{1012 \times 1.1906 \times 0.01 + 6 \times 66}$$
$$= 0.9705$$

For a forced warm air heating system, K is 100%, so $K = 1$ and $E = 0$. F_1 and F_2 can be found:

$$F_1 = \frac{E \times (F_v \times 1012 \times 1.1906Q + F_u\Sigma(AU))}{\Sigma(AU) + 18\Sigma(A) \times (KF_v + EF_u)} + F_u$$

$$= \frac{0 \times (0.9705 \times 1012 \times 1.1906 \times 0.01 + 0.9697 \times 37.14)}{37.14 + 18 \times 66 \times (1 \times 0.9705 + 0 \times 0.9697)} + 0.969$$

It can be seen that when the radiant fraction E is zero, the numerator is al zero, and F_1 will always be equal to F_u. So,

$$F_1 = F_u$$

$$= 0.9697$$

$$F_2 = \frac{K \times (F_v \times 1012 \times 1.1906Q + F_u\Sigma(AU))}{1012 \times 1.1906Q + 6\Sigma(A) \times (KF_v + EF_u)} + F_v$$

$$= \frac{1 \times (0.9705 \times 1012 \times 1.1906 \times 0.01 + 0.9697 \times 37.14)}{1012 \times 1.1906 \times 0.01 + 6 \times 66 \times (1 \times 0.9705 + 0 \times 0.9697)} + 0.970$$

$$= \frac{47.708}{396.367} + 0.9705$$

$$= 1.0909$$

In order to read values for F_1 and F_2 from the *CIBSE Guide* A (CIBSE, 198 Table A9.1, it is necessary to calculate the ratios

$$\frac{\Sigma(AU)}{\Sigma(A)} = \frac{37.14}{66}$$

$$= 0.5627$$

$$\text{Room volume } V = 4 \times 3 \times 3\,\text{m}^3$$

$$= 36\,\text{m}^3$$

$$\frac{NV}{3\Sigma(A)} = \frac{1 \times 36}{3 \times 66}$$

$$= 0.1818$$

CIBSE Table A9.1 $F_1 = 0.97$, calculated $F_1 = 0.9697 = 0.97$. CIBSE Table A9 $F_2 = 1.1$, calculated $F_2 = 1.0614 = 1.1$. The calculated values agree with tho from the table to the nearest significant decimal place.

$$Q_p = [F_1\Sigma(AU) + 0.33NVF_2] \times (t_c - t_{ao})\,\text{W}$$

$$= (0.9697 \times 37.14 + 0.33 \times 1 \times 36 \times 1.0909) \times (22 - 2)\,\text{W}$$

$= 979.491$ W, 980 W to nearest significant watt

$$t_{ei} = t_{ao} + F_1(t_c - t_{ao})\,^\circ\text{C}$$

$$= 2 + 0.9697 \times (22 - 2)\,^\circ\text{C}$$

$$= 21.4\,^\circ\text{C}$$

$$t_{ai} = t_{ao} + F_2(t_c - t_{ao})\,^\circ\text{C}$$

$$= 2 + 1.0909 \times (22 - 2)\,^\circ\text{C}$$

$$= 23.8\,^\circ\text{C}$$

$$t_r = 2t_c - t_{ai}\,^\circ\text{C}$$

$$= 2 \times 22 - 23.8\,^\circ\text{C}$$

$$= 20.2\,^\circ\text{C}$$

Now calculate the room surface temperatures from

$$t_s = t_{ao} + (t_c - t_{ao}) \times F_1 \times (1 - UR_{si})\,^\circ\text{C}$$

The R_{si} values are taken from references as

wall and window $R_{si} = 0.12\,\text{m}^2\,\text{K/W}$
floor $R_{si} = 0.14\,\text{m}^2\,\text{K/W}$
ceiling $R_{si} = 0.10\,\text{m}^2\,\text{K/W}$

For this calculation, t_{ao} is the temperature on the other side of the surface, t_{c2} or t_{ao}. The U value is unmodified. The internal surface temperature of the window is found from

$$t_s = 2 + (22 - 2) \times 0.9697 \times (1 - 5.7 \times 0.12)\,^\circ\text{C}$$

$$= 8.1\,^\circ\text{C}$$

The other surface temperatures are found in the same way, and the data are represented in Table 5.4.

Table 5.4 Area and surface temperature data for Example 5.6

Surface	U	t_{ao}	R_{si}	t_s	A	At_s
Glazing	5.7	2	0.12	8.1	3	24.3
North wall	0.6	2	0.12	20.0	6	120.0
West wall	0.6	2	0.12	20.0	12	240.0
South wall	1.2	18	0.12	21.3	9	191.9
East wall	1.2	18	0.12	21.3	12	255.8
Floor	1.3	22	0.14	22.0	12	264.0
Roof	0.35	2	0.10	20.7	12	248.6
$\Sigma(A) = 66\,\text{m}^2$						
$\Sigma(At_s) = 1344.6\,\text{m}^2\,^\circ\text{C}$						

$$\text{Area weighted mean } t_s = \frac{\Sigma(At_s)}{\Sigma(A)}\,^\circ\text{C}$$

$$= \frac{1344.6}{66}\,^\circ\text{C}$$

$$= 20.4\,^\circ\text{C}$$

Notice the environmental criteria for this room. A ducted air heating system is used, and there is no source of radiant heat. An air dry-bulb temperature of 23.8 °C is needed to maintain the design dry resultant temperature of 22 °C. The environmental temperature will be 21.4 °C. The mean radiant temperature will be 20.2 °C, and this is practically the same as the mean of the room surface temperatures, 20.4 °C. The air temperature detector that will be used to control the warm air heating system will need to be set at 24 °C. The detector will need to be sampling an average of the room air condition. This could be achieved when it is located in the room air extract duct.

The plant heat output per unit floor area and room volume are found for comparison with the expected values and with those for other rooms:

$$\frac{Q_p}{A} = \frac{Q_p\ \text{W}}{\text{floor area } A\ \text{m}^2}$$
$$= \frac{980\ \text{W}}{12\ \text{m}^2}$$
$$= 81.7\ \text{W/m}^2$$
$$\frac{Q_p}{V} = \frac{Q_p\ \text{W}}{\text{room volume } V\ \text{m}^3}$$
$$= \frac{980\ \text{W}}{4 \times 3 \times \times 3\ \text{m}^3}$$
$$= 27.2\ \text{W/m}^3$$

EXAMPLE 5.7

Calculate the heating plant output that is required for the industrial building shown in Fig. 5.3. The factory area is to be maintained at a resultant temperature of 16 °C and the offices at a resultant temperature of 20 °C when the outdoor air is at −5°C. A low-temperature hot-water heating system has panel radiators in the factory and office areas. The rate of infiltration of outdoor air, N, is 0.75 air changes per hour in the factory and one air change per hour in the offices. The thermal transmittances are 6 W/m^2 for the glazing and external doors, 0.6 W/m^2 K for the external walls, 0.35 W/m^2 K for the roof, 0.48 W/m^2 K for the floor, and 1.7 W/m^2 K for the interior wall. The designer is to assume that the void above the office flat ceiling is at a temperature of 8 °C. Take the building dimensions from Fig. 5.3. All the windows are 2 m long and 2 m high. Calculate F_u, F_v, F_1, F_2, $\Sigma(AU)/\Sigma(A)$ and $NV/3\Sigma(A)$. Compare the calculated values for F_1 and F_2 with those in Tables A9.1–A9.7 in the *CIBSE Guide A* (CIBSE, 1986a). Calculate the environmental, air, mean surface and mean radiant temperatures and the plant heat output per m^2 of floor area and per m^3 of volume for the room.

Factory area

$$Q_p = [F_1\Sigma(AU) + 0.33NVF_2] \times (t_c - t_{ao})\ \text{W}$$

The modified U value for the internal wall from the factory to the office is

$$U_2 = U_1 \frac{t_{c1} - t_{c2}}{t_{c1} - t_{ao}}\ \text{W/m}^2\,\text{K}$$
$$= 1.7 \times \frac{16 - 20}{16 - -5}\ \text{W/m}^2\,\text{K}$$
$$= -0.32\ \text{W/m}^2\,\text{K}$$

This has a minus sign because there will be a negative flow of heat from the factory area. There will be an equal positive flow of heat from the office into the factory. The modified U value for the internal wall to the roof void above the office is

$$U_2 = 1.7 \times \frac{16 - 8}{16 - -5}\ \text{W/m}^2\,\text{K}$$
$$= 0.65\ \text{W/m}^2\,\text{K}$$

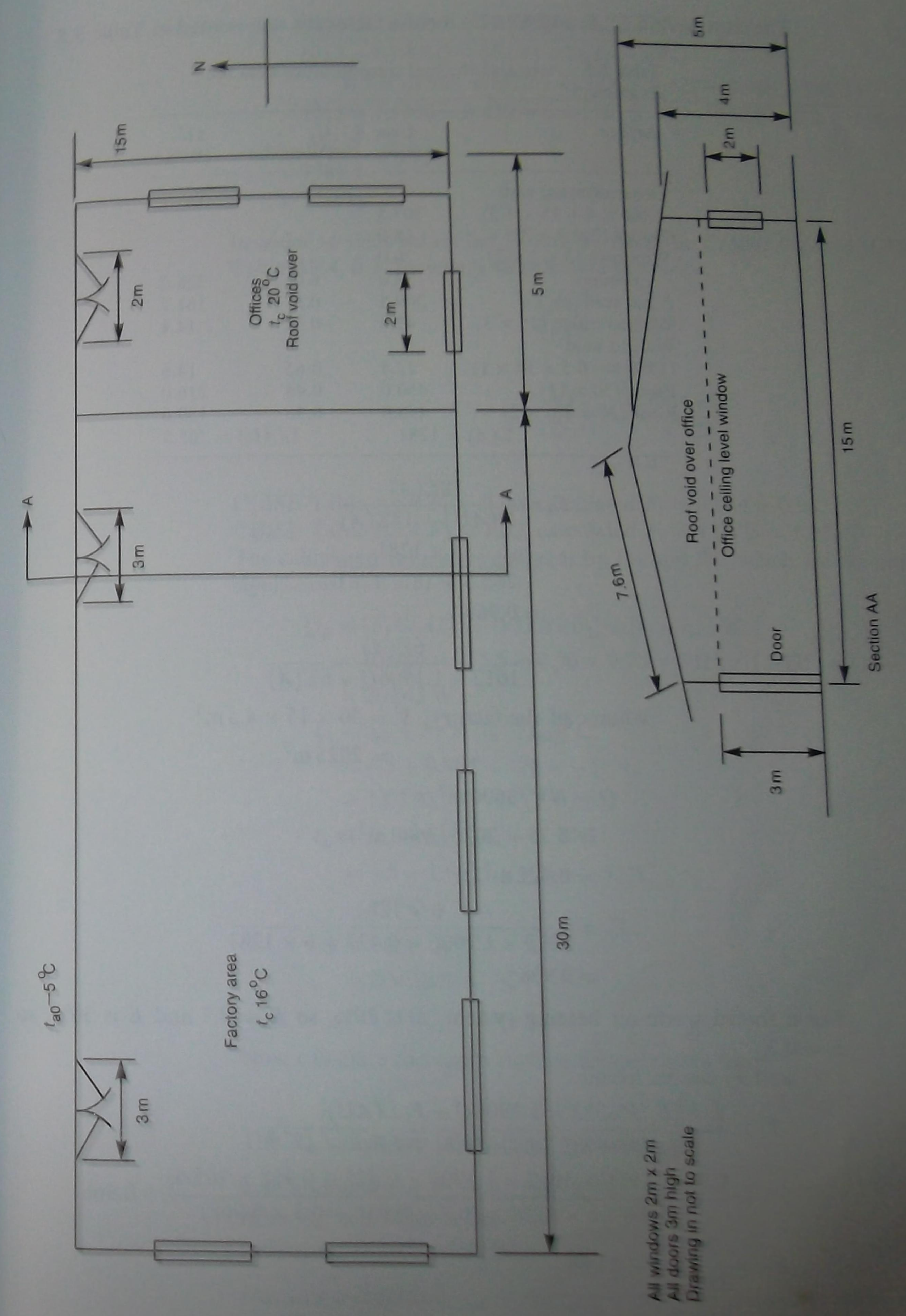


Fig. 5.3 Factory building in Example 5.7.

The data to find $\Sigma(A)$ and $\Sigma(AU)$ for the factory is represented in Table 5.5.

Table 5.5 Area and thermal transmittance data for Example 5.7

Surface	*Area, A* (m^2)	*U* ($W/m^2 K$)	*AU* (W/K)
Gross external wall ($60 \times 4 + 15 \times 4.5$)	307.5		
Doors ($2 \times 3 \times 3$)	18.0		
Windows ($5 \times 2 \times 2$)	20.0		
openings	38.0	6.0	228.0
net wall	269.5	0.6	161.7
Wall to office (15×3)	45.0	−0.32	−14.4
Wall to void ($15 \times 1 + 0.5 \times 15 \times 1$)	22.5	0.65	14.6
Floor (30×15)	450.0	0.48	216.0
Roof ($30 \times 7.6 \times 2$)	456.0	0.35	159.6
	$\Sigma(A) = 1281$		$\Sigma(AU) = 765.5$

$$F_u = \frac{18\Sigma(A)}{\Sigma(AU) + 18\Sigma(A)}$$
$$= \frac{18 \times 1281}{765.5 + 18 \times 1281}$$
$$= 0.968$$

$$F_v = \frac{6\Sigma(A)}{1012 \times 1.1906Q + 6\Sigma(A)}$$

volume of the factory, $V = 30 \times 15 \times 4.5\,m^3$
$$= 2025\,m^3$$

$$Q = NV/3600\,m^3/s$$
$$= 0.75 \times 2025/3600\,m^3/s$$
$$= 0.422\,m^3/s$$

$$F_v = \frac{6 \times 1281}{1012 \times 1.1906 \times 0.422 + 6 \times 1281}$$
$$= 0.938$$

For a forced warm-air heating system, K is 70%, so $K = 0.7$ and E is 30%, $E = 0.3$.

F_1 and F_2 can be found:

$$F_1 = \frac{E \times [F_v \times 1012 \times 1.1906Q + F_u\Sigma(AU)]}{\Sigma(AU) + 18\Sigma(A) \times (KF_v + EF_u)} + F_u$$
$$= \frac{0.3 \times (0.938 \times 1012 \times 1.1906 \times 0.422 + 0.968 \times 765.5)}{765.5 + 18 \times 1281 \times (0.7 \times 0.938 + 0.3 \times 0.968)} + 0.968$$
$$F_1 = \frac{365.4}{22601.4} + 0.968$$
$$= 0.01617 + 0.968$$
$$= 0.984$$

$$F_2 = \frac{K \times [F_v \times 1012 \times 1.1906Q + F_u\Sigma(AU)]}{1012 \times 1.1906Q + 6\Sigma(A) \times (KF_v + EF_u)} + F_v$$
$$= \frac{0.7 \times (0.938 \times 1012 \times 1.1906 \times 0.422 + 0.968 \times 765.5)}{1012 \times 1.1906 \times 0.422 + 6 \times 1281 \times (0.7 \times 0.938 + 0.3 \times 0.968)} + 0.938$$
$$= \frac{852.6}{7787.1} + 0.938$$
$$= 1.0475$$

In order to read values for F_1 and F_2 from the *CIBSE Guide A* (CIBSE, 1986a) Table A9.4, it is necessary to calculate the ratios,

$$\frac{\Sigma(AU)}{\Sigma(A)} = \frac{765.5}{1281}$$
$$= 0.6$$
$$\frac{NV}{3\Sigma(A)} = \frac{0.75 \times 2025}{3 \times 1281}$$
$$= 0.4$$

CIBSE Table A9.4 $F_1 = 0.99$, calculated $F_1 = 0.984 = 0.98$.
CIBSE Table A9.4 $F_2 = 1.02$, calculated $F_2 = 1.0475 = 1.05$.
The calculated values are different by 1% and 3%, which are not considered to be significant.

$$Q_p = [F_1\Sigma(AU) + 0.33NVF_2] \times (t_c - t_{ao})\,W$$
$$= (0.984 \times 765.5 + 0.33 \times 0.75 \times 2025 \times 1.0475) \times (16 - -5)\,W$$
$$= 26\,843\,W$$
$$t_{ei} = t_{ao} + F_1 \times (t_c - t_{ao})\,°C$$
$$= -5 + 0.984 \times (16 - -5)\,°C$$
$$= 15.7\,°C$$
$$t_{ai} = t_{ao} + F_2 \times (t_c - t_{ao})\,°C$$
$$= -5 + 1.0475 \times (16 - -5)\,°C$$
$$= 17\,°C$$
$$t_r = 2t_c - t_{ai}\,°C$$
$$= 2 \times 16 - 17\,°C$$
$$= 15\,°C$$

Now calculate the room surface temperatures from

$$t_s = t_{ao} + (t_c - t_{ao}) \times F_1 \times (1 - UR_{si})\,°C$$

The R_{si} values are taken from references as

wall and window $R_{si} = 0.12\,m^2 K/W$
floor $R_{si} = 0.14\,m^2 K/W$
ceiling $R_{si} = 0.10\,m^2 K/W$

For this calculation, t_{ao} is the temperature on the other side of the surface, t_{ai} or t_{ao}. The U value is unmodified. The internal surface temperature of the window is found from

$$t_s = -5 + (16 - -5) \times 0.984 \times (1 - 6 \times 0.12)\ ^\circ\mathrm{C}$$
$$= 0.8\ ^\circ\mathrm{C}$$

The other surface temperatures are found in the same way, and the data is shown in Table 5.6.

Table 5.6 Surface temperatures for the factory in Example 5.7

Surface	U	t_{ao}	R_{si}	t_s	A	At_s
Glazing/doors	6	−5	0.12	0.8	38	30.4
North wall	0.6	−5	0.12	14.2	102	1448.4
West wall	0.6	−5	0.12	14.2	59.5	844.9
South wall	0.6	−5	0.12	14.2	108	1533.6
Office wall	1.7	20	0.12	16.9	45	760.5
Floor	0.48	−5	0.14	14.3	450	6435
Roof	0.35	−5	0.10	14.9	456	6794.4
Void wall	1.7	8	0.12	14.3	22.5	321.8
$\Sigma(A) = 1281\ \mathrm{m}^2$						
$\Sigma(At_s) = 18168.95\ \mathrm{m}^2\ ^\circ\mathrm{C}$						

$$\text{Area-weighted mean } t_s = \frac{\Sigma(At_s)}{\Sigma(A)}\ ^\circ\mathrm{C}$$
$$= \frac{18\,168.95}{1281}\ ^\circ\mathrm{C}$$
$$= 14.2\ ^\circ\mathrm{C}$$

$$\frac{Q_P}{A} = \frac{Q_P\ \mathrm{W}}{\text{floor area } A\ \mathrm{m}^2}$$
$$= \frac{26\,843\ \mathrm{W}}{450\ \mathrm{m}^2}$$
$$= 59.7\ \mathrm{W/m}^2$$

$$\frac{Q_P}{V} = \frac{Q_P\ \mathrm{W}}{\text{room volume } V\ \mathrm{m}^3}$$
$$= \frac{26\,843\ \mathrm{W}}{2025\ \mathrm{m}^3}$$
$$= 13.3\ \mathrm{W/m}^3$$

Office

The explanations are omitted for this room. For the internal wall from the office to the factory:

$$U_2 = 1.7 \times \frac{20 - 16}{20 - -5}\ \mathrm{W/m^2\,K}$$
$$= 0.27\ \mathrm{W/m^2\,K}$$

The data to find $\Sigma(A)$ and $\Sigma(AU)$ for the office is represented in Table 5.7.

Table 5.7 Area and thermal transmittance data for Example 5.7

Surface	*Area, A* (m^2)	*U* ($\mathrm{W/m^2\,K}$)	*AU* (W/K)
Gross external wall (25 × 3)	75		
Doors (2 × 3)	6		
Windows (3 × 2 × 2)	12		
openings	18	6.0	108
net wall	57	0.6	34.2
Wall to factory (15 × 3)	45	0.27	12.2
Floor (15 × 5)	75	0.48	36
Roof (15 × 5)	75	0.35	26.3
	$\Sigma(A) = 270\ \mathrm{m}^2$		$\Sigma(AU) = 216.7\ \mathrm{W/K}$

$$F_u = \frac{18 \times 270}{216.7 + 18 \times 270}$$
$$= 0.957$$

$$F_v = \frac{6\Sigma(A)}{1012 \times 1.1906Q + 6\Sigma(A)}$$
$$V = 15 \times 5 \times 3\ \mathrm{m}^3$$
$$= 225\ \mathrm{m}^3$$
$$Q = 1 \times 225/3600\ \mathrm{m}^3/\mathrm{s}$$
$$= 0.0625\ \mathrm{m}^3/\mathrm{s}$$

$$F_v = \frac{6 \times 270}{1012 \times 1.1906 \times 0.0625 + 6 \times 270}$$
$$= 0.956$$

$$F_1 = \frac{0.3 \times (0.956 \times 1012 \times 1.1906 \times 0.0625 + 0.957 \times 216.7)}{216.7 + 18 \times 270(0.7 \times 0.956 + 0.3 \times 0.957)} + 0.957$$

$$F_1 = \frac{83.81}{4864.3} + 0.957$$
$$= 0.0172 + 0.957$$
$$= 0.974$$

$$F_2 = \frac{0.7 \times (0.956 \times 1012 \times 1.1906 \times 0.0625 + 0.957 \times 216.7)}{1012 \times 1.1906 \times 0.0625 + 6 \times 270 \times (0.7 \times 0.956 + 0.3 \times 0.957)} + 0.956$$
$$= \frac{195.6}{1624.5} + 0.956$$
$$= 1.076$$

$$\frac{\Sigma(AU)}{\Sigma(A)} = \frac{216.7}{270}$$
$$= 0.8$$

$$\frac{NV}{3\Sigma(A)} = \frac{1 \times 225}{3 \times 270}$$
$$= 0.28$$

CIBSE Table A9.4 $F_1 = 0.99$, calculated $F_1 = 0.974 = 0.97$.
CIBSE Table A9.4 $F_2 = 1.05$, calculated $F_2 = 1.076 = 1.08$.

$$Q_p = (0.974 \times 216.7 + 0.33 \times 1 \times 225 \times 1.076) \times (20 - -5)\ \text{W}$$
$$= 7274\ \text{W}$$
$$t_{ei} = -5 + 0.974 \times (20 - -5)\ ^\circ\text{C}$$
$$= 19.4\ ^\circ\text{C}$$
$$t_{ai} = -5 + 1.076 \times (20 - -5)\ ^\circ\text{C}$$
$$= 21.9\ ^\circ\text{C}$$
$$t_r = 2 \times 20 - 21.9\ ^\circ\text{C}$$
$$= 18.1\ ^\circ\text{C}$$

The surface temperatures are shown in Table 5.8.

Table 5.8 Surface temperatures for the office in Example 5.7.

Surface	U	t_{ao}	R_{si}	t_s	A	At_s
Glazing/door	6	−5	0.12	1.8	18	32.4
External walls	0.6	−5	0.12	17.6	57	1003.2
Internal wall	1.7	16	0.12	19.1	45	859.5
Floor	0.48	−5	0.14	17.7	75	1327.5
Roof	0.35	−5	0.10	18.5	75	1387.5
$\Sigma(A) = 270\ \text{m}^2$						
$\Sigma(At_s) = 4610.1\ \text{m}^2\ ^\circ\text{C}$						

$$\text{Area-weighted mean } t_s = \frac{4610.1}{270}\ ^\circ\text{C}$$
$$= 17.1\ ^\circ\text{C}$$
$$\frac{Q_p}{A} = \frac{7274\ \text{W}}{75\ \text{m}^2}$$
$$= 97\ \text{W/m}^2$$
$$\frac{Q_p}{V} = \frac{7274\ \text{W}}{225\ \text{m}^3}$$
$$= 32.3\ \text{W/m}^3$$

DATA REQUIREMENT

The worksheet has been set to display all numbers in fixed format to two decimal places except where this has been changed in some cells. Zero, one or three decimal places have been used, and these are indicated as F(0), F(1) or F(3) format. The user can enter any number of decimal places into a cell, for example 123.456789, but only the set number of places will be displayed on the screen; 123.4 when the format is F(1). The user can change the display format. The computer uses all the numbers that are typed and entered.

Create new files on drives A and C in subdirectories so that the filename reflects the job title: for example, A:\HOLDEN\HLOSS\LOSS1.WK1. This means that

there is a job data disk in drive A. This disk has a directory named HOLDEN. All the files for the job for Holden Motor Company will be located in the HOLDEN directory. A subdirectory of the HOLDEN main directory is reserved for heat loss calculation files; this is the HLOSS area of the disk. A worksheet named LOSS1.WK1 is to be used for the current data and calculations.

Each room can consist of two rectangular spaces. Where the room cannot be simply described in this manner, it may be necessary to divide it into more than one space. The parts of the room can be given names such as LoungeA1, Loun- geA2, etc., so that they are correctly added when it is time to find the total heat load. Each room can have four walls. Each wall can have three openings for windows or doors. Where a heated space has more than four walls and three openings in each wall, it may be appropriate to add some of the lengths together so that all the components can be included. Floor, ceiling and roof data can be entered under any of these surface categories provided that the correct data is identified and entered somewhere in the cell range from B95 to F100.

The temperature on the other side of each wall, floor, ceiling and opening is entered by the user. This will be either the outdoor air temperature or the resultant temperature of the room or space on that side. The thermal transmittance of each wall and opening is entered as the unmodified U value. These will be taken from the references or the UVAL.WK1 worksheet output by the user. Modifications for intermediate temperature differences are carried out in the cell formulae. Make sure that zero is present, or is typed, in cells for the dimensions of walls and openings, the second temperature and the thermal transmittance where there are no surfaces to be calculated for. This also ensures that previous applications that have used this copy of the file have been overwritten. If cells contain old data that is correct for the new application, there is no need to overwrite it.

Check that the dimensions of the rooms have been accurately entered, because the only formula that will validate the input data is a simple area check. Visual, measurement or typing errors by the user may not be found by this check. Make sure that you have entered only numbers into the cells that require numeric data. If the letter 'o' or 'O' is typed instead of zero '0', an error message will be displayed in the cell where the calculation is produced. Alphabetic characters in cells always have an apostrophe (') at the beginning of the string in the formula line on screen. Spreadsheet programs do not calculate with alphabetic characters, only numbers. You may think that a cell contains numbers because they appear in the cell on the screen, but you may have typed them in as words and then found an error message in the formula cell. This is an easy mistake to make. Review all the data that is on the screen for each room to double check that there have not been any omissions. Look at the dimensions, temperatures and thermal transmittances. Are they correct?

The worksheet is set up for 10 rooms. Each room must have valid data to avoid the formulae generating an error, ERR, message in cells. When all the real rooms have been described, for example seven of them, the remaining three rooms are give fictitious data and they become fictitious rooms. The original data disk contains these non-existent rooms with appropriate data. Each fictitious room must be 1 m long, 1 m wide and 1 m high. It must have a design resultant temperature of 1 °C and an outdoor air temperature of 0 °C. Wall 1 must be 1 m long, 1 m high and have a thermal transmittance of zero W/m² K. This is the only data that is needed for each fictitious room. The worksheet calculates the 1 m² loss component. This is then multiplied by the difference between the possible ro[om] air temperature and the outdoor air temperature. A negative loss of heat sh[ows] that the outdoor air is warmer than the current value of the room air tempera[ture].

$$\text{heat loss} = [\Sigma(AU) + 0.33NV] \times (t_a - t_{ao})\ \text{W}$$
$$\text{D130} = (\text{G118} + 0.33 \times \text{C30} \times \text{D39}) \times (\text{B130} - \text{C29})\ \text{W}$$

Cell E130

(F0) +C130–D130

The difference between the heat gain to and the heat loss from the unheated roo[m] is found. The user can see which room air temperature satisfies both heat flow[s]; this is also known as the balance temperature.

$$\text{balance heat flow} = \text{gain} - \text{loss W}$$
$$\text{E130} = \text{C130} - \text{D130 W}$$

> EXAMPLE 5.16
>
> Calculate the temperature that is likely to be established in the unheated store of the factory building shown in Fig. 5.4.

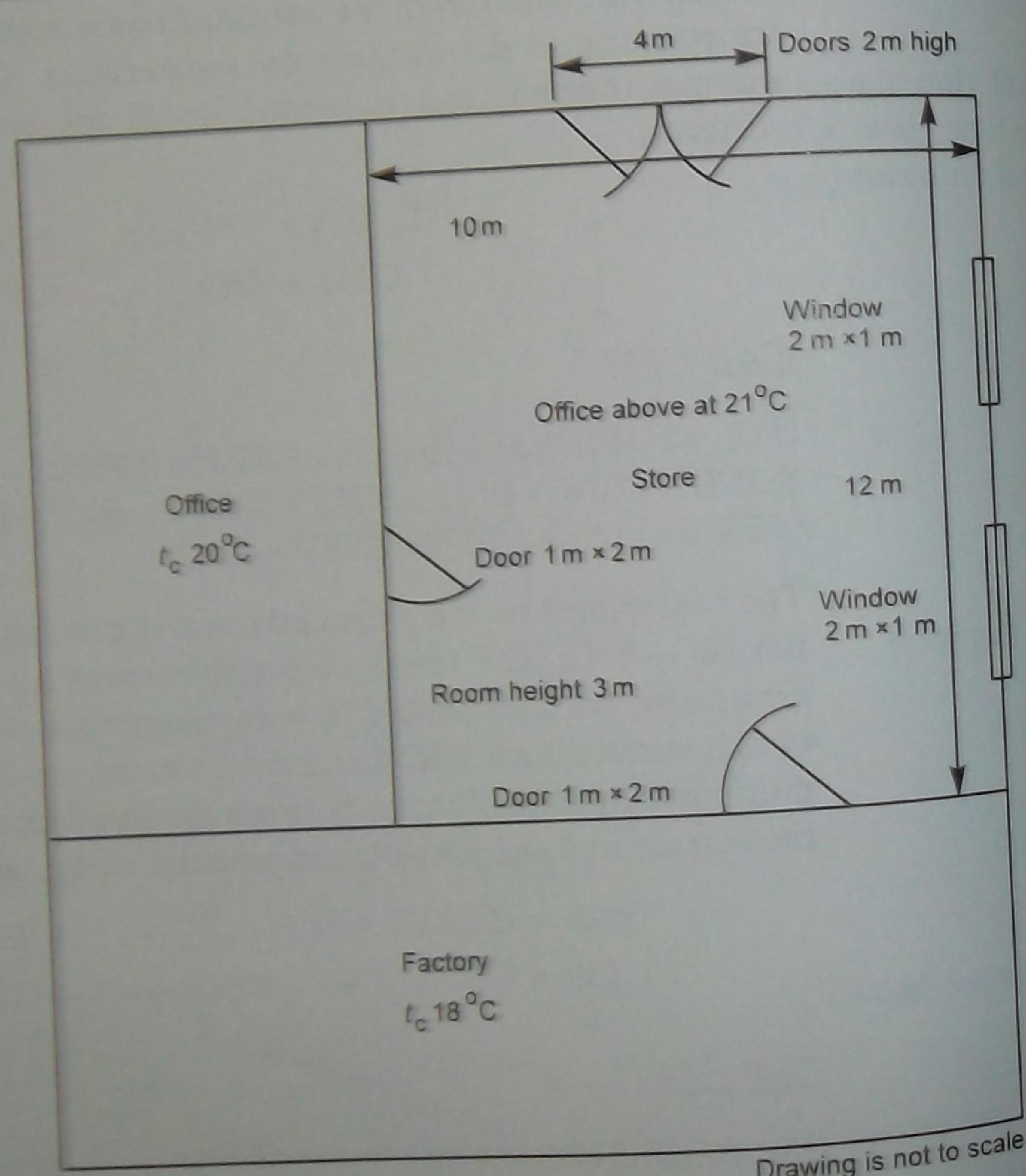


Fig. 5.4 Unheated store room in Example 5.16.

Use the dimensions and data on the figure and the following information: outdoor air temperature for design −2 °C; the store is expected to have 0.75 air changes per hour due to the infiltration of outdoor air; the office above the store is maintained at a resultant temperature of 21 °C; the internal doors have a thermal transmittance of 5.3 W/m² K; the store double door and the windows have a thermal transmittance of 5.3 W/m² K; the internal wall thermal transmittance is 1.7 W/m² K; the store ceiling has a thermal transmittance of 1.3 W/m² K; the thermal transmittance of the external wall is 0.7 W/m² K; the thermal transmittance of the floor is 0.8 W/m² K.

The solution is shown on the data disk as file A:UNHEAT.WK1.

Using the worksheet reveals that the store is likely to be at an air temperature of 8 °C. Calculate the heat gains and losses for an assumed temperature of 8 °C for the store. The data is shown in Table 5.9. An example of a heat gain and a heat loss calculation is detailed.

$$\text{factory door heat gain} = 2\,\text{m}^2 \times 3\,\text{W/m}^2\,\text{K} \times (18 - 8)\,^\circ\text{C}$$
$$= 60\ \text{W}$$

$$\text{store external wall net area} = [(22 \times 3) - 8 - 4]\,\text{m}^2$$
$$= 54\,\text{m}^2$$

$$\text{store wall heat loss} = 54\,\text{m}^2 \times 0.7\,\text{W/m}^2\,\text{K} \times (8 - -2)\,\text{K}$$
$$= 378\ \text{W}$$

Table 5.9 Data for the unheated store in Example 5.16.

Surface	*L* (m)	*H* (m)	*Area* m²	*U* (W/m² K)	t_{c1} (°C)	Q_p (W)
Heat gains			36			
Wall	12	3	2	3	20	72
Door	1	2	34	1.7	20	693.6
net			30			
Wall	10	3	2	3	18	60
Door	1	2	28	1.7	18	476
net			120	1.3	21	2028
Ceiling	12	10				
				Total heat gain = 3330		
Heat losses			66			
Wall	22	3	8	5.3	8	424
Door	4	2	4	5.3	8	212
Windows	4	1	54	0.7	8	378
net			120	0.8	8	960
Floor	12	10				
			Total conduction heat loss = 1974			

$$\text{ventilation } Q_p = 0.33NV \times (t_a - t_{ao})\ \text{W}$$
$$= 0.33 \times 0.75 \times 12 \times 10 \times 3 \times (8 - -2)\ \text{W}$$
$$= 891\ \text{W}$$

$$\text{total heat loss } Q_p = 1974 + 891\ \text{W}$$
$$= 2865\ \text{W}$$

The difference between the heat gain and heat loss is

$$\text{heat gain} - \text{heat loss} = 3330 - 2865\ \text{W}$$
$$= 465\ \text{W}$$

It can be seen from the worksheet that this is the lowest positive difference. The store is likely to stabilize at around 8 °C provided that there are no other sources of heat, such as working people, solar radiation, lighting or electrically powered equipment. It is also assumed that the doors and windows remain shut.

Questions

Questions 1–16 require the use of a written response and some manual calculations, not the use of the work- sheet When the user understands each part of the cal- culation procedure and has manually evaluated all the data at least once, further cases should be entered onto the worksheet.

1. Explain the basis for the calculation of the heating plant power needed to provide for the thermal comfort of the occupants of a building. Include in your explanation the factors that are taken into consideration, the physical conditions of the building, the type of heating system, the climate design data used and the method of calculation.
2. Discuss the difference between steady-state and transient heat loss calculation data and methods. Give examples of building types that may require these different approaches. Explain how the heat- ing system is selected in relation to the steady- state and transient heat flows from a building.
3. Explain why resultant temperature is used to spe- cify the thermal comfort condition for an occupied room. Include in your explanation the heat trans- fer methods that are involved and how they are included in the calculation of heat loss.
4. Calculate the volumetric specific heat capacity for air when it is at a density of 1.185 kg/m^3 and its spe- cific heat capacity is 1008 J/kg K.
5. Explain the reason for, and the methodology of, the use of a modified thermal transmittance for a wall, floor or ceiling that separates two areas of the building that are at different temperatures from that of the outdoor air. Compare this approach with an alternative method of finding the heat flow through such a surface. State which method is preferable and why this is the case.
6. Calculate the thermal transmittance U_1, the modi- fied thermal transmittance U_2 and the heat flow Q_p through an internal wall of 13 m^2 that separates a lounge from an unheated garage in a house. The lounge is maintained at a resultant temperature of 21 °C. The garage air temperature is expected to stabilize at 8 °C when the outdoor air tempera- ture is -1 °C. The wall is constructed from 125 mm thick medium-density concrete blockwork with 15 mm plaster on the lounge side only. The thermal conductivity of the blockwork is 0.9 W/m K and that of the plaster is 0.5 W/m K. The surface film thermal resistance is 0.12 m^2 K/W. The air tem- perature factor, F_1, is 0.94.
7. Calculate the thermal transmittance, the modified thermal transmittance and the heat flow through a wall between two heated rooms. The wall is 10 m long and 3.5 m high, and is constructed from 150 mm thick lightweight concrete blockwork. The wall is plastered on both sides with 12 mm thickness of lightweight plaster. The rooms are maintained at resultant temperatures of 20 °C and 14 °C when the outdoor air is -6 °C. The thermal conductivity of the blockwork is 0.19 W/m K, and that of the plaster is 0.48 W/m K. The surface film thermal resistance is 0.12 m^2 K/W, and the air tem- perature factor F_1 is 0.89.
8. Calculate the thermal transmittance and the mod- ified thermal transmittance for an intermediate floor of a bedroom that has a lounge directly below it. The floor consists of a 20 mm thickness of carpet and underlay, 25 mm floorboards, 200 mm air space and 25 mm of plasterboard and plaster. Use the data provided to find the heat loss from the lounge and the heat gained by the bedroom. Conduct these two heat flow calculations sepa- rately. Data: resultant temperatures, lounge 21 °C, bedroom 16 °C; R_{si} ceiling, heat flow upwards 0.1 m^2 K/W; R_{si} floor, heat flow upwards 0.14 m^2 K/ W; R_a floor air space 0.18 m^2 K/W; outdoor air tem- perature 3°C; thermal conductivity of carpet 0.14 W/m K; thermal conductivity of timber 0.5 W/ m K; thermal conductivity of plasterboard 0.5 W/ m K; lounge ceiling is 7 m × 4 m; air temperature factor F_1 is 0.97.
9. An unheated store room is expected to be at a resultant temperature of 8 °C when the outdoor air temperature is at 1 °C. One wall of the store is adjacent to an office that is maintained at a resul- tant temperature of 20 °C. The wall is 8 m long, 2.8 m high and is constructed of 100 mm thick med- ium-density concrete blockwork with 12 mm dense plaster on both sides. The air temperature factor F_1 is 0.9. The surface film thermal resistance is 0.12 m^2 K/W. The thermal conductivity of the blockwork is 0.7 W/m K, and that of the plaster is 0.8 W/m K. Calculate the heat loss that is expected to flow from the office into the store room.
10. Calculate the surface temperature of an external wall that has a thermal transmittance of 0.72 W/m^{2}K when the indoor resultant temperature is 19 °C and the outdoor air is at -3 °C. The indoor sur- face film thermal resistance is 0.12 m^2 K/W and the air temperature factor F_1 is 0.93.
11. Calculate the minimum value that must be created for the thermal transmittance of an external peri- meter construction if the indoor surface tempera- ture is not to fall below 5 °C when the outdoor air temperature remains constant at -4 °C. The sur- face film thermal resistance of the walling is 0.12 m^2 K/W, the air temperature factor F_1 is 0.95, and the room will be maintained at a resultant temperature of 16 °C. State the type of external perimeter construction that would satisfy this requirement.
12. Find the thermal transmittance necessary to avoid surface condensation on a wall that sepa- rates a room that is maintained at a resultant tem- perature of 18 °C from an unheated room which is expected to fall to 4 °C during winter. The room dew point is 11 °C and the air temperature factor F_1 is 0.88. The internal surface film thermal resis- tance of the wall is 0.12 m^2 K/W.
13. Calculate the area-weighted mean surface tem- perature of a factory that has an overhead radiant strip heating system. A survey revealed the following areas and surface temperatures: floor 200 m^2, 11 °C; roof 230 m^2, 13 °C; walls 240 m^2, 12 °C; radiant strip 20 m^2, 110 °C.
14. A room has an F_1 of 0.96 and an F_2 of 1.03 when the resultant temperature is designed to be 20 °C at an outdoor air temperature of -2 °C. Calculate the environmental, air and mean radiant tempera- tures that will be produced in the room if the heat- ing plant is correctly sized. Deduce what type of heating system is being used.
15. A factory has a 100% convective heating system, where F_1 is 0.92 and F_2 is 1.23. The resultant tem- perature is designed to be 16 °C at an outdoor air temperature of 5 °C. Calculate the environmental, air and mean radiant temperatures that will be produced in the room if the heating plant is cor- rectly sized.
16. An industrial building has a 90% radiant heating system, where F_1 is 1.05 and F_2 is 0.86. The resultant temperature is designed to be 14 °C at an outdoor air temperature of 1 °C. Calculate the environ- mental, air and mean radiant temperatures that will be produced in the room if the heating plant is correctly sized.
17. Calculate the heating plant output that is required for a hotel dining room from the data provided: room dimensions 20 m × 10 m and 3 m high; design resultant temperature 21 °C; outdoor air temperature -3 °C; infiltration of outdoor air 1.5 changes per hour; ducted air heating system; one long and one short wall adjoin another room that is at a resultant temperature of 18 °C; three exter-

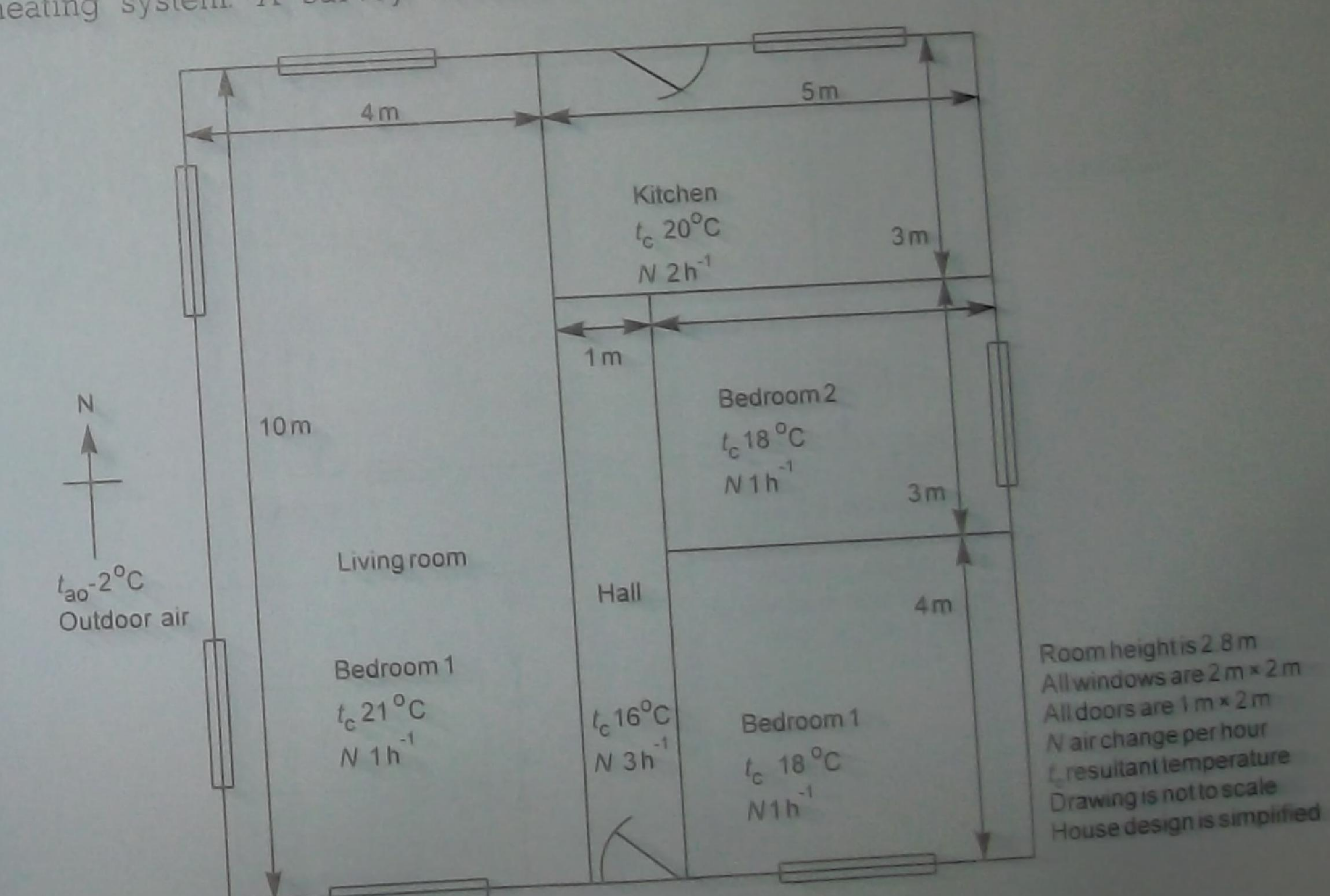


Fig. 5.5 Plan of house in question 18.

and roof space $0.25\ W/m^2K$, sloping tiled roof $4.6\ W/m^2K$, vertical walls at ends of roof space $1.2\ W/m^2K$.

(a) Calculate the air temperature that is expected to be found within the roof space.
(b) Experiment with different combinations of outdoor air temperature and infiltration rate, to discover the relationship between the roof space and outdoor air temperatures.
(c) Relocation of the roof space thermal insulation from the flat ceiling to the underside of the tiled roof causes the flat ceiling thermal transmittance to become $3.6\ W/m^2K$ and that of the sloping roof to be $0.25\ W/m^2K$. Assess the effects of this change.
(d) Comment on the risk of freezing that is experienced by the water services within the roof space, and the measures that are necessary to reduce the risk.

I was told

6 Fan and system selection

INTRODUCTION

The characteristic curves for the performance of fans are used in graphical form by the design engineer. Students have to learn to recognize the shape of the curve for each type of fan, and be able to manipulate data in graphs of pressure against flow. This chapter makes use of fan graphs, and shows how the pressure rise and flow data from manufacturers' catalogues can be entered into a computer and used by the design engineer.

The selection of a suitable fan is made by plotting the ductwork system design flow rate and its index route pressure drop onto the fan performance curve. This can now be done on a worksheet, and the resulting graph viewed and interrogated. A graphical enlargement of the crossover region between the ductwork system and fan curves allows the designer to make any tolerance adjustments that are desired. Fan speed changes can be easily made and the results displayed numerically and graphically.

Seven independent spreadsheets are presented. They are provided for five specific fan types – forward-curved centrifugal, backward-curved centrifugal, axial flow, propeller and mixed flow, a user-defined fan – and a comparison of fan types. In the user-defined fan spreadsheet, the user types pressure and flow data from the manufacturer's literature into the columns. The spreadsheet has been programmed to plot the fan curves and calculate the performance at other speeds. The seventh worksheet is to demonstrate the common curves for five different types of fan.

LEARNING OBJECTIVES

Study and use of this chapter will enable the user to:

1. recognize the shapes of the performance curves of fans;
2. know the types of fan in use;
3. know representative formulae for fan characteristic curves;
4. understand and use fan performance curves for different fan rotational speeds;
5. know the range of fan speeds used;
6. understand the meaning and use of polynomial equations;
7. use polynomial data in the creation of graphs;
8. use fan performance curves for centrifugal, axial, mixed flow and propeller fans;

9. know how to generate a ductwork system resistance curve;
10. relate the ductwork system resistance curve to fan performance curves;
11. know how to include the commissioning tolerance in the air flow design of ductwork systems;
12. select the intersection point for fan and ductwork system operation;
13. draw an enlargement of the fan and system intersection;
14. enter data for any fan type;
15. understand the gearing effect of fan and motor pulley diameters;
16. know the terms used in fan engineering;
17. be able to convert measurement units;
18. calculate the electrical input power for different fans.

Key terms and concepts

FAN CHARACTERISTIC CURVES

The performance of a fan is specified by the pressure rise that it generates in the air volume flow rate that is passing through it. Typical curves for axial, mixed, propeller, forward-curved centrifugal and backward-curved centrifugal fans are shown in Fig. 6.1.

The data for this figure can be seen in the worksheet file COMPARE.WK1. The shapes can be viewed on graph 1. Fan curves can be reproduced mathematically once the type is known and the test data is published. Manufacturers predict the pressure output of a new size of fan from a knowledge of the performance of geometrically similar models that have been tested. Curves of pressure against volume flow are scaled upwards and downwards depending upon the rotational

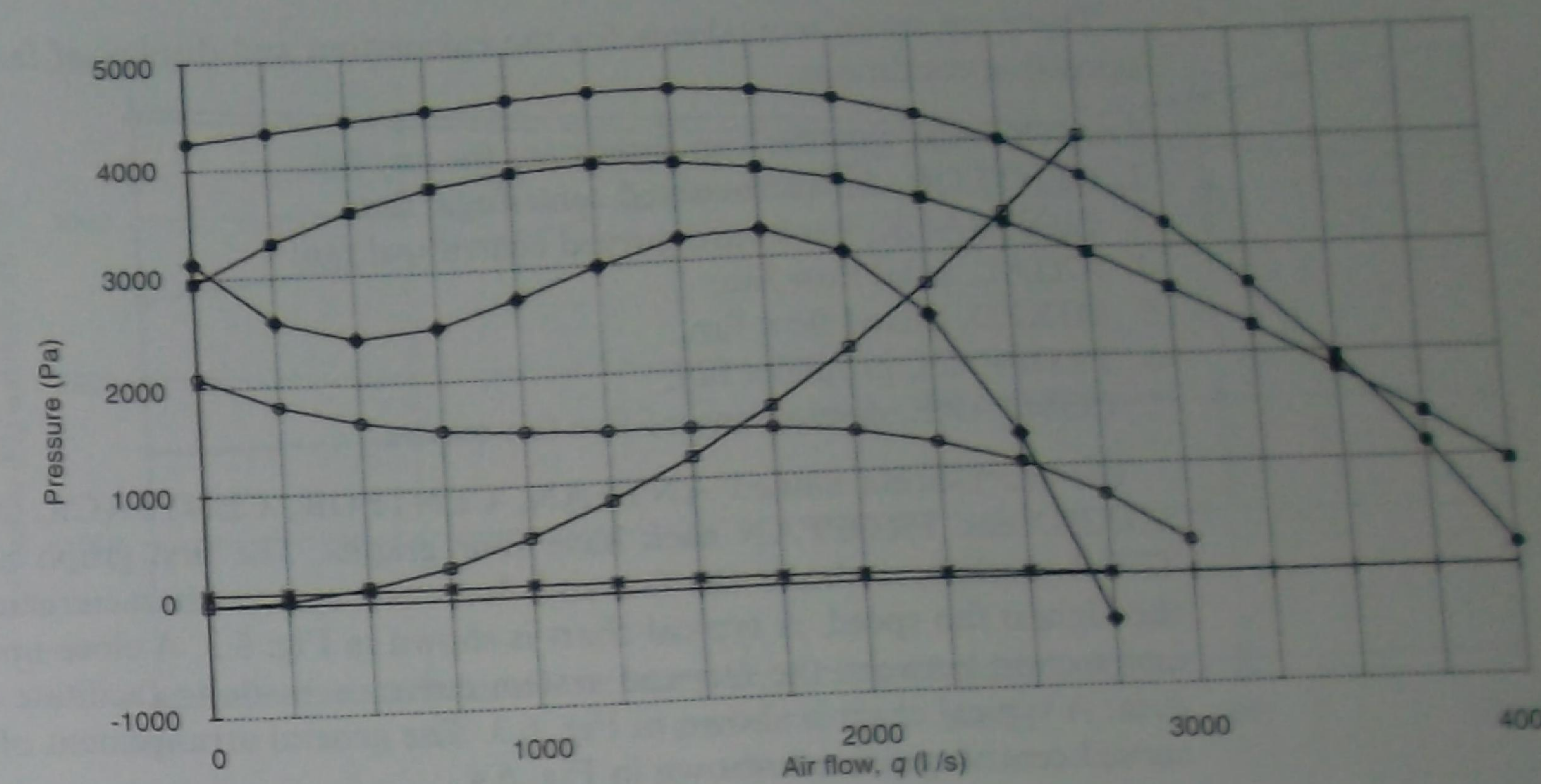


Fig. 6.1 Comparison of fan types. (fans are not directly comparable): ●, backward-curved centrifugal; ■, forward-curved centrifugal; ◆, axial; *, propeller; ○, mixed flow; □, system resistance.

speed at which the new fan is to operate. New fan models are subsequently tested to confirm their predicted performance.

The approach used here has been to read the fan total pressure and volume data from manufacturers' published curves and fit an equation to the curve. The curves of fan static pressure, fan velocity pressure and power consumption are calculated for the range of flow rates employed. All these curves can be automatically scaled up or down when the fan data is required at a different rotational speed.

Manufacturers' data may be fan static pressure against volume flow rate. The minor editing task that is necessary applies only when entering data into the ANYFAN sheet, and is demonstrated in question 3.

The curve of fan total pressure, *FTP* Pa, against air volume flow rate, *Q* l/s, usually corresponds to a four-term polynomial equation. These are of the form

$$FTP = a \pm bQ \pm cQ^2 \pm dQ^3$$

where *a*, *b*, *c* and *d* are constants for the particular type and size of fan, *FTP* is the fan total pressure rise in Pa, and *Q* is the air volume flow rate delivered by the fan in l/s. It may be necessary to add the fifth and higher terms, but four are expected to give a sufficiently good curve fit. The first term includes *Q* raised to the power of zero. As anything raised to zero is unity, the constant appears on its own. The second term is *Q* raised to the power of unity, the third is the power 2 and the fourth is the power 3. The second and higher terms can be positive or negative. Notice:

$$FTP = aQ^0 \pm bQ^1 \pm cQ^2 \pm dQ^3$$

as $Q^0 = 1$ and $Q^1 = Q$, so

$$FTP = a \pm bQ \pm cQ^2 \pm dQ^3$$

There are seven worksheets for the calculation and display of fan and system operating conditions:

1. ANYFAN, the user can enter data for any fan;
2. CENTFOR, forward-curved centrifugal fan;
3. CENTBACK, backward-curved centrifugal fan;
4. AXIAL, axial flow fan;
5. MIXED, mixed flow fan.
6. PROPFAN, propeller fan;
7. COMPARE, display any of five fan curves.

The worksheets named ANYFAN, CENTFOR, CENTBACK, AXIAL and MIXED and PROPFAN each have four graphs. The first graph in the CENTFOR worksheet shows the fan and ductwork system characteristic curves for the highest fan speed. A typical chart is shown in Fig. 6.2. A close-up view of the intersection between the fan and system curves is made to facilitate reading the data. A typical chart is shown in Fig. 6.3. The general arrangement of a forward-curved centrifugal fan is shown in Fig. 6.4.

ANYFAN

This file is for a user-defined fan characteristic. Air flow and pressure data can be entered for axial flow, centrifugal, propeller or mixed flow fans. Four graphs have been prepared using the sample fan data on the worksheet. The graphs are in pairs. Graph 1 is for the first fan, and it shows the whole range of performance at the highest speed. This is expected to be no more than 2900 rev/min, the highest practical motor rotational speed on a 50 Hz electrical power supply. 50 Hz corresponds to 3000 rev/min. A 60 Hz supply corresponds to 3600 rev/min. There is

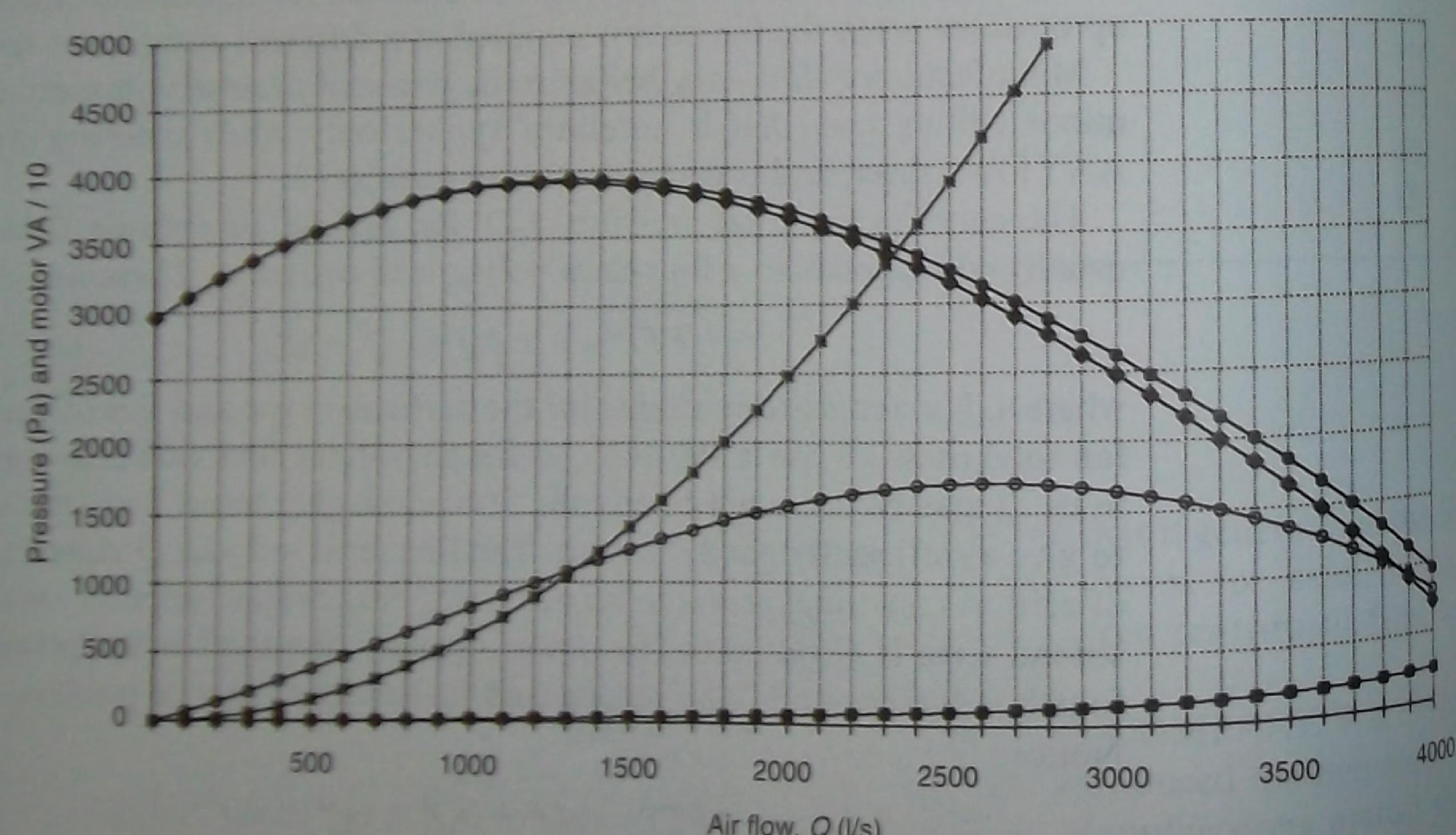


Fig. 6.2 Forward-curved centrifugal fan, highest speed: ●, fan total; ■, fan velocity; ◆, fan static; ∗, system resistance; ○, motor VA/10.

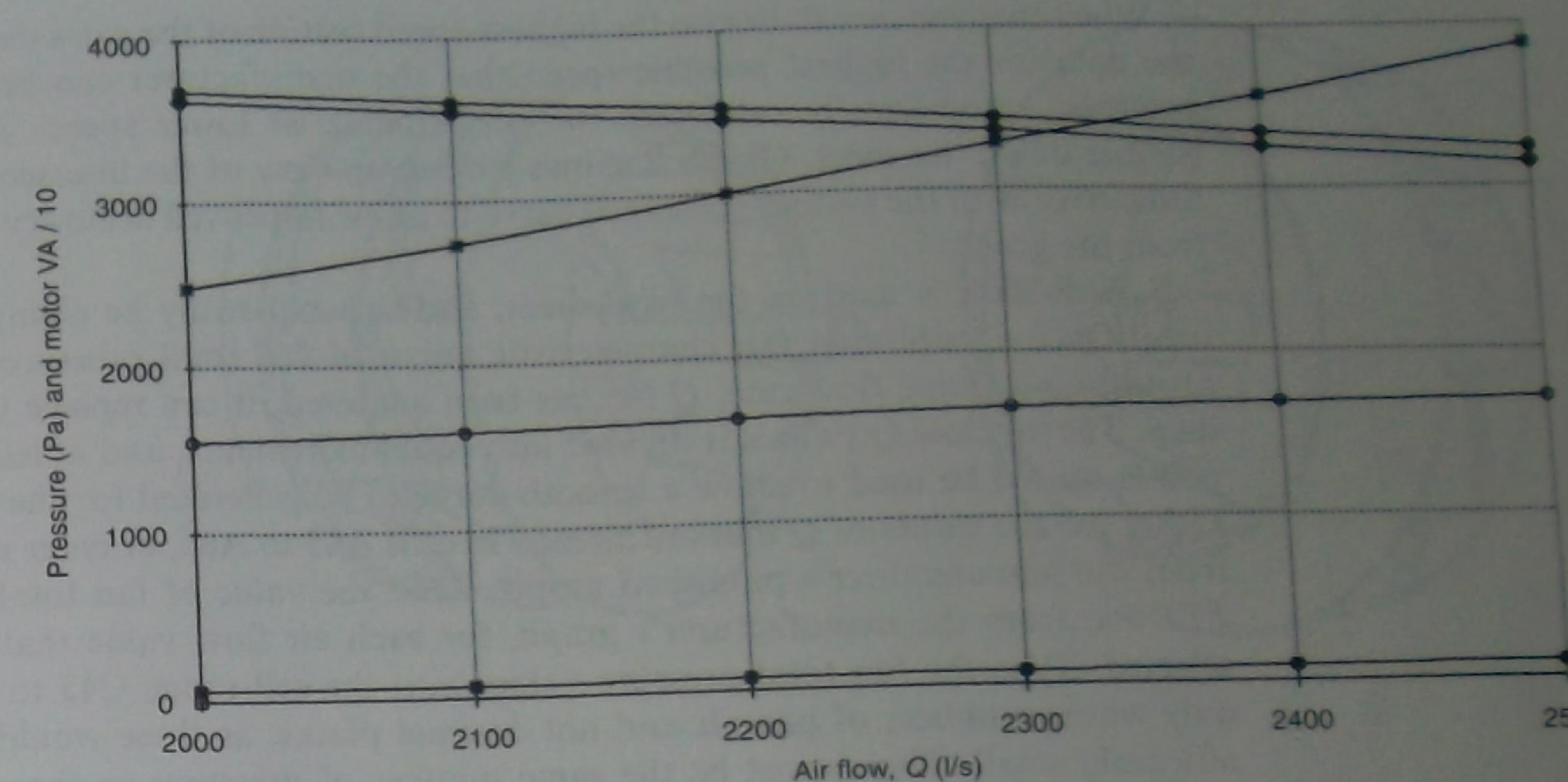


Fig. 6.3 Intersection of fan and system curves for a forward-curved centrifugal fan, highest speed: ●, fan total; ■, fan velocity; ◆, fan static; ∗, system resistance; ○, motor VA/10.

always a slip between the alternating current frequency in the stator of the motor and the rotor shaft speed. Fans are generally driven with a V-belt drive running on pulleys. There is a larger pulley on the fan shaft than on the motor shaft, so that there is a reduction gearing effect. Fan rotational speeds are usually within the range 350–1450 rev/min in order to minimize the generation of noise. Some manufacturers show curves for fan speeds up to 4000 rev/min. These have a smaller diameter pulley on the fan shaft than on the motor shaft in order to raise the gear ratio. Up to 100 dB may be produced in the outlet duct at such speeds, and attenuators may be required.

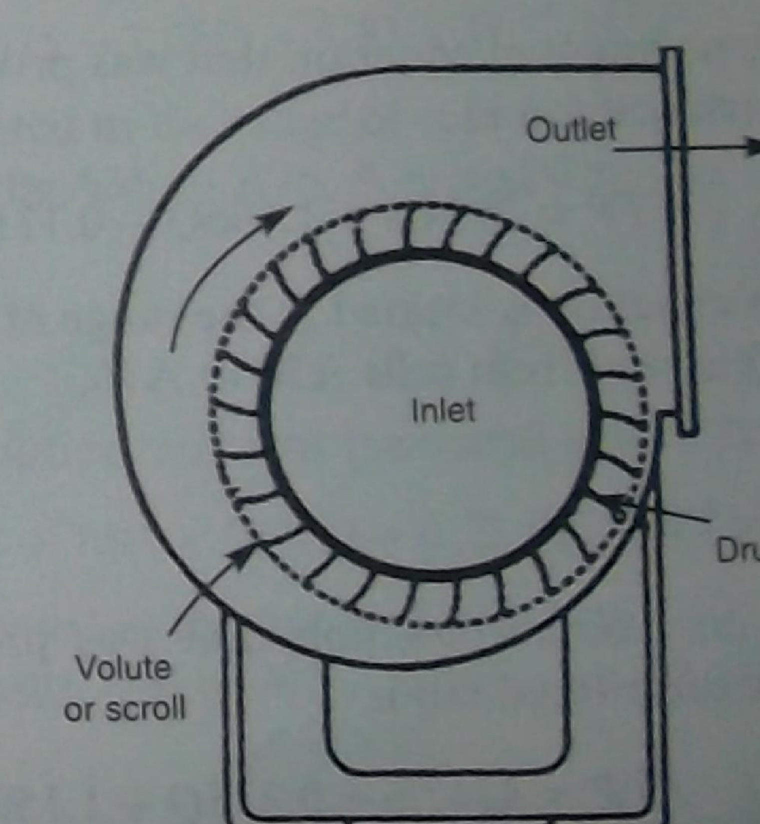


Fig. 6.4 Forward-curved centrifugal fan: general arrangement (reproduced by permission of Woods of Colchester Ltd).

When entering new data into the highest-speed section of the worksheet, acquire the data for the highest possible speed that the manufacturer can provide. The program automatically calculates the performance at lower speeds in the cells further down the sheet. Graph 2 shows a close-up view of the intersection of the fan curves with the system resistance curve to allow improved accuracy of reading from the graph.

Sample data is used on the worksheet, and this can easily be changed by the user. Once a published fan characteristic curve of fan total pressure, *FTP* Pa, against air volume flow rate, Q l/s, has been acquired, it can replace the sample data. The air flow Q l/s axis is divided into equal increments, and at least 10 data points should be used to allow a smooth curve to be generated for the new data. Either use the values of Q that are already in cells A45 to A85, or enter new values from the manufacturer's published graph. Read the value of fan total pressure, *FTP* Pa, from the manufacturer's graph, for each air flow value that has been selected. Type the fan total pressure values into the cell range C45 to C85. Use only whole numbers of pascals and not decimal places, as these would be insignificantly small. There must be the same number of pressures as there are flow rates. If a different number of data points are used, in comparison with the original cell range A45 to A85, then the ranges of data that have been specified in the graph files will have to be edited. If the published fan total pressure is in millimetres H_2O, multiply it by the gravitational acceleration, g m/s^2, to convert it to pascals. That is:

$$\text{pressure, } p = gH$$

where p = pressure (Pa);
g = gravitational acceleration, 9.907 m/s^2; and
H = manometer head (mm H_2O).

$$p\,\text{Pa} = 9.907 \times H\,\text{mm}\,H_2O$$

CENTFOR

The polynomial equation that was produced from a typical forward-curved blade centrifugal fan is

$$FTP = 2954.4 + 1.606Q - 0.17148 \times 10^{-3}Q^2 + 4.7115 \times 10^{-8}Q^3\ \text{Pa}$$

This equation is located in the range of cells from D36 to D76. The relevant value of Q is read from cells A36 to A76.

CENTBACK

The polynomial equation that was produced from a typical backward-curved blade centrifugal fan is

$$FTP = 4247.5 + 0.315Q + 1.159 \times 10^{-4}Q^2 - 0.1122 \times 10^{-6}Q^3\ \text{Pa}$$

This equation is located in the range of cells from D36 to D76. The relevant value of Q is read from cells A36 to A76. Backward-curved centrifugal fan impellers are shown in Fig. 6.5.

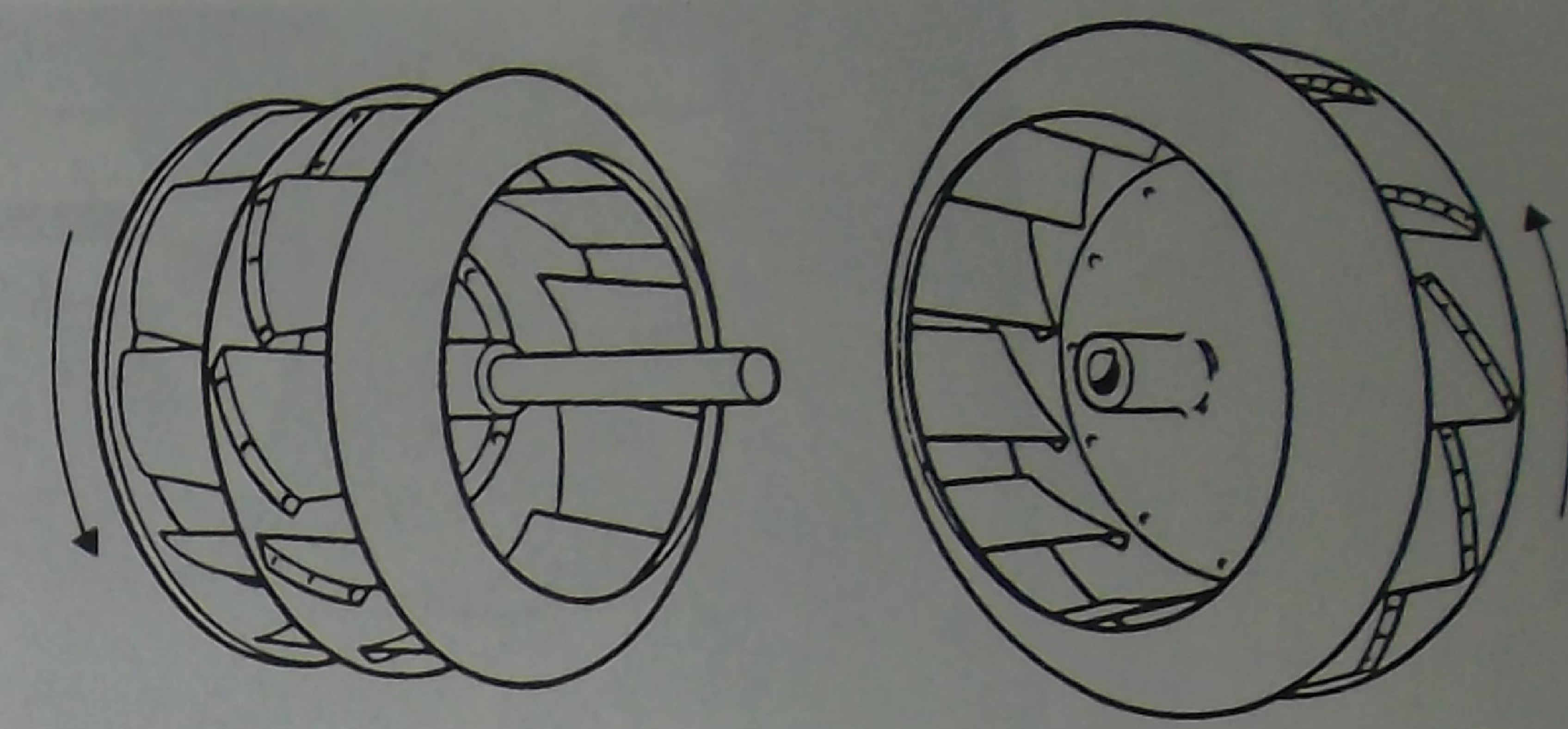


Fig. 6.5 Backward-curved centrifugal fan impellers (reproduced by permission of Woods of Colchester Ltd).

AXIAL

The polynomial equation that was produced from a typical axial flow fan is

$$FTP = 3136 - 3.065Q + 3.791 \times 10^{-3}Q^2 - 0.1144 \times 10^{-5}Q^3\ \text{Pa}$$

This equation is located in the range of cells from D36 to D76. The relevant value of Q is read from cells A36 to A76. An axial flow fan is shown in Fig. 6.6.

MIXED

The polynomial equation that was produced from a typical mixed flow fan is

$$FTP = 2075 - 1.28Q + 9.33 \times 10^{-4}Q^2 - 0.235 \times 10^{-6}Q^3\ \text{Pa}$$

This equation is located in the range of cells from D36 to D76. The relevant value of Q is read from cells A36 to A76. A mixed flow fan is shown in Fig. 6.7.

PROPFAN

The polynomial equation that was produced from a typical propeller fan is

$$FTP = 102 + 7.03 \times 10^{-3}Q - 0.23 \times 10^{-5}Q^2 - 0.41 \times 10^{-8}Q^3\ \text{Pa}$$

This equation is located in the range of cells from D36 to D76. The relevant value of Q is read from cells A36 to A76. A propeller fan is shown in Fig. 6.8.

COMPARE

This file contains comparative performance data and equations for backward-curved centrifugal, forward-curved centrifugal, axial flow, mixed flow and

Fig. 6.6 Axial flow fan (reproduced by permission of Woods of Colchester Ltd).

propeller fans. The file is not meant to be used for calculations. It provides only a visual comparison of the curves for the different types. The fans are not of equal size or performance capacity. They are plotted on the same axes to the same scale against the same system resistance curve. The reader can gain an understanding of how these fans would perform upon the same system.

DATA REQUIREMENT

The general data entry procedure is the same for each of the worksheets. Where there are differences, specific instructions are given with the fan type.

The user will enter new data for:

1. Your own name in cells B6.
2. The name of the job or the college assignment in cells B9 and C9; further cells along the row can be used.
3. Current date in cell B10.
4. The filename in cell B12.
5. The fan manufacturer's name in cells C16 to E16.
6. The fan model reference numbers in cells C17 to G17.
7. Highest fan speed to be used (rev/min) in cell B18.
8. Fan impeller diameter (mm) in cell B19.

Fig. 6.7 Mixed flow fan (reproduced by permission of Woods of Colchester Ltd).

Fig. 6.8 Propeller fan (reproduced by permission of Woods of Colchester Ltd).

9. Air density in cell B20. This is taken as 1.1906 kg/m^3 at 20 °C dry bulb, 60% relative humidity and 1013.25 mb atmospheric pressure.
10. Fan air outlet width (mm) in cell D22.
11. Fan air outlet height (mm) in cell F22.
12. Fan motor voltage (V) in cell D24. This will normally be 110, 240 or 415 V depending upon the supply voltage and voltage drop in the distribution cables due to their resistance.
13. Fan motor power factor in cell D25. This is less than 1.0 and is typically between 0.6 and 0.90. Power factor includes the electrical losses in the motor due to the resistance and inductance of the wiring.
14. Fan impeller efficiency (%) in cell D26. Typical values are 85% for a backward-curved centrifugal, 75% for a forward-curved centrifugal, 80% for a mixed flow, 70% for an axial and 60% for a propeller fan impeller. The fan manufacturer and the references can be used as necessary.
15. Fan motor overall efficiency (%) in cell D27. This should include all the mechanical and aerodynamic losses plus those in the drive system. A typical value is 80%. The electrical power losses were included in the power factor.
16. The design air flow, Q l/s, through the air ductwork system is entered into cell C31. This figure may be increased by 0–20% above the calculated design air flow to provide an excess air flow. This will generate the commissioning tolerance. Adding 20% to the design air flow will not necessarily produce a 20% commissioning tolerance because of the shape of the intersecting curves.
17. The total frictional resistance, d_p Pa, of the air ductwork system is entered into cell C32. This includes the pressure drop through the ductwork along the index route, duct fittings, dampers, filters, louvres, heating and cooling coils and the terminal unit. The index route is that series of air flow paths that have the greatest resistance. Other routes are in parallel to the index and they may have surplus pressure available to overcome their resistances.
18. The increments of the air flow, Q l/s, that are delivered by the fan are entered into cells A36 to A76. Commence with 0 l/s and end with the air flow that corresponds to the fan's delivering zero static pressure. Use equal increments of Q, such as 50 l/s or 100 l/s. The values of air flow that are already on the worksheet may be left if they are appropriate.
19. Enter a reduced value for the fan rotational speed, N rev/min, when compared with the previously entered full speed, into cell B92.
20. The design air flow, Q l/s, for the reduced fan speed is entered into cell C105. This figure may be increased by 0–20% above the calculated design air flow to provide an excess air flow. This will produce the commissioning tolerance.
21. The total frictional resistance, d_p Pa, of the air ductwork system that is supplied from the reduced fan speed is entered into cell C106. This includes the pressure drop through the ductwork along the index route, duct fittings, dampers, filters, louvres, heating and cooling coils and the terminal unit.

This concludes the data entry. Most of the other cells are either calculated automatically or are copied from the data that has been entered. Save this file onto the hard disk in drive C for the correct directory of your files, and on your own floppy disk in drive A now. For example, save it as C:\ASEASY\DAVIDC11\FANS\AXIAL9.WK1 and A:\FANS\AXIAL9.WK1, where ASEASY is the directory on the hard disk containing As-Easy-As, DAVIDC11 is your own subdirectory, FANS is your subdirectory for all work on fans, and AXIAL9.WK1 is the file name of the worksheet.

OUTPUT DATA

1. Fan outlet area is given in cell D23.
2. Air velocity in the fan outlet area is given in cell C36.
3. Fan total pressure rise that is calculated from the polynomial equation for the fan type is in cell D36.
4. Fan velocity pressure is calculated from the fan air discharge velocity and is in cell E36.
5. Fan static pressure is given in cell F36
6. Ductwork system resistance for each value of air flow from the fan is calculated in cell G36.
7. Electrical power input to the motor driving the fan is calculated from the air flow and fan total pressure rise in cell H36.

The calculated data for each air flow is repeated down to line 76.

FORMULAE

Representative samples of the formulae are given. Each formula can be read on the spreadsheet by moving the cursor to the cell. The equation is presented in the form in which it would normally be written and in the format that is used by the spreadsheet.

Cell D23

(F3) +D22*F22/10^6

The fan outlet duct cross-sectional area is calculated from the width and height of the opening. An axial, propeller or mixed flow fan has a circular outlet, and the area is calculated from $\pi d\ mm^2/4/10^6\ m^2$.

$$\text{fan outlet area D23} = \text{width D22 mm} \times \text{height F22 mm} \times \frac{1\ m^2}{10^6\ mm^2}$$

$$= \frac{\text{D22} \times \text{F22}}{10^6}\ m^2$$

Cell C36

+A36/1000/$D$23

The air velocity leaving the fan through the discharge duct is found by dividing the air volume flow rate by the cross-sectional area of the outlet. It is defined as the fan air velocity, and is used to calculate the fan velocity pressure.

$$\text{fan air velocity C39} = \frac{\text{air flow A36 l/s}}{\text{outlet area D23 } m^2} \times \frac{1\ m^3}{10^3\ \text{litre}}$$

$$= \frac{\text{A36}}{\text{D23} \times 10^3}\ m/s$$

Enter this data into the ANYFANWK1 worksheet. The outlet air duct is 450 mm diameter. The fan impeller has a diameter of 430 mm. The air density is 1.205 kg/m^3. A 240 V alternating electrical supply is to drive the fan. The motor power factor is 0.6, the fan impeller efficiency is 70%, and the overall motor and drive system efficiency is 60%. The fan reference code is MXFA.56, and the fan is manufactured by Joule Fan Co. Ltd. The extract duct design air flow requirement is 1100 l/s when the ductwork frictional resistance is 750 Pa.

(a) Find the operational values for the fan and system.
(b) Find the fan speed that must be used to meet a commissioning tolerance of $Q \pm 15\%$ l/s. State the operating data.
(c) State the excess air pressure from the fan.

10. A propeller fan is connected to a short extract duct from toilet accommodation. The fan reference code is WGP8004, and the fan is manufactured by Lyndhurst Fans Ltd. The fan is tested at a speed of 920 rev/min by the manufacturer and has the following performance:

Q (l/s)	*Fan static pressure, FSP* (Pa)
0	200
500	197
1000	193
1500	190
2000	180
2500	168
3000	160
3500	153
4000	144
4500	125
5000	90
5500	20

The outlet air duct is 750 mm diameter. The fan impeller has a diameter of 720 mm. The air density is 1.22 kg/m^3. Enter this data into the ANYFANWK1 worksheet. The air flow quantities, density and outlet diameter are entered first. Note the fan velocity pressure that the worksheet calculates. Manually add the fan velocity pressure to the fan static pressure in the question, and enter these fan total pressure values into the *FTP* column of the worksheet. Then progress to the remainder of the question. A 240 V alternating electrical supply is to drive the fan. The motor power factor is 0.7, the fan impeller efficiency is 60%, and the overall motor and drive system efficiency is 68%. The extract duct design air flow requirement is 3500 l/s when the ductwork frictional resistance is 120 Pa.

(a) Find the operational values for the fan and system.
(b) State the commissioning percentage tolerance that is produced.

The prison architect.

7 Air duct design

INTRODUCTION

The design procedure for air duct systems is explained. The principles are shown from the use of Bernoulli's equation for straight ducts, changes in duct cross-section, flows through branches, grilles and fans, through to a spreadsheet that can be used for duct routes. Both supply and extract air duct systems are used. The spreadsheet can be used for individual air ducts and systems having up to five branches. It can be expanded by the user to accommodate larger systems once the methods are understood. Node numbers are used throughout the duct systems, and these can easily be changed by the user for applications. The air temperature can be changed for each section of duct. Air pressure terms and methods of measurement are explained. Detailed pressure changes at duct fittings and fans are calculated. The worksheet calculates all the total and static air pressures throughout the duct system and in the ventilated room or space. The index duct route and the fan pressure rise are automatically selected. There is no need to refer to reference data, tables or charts to be able to calculate the correct sizes of air ducts, because sufficient data for many applications is on the worksheet. Data on fittings' velocity pressure loss factors can be entered from other sources.

The method that is used to calculate the duct carrying capacity is the Colebrook and White equation for the design friction factor in the D'Arcy formula including the fluid properties and pipe material factors. The data is for clean galvanized sheet metal ducts having joints made in accordance with good practice (CIBSE, 1986c). The overall formula for duct air flow is in a reliable and usable form, which reproduces published data within acceptable accuracy. It can easily be verified with manual calculations and by reference to the source equations and duct chart.

LEARNING OBJECTIVES

Study and use of this chapter will enable the user to:

1. understand the terms total, static and velocity air pressures;
2. calculate the density of air;
3. calculate total, static and velocity air pressures;
4. understand and use frictional resistance air pressure drops in duct systems and changes in duct size;
5. understand and use static regain air pressure;

6. understand the use of velocity pressure loss factors;
7. calculate the changes in air pressure when air flows through ductwork and branches;
8. calculate the carrying capacities of air ducts;
9. calculate air flow and pressure data for air at different temperatures and pressures;
10. calculate the carrying capacity of air ducts for values of pressure drop rate and duct diameter;
11. understand how air pressure changes take place though duct systems;
12. draw air pressure gradient graphs;
13. understand and use the changes in air pressure that occur at a fan;
14. understand and use the terms fan total, static and velocity pressures;
15. calculate the equivalent diameter of rectangular air ducts;
16. calculate the static air pressure in a mechanically ventilated room;
17. use numbered nodes in the design of air duct systems;
18. find the correct dimensions for air duct systems;
19. make adjustments to air flows in duct systems for variations in air temperature;
20. use limiting air velocity and pressure drop rate data;
21. know the limiting air velocities for low-velocity duct systems;
22. analyse supply and extract air ductwork systems;
23. know the equations for air duct sizing and their method of use;
24. find the index route that offers the highest frictional resistance to the fan;
25. know how a spreadsheet uses conditional 'IF' statements to select a highest value;
26. know how to specify the performance criteria for a fan;
27. know how to calculate the balancing pressure in branch ducts from the index route.

Key terms and concepts

AIR PRESSURES IN A DUCT

Air flowing through a duct exerts two types of pressure on its surroundings. The first is dynamic pressure due to its motion, or kinetic energy. This is velocity pressure, p_v; it acts only in the direction of the flow of air.

$$p_v = 0.5\rho v^2 \text{ Pa}$$

where ρ = air density (kg/m^3) and v = air velocity (m/s).

The second is the bursting pressure of the air trying to escape from the enclosure or of the surrounding air trying to enter the enclosing duct. This is static pressure, p_s Pa, and it acts in all directions.

The sum of these two pressures is the total pressure p_t:

$$\text{total pressure} = \text{static pressure} + \text{velocity pressure}$$

$$p_t = p_s + p_v \text{ Pa}$$

Losses of pressure due to the frictional resistance of the duct, its bends, branches, changes in cross-section, dampers, filters, air heating or cooling coils, the entry of air into a duct system and the exit of air from a duct, cause loss of total pressure.

Bernoulli's theorem means that, for a fluid flowing from position 1 to position 2, the total pressure energy at point 1 equals the total pressure energy at point 2 plus the loss of energy due to friction between the two locations. All energy lost in friction is dissipated as heat, and in some cases this can cause a noticeable rise in the air temperature. A plot of the pressures that occur when air flows through an enlargement in a duct is shown in Fig. 7.1. The data in Fig. 7.1 relates to Example 7.1. It can be seen that the air static pressure increases through the enlargement because the drop of velocity pressure is greater than the loss of pressure due to friction.

The balance of air pressures between 1 and 2 is:

$$p_{t1} = p_{t2} + dp_{1-2}$$

where p_{t1} = total pressure at 1; p_{t2} = total pressure at 2; and dp_{1-2} = pressure drop due to friction between nodes 1 and 2.

The balance of pressures between nodes 2 and 3 is

$$p_{t2} = p_{s2} + p_{v2}$$

$$p_{t3} = p_{s3} + p_{v3}$$

but

$$p_{t2} = p_{t3} + dp_{2-3}$$

and

$$\begin{aligned} dp_{2-3} &= p_{t2} - p_{t3} \\ &= p_{s2} + p_{v2} - (p_{s3} + p_{v3}) \\ &= p_{v2} - p_{v3} + p_{s2} - p_{s3} \\ p_{s3} - p_{s2} &= (p_{v2} - p_{v3}) - dp_{2-3} \end{aligned}$$

This shows that a regain of static pressure occurs when a reduction in the velocity pressure is greater than the pressure drop due to the frictional resistance of the

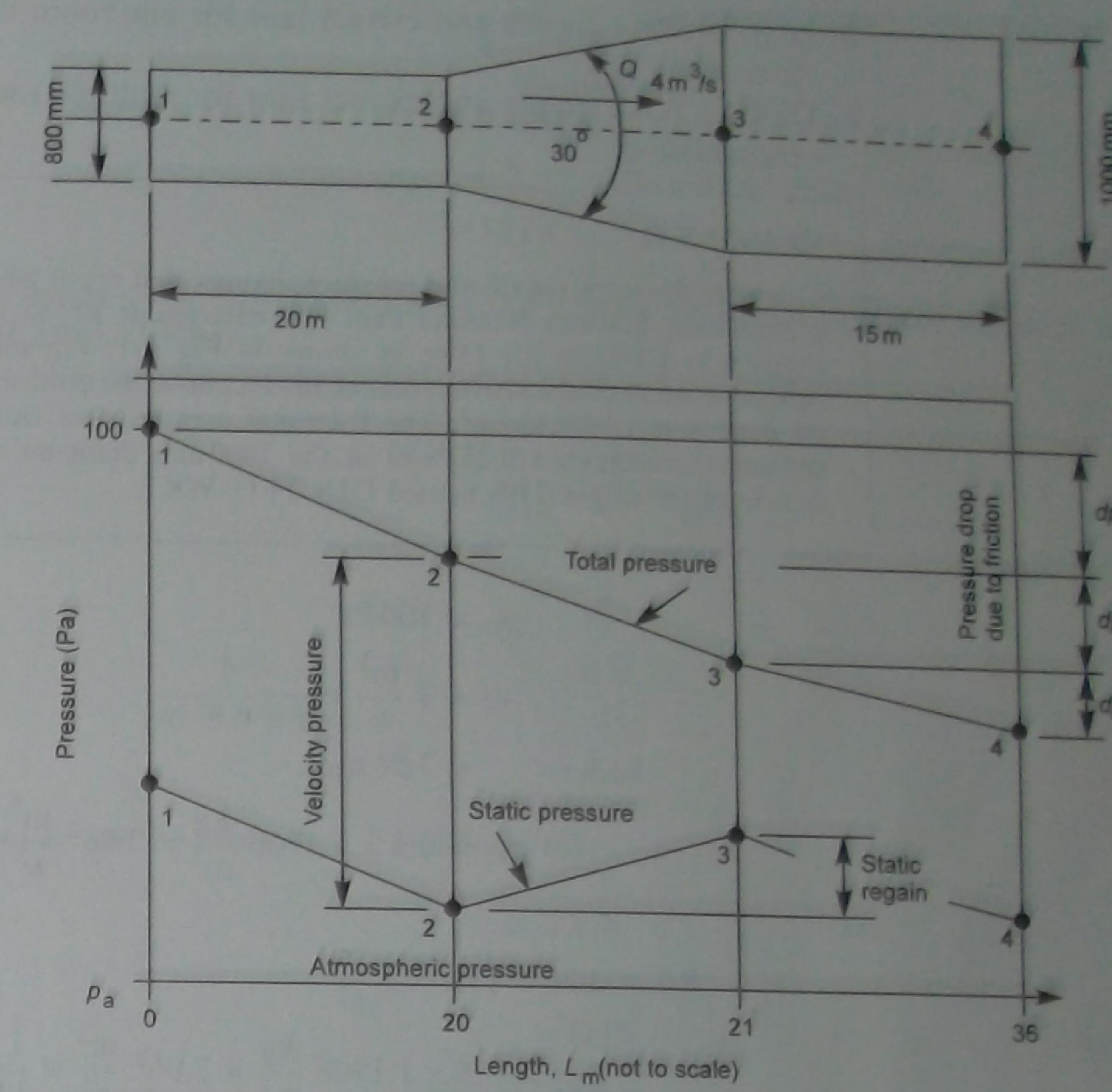


Fig. 7.1 Changes of pressure as air flows through an enlargement.

section of duct. This can take place at a branch fitting, at a change of duct dimensions, or when air discharges into a room or plenum space such as a ceiling void.

The worked examples can be entered onto an appropriate worksheet. The worksheet uses more decimal places than the user would write, so minor differences between calculated and computed answers will arise. These differences will normally be within 1 Pa, 1 mm or 1 litre/s. Two worksheet files are provided on the disk. The file DUCTFIT.WK1 is to calculate the pressure changes through air duct enlargements, contractions and branch fittings. The file DUCT1.WK1 is for the most common application, where there is a supply air fan and ductwork which has up to five branches. These two files have been used for all the worksheet calculations in this chapter. When the user has gained experience from using the information and methods that have been provided, worksheets may be adapted for other applications. It is suggested that the user may produce adaptations to the file DUCT1.WK1 for the following applications:

1. DUCT2.WK1 extract fan and ductwork with five branches;
2. DUCT3.WK1 supply and extract fans for one room and no recirculation of room air;

3. DUCT4.WK1 supply and extract fans for one room with room air recirculation;
4. DUCT5.WK1 supply and extract fans for multiple rooms and with room air recirculation.

EXAMPLE 7.1

Calculate the static regain and pressure changes that occur when $4\,m^3/s$ flow through a 20 m long, 800 mm diameter duct that enlarges at 30° to 1000 mm diameter and remains at 1000 mm for 15 m, as shown in Fig. 7.1. The air total pressure at the commencement of the 800 mm duct is 100 Pa above the atmospheric air pressure. The air density is $1.1906\,kg/m^3$. The frictional pressure loss rates are 0.7 Pa/m in the 800 mm diameter and 0.25 Pa/m in the 1000 mm diameter ducts. The solution is shown on the original file named DUCTFIT.WK1.

$$p_{t1} = 100\,\text{Pa}$$

$$v_1 = 4\,\frac{\text{m}^3}{\text{s}} \times \frac{4}{\pi \times 0.8^2\,\text{m}^2}$$
$$= 7.96\,\text{m/s}$$

$$p_{v1} = 0.5 \times 1.1906\,\frac{\text{kg}}{\text{m}^3} \times 7.96^2\,\frac{\text{m}^2}{\text{s}^2}$$

As

$$1\,\text{N} = 1\,\text{kg} \times 1\,\frac{\text{m}}{\text{s}^2}$$

$$p_{v1} = 0.5 \times 1.1906\,\frac{\text{kg}}{\text{m}^3} \times 7.96^2\,\frac{\text{m}^2}{\text{s}^2} \times \frac{1\,\text{N}\,\text{s}^2}{1\,\text{kg}\,\text{m}}$$
$$= 38\,\frac{\text{N}}{\text{m}^2}$$

and

$$1\,\text{Pa} = 1\,\text{N/m}^2$$
$$p_{v1} = 38\,\text{Pa}$$
$$p_{s1} = p_{t1} - p_{v1}$$
$$= 100 - 38\,\text{Pa}$$
$$= 62\,\text{Pa}$$
$$dp_{1-2} = 0.7\,\frac{\text{Pa}}{\text{m}} \times 20\,\text{m}$$
$$= 14\,\text{Pa}$$
$$p_{t2} = p_{t1} - dp_{1-2}$$
$$= 100 - 14\,\text{Pa}$$
$$= 86\,\text{Pa}$$
$$p_{s2} = p_{t2} - p_{v2}$$
$$p_{v2} = p_{v1}$$
$$p_{s2} = 86 - 38\,\text{Pa}$$
$$p_{s2} = 48\,\text{Pa}$$

$$v_3 = 4\,\frac{\text{m}^3}{\text{s}} \times \frac{4}{\pi \times 1^2\text{m}^2}$$
$$= 5.09\,\text{m/s}$$
$$p_{v3} = 0.5 \times 1.1906 \times 5.09^2\,\text{Pa}$$
$$= 15\,\text{Pa}$$

Figures 7.14–7.16 show the velocity pressure loss factors for air duct fittings (CIBSE, 1986c).

For an abrupt enlargement of a circular duct:

$$\frac{A_2}{A_3} = \frac{\pi d_2^2}{4} \times \frac{4}{\pi d_3^2}$$
$$= \frac{d_2^2}{d_3^2}$$
$$= \frac{0.8^2}{1^2}$$
$$= 0.64$$

so k_1 can be taken as 0.2 and k_2 as 0.8:

$$k = k_1 k_2$$
$$= 0.2 \times 0.8$$
$$= 0.16$$
$$dp_{2-3} = k p_{v2}$$
$$= 0.16 \times 38\,\text{Pa}$$
$$= 6\,\text{Pa}$$
$$p_{t3} = p_{t2} - dp_{2-3}$$
$$= 86 - 6\,\text{Pa}$$
$$= 80\,\text{Pa}$$
$$p_{s3} = p_{t3} - p_{v3}$$
$$= 80 - 15$$
$$= 65\,\text{Pa}$$
$$dp_{3-4} = 15\,\text{m} \times 0.25\,\frac{\text{Pa}}{\text{m}}$$
$$= 4\,\text{Pa}$$
$$p_{t4} = p_{t3} - dp_{3-4}$$
$$= 80 - 4\,\text{Pa}$$
$$= 76\,\text{Pa}$$
$$p_{s4} = p_{t4} - p_{v4}$$
$$p_{v3} = p_{v4}$$
$$p_{s4} = 76 - 15\,\text{Pa}$$
$$= 61\,\text{Pa}$$

The regain of static pressure is

$$SR = p_{s3} - p_{s2}\ \text{Pa}$$
$$= 65 - 48\ \text{Pa}$$
$$= 17\ \text{Pa}$$

EXAMPLE 7.2

Calculate the total pressure of air flowing at $0.75\ \text{m}^3/\text{s}$ in a 400 mm internal diameter duct when the air temperatures is 18 °C d.b. and the static pressure in the duct is 40 mm water gauge above the atmospheric pressure of 101 560 Pa. Also calculate the total pressure when the air is at −5 °C and 30 °C during winter and summer operation of the air conditioning system. A worksheet is not used for this example.

The atmospheric pressure within the duct:

$$p_a = 101\,560 + p_s\ \text{Pa}$$
$$p_s = 9.807 \times 40\ \text{mm}\ H_2O\ \text{Pa}$$
$$= 392.3\ \text{Pa}$$
$$p_a = 101\,560 + 392.3$$
$$= 101\,952.3\ \text{Pa}$$

It is reasonable to ignore decimal parts of pascal pressures, so $p_s = 392$ Pa and $p_a = 101\,952$ Pa.

$$\rho = 1.1906 \times \frac{p_a}{101\,325} \times \frac{273 + 20}{273 + t}\ \frac{\text{kg}}{\text{m}^3}$$
$$= 1.1906 \times \frac{101\,952}{101\,325} \times \frac{273 + 20}{273 + 18}\ \frac{\text{kg}}{\text{m}^3}$$
$$= 1.2062\ \text{kg/m}^3$$
$$v = \frac{4 \times 0.75}{\pi \times 0.4^2}\ \frac{\text{m}}{\text{s}}$$
$$= 5.97\ \text{m/s}$$
$$p_v = 0.5 \times 1.2062 \times 5.97^2\ \text{Pa}$$
$$= 21.5\ \text{Pa}$$
$$= 22\ \text{Pa}$$

$$\text{total pressure } p_t = p_s + p_v$$
$$p_t = 392 + 22\ \text{Pa}$$
$$= 414\ \text{Pa}$$

When the air is at −5 °C but Q remains at $0.75\ \text{m}^3/\text{s}$:

$$\rho = 1.1906 \times \frac{101\,952}{101\,325} \times \frac{273 + 20}{273 + -5}\ \frac{\text{kg}}{\text{m}^3}$$
$$= 1.3097\ \text{kg/m}^3$$
$$v = 5.97\ \text{m/s}$$
$$p_v = 0.5 \times 1.3097 \times 5.97^2\ \text{Pa}$$
$$= 23\ \text{Pa}$$

$$\text{total pressure } p_t = p_s + p_v$$
$$p_t = 392 + 23\ \text{Pa}$$
$$= 415\ \text{Pa}$$

When the air is at 30 °C but Q remains at $0.75\ \text{m}^3/\text{s}$:

$$\rho = 1.158\ \text{kg/m}^3$$
$$v = 5.97\ \text{m/s}$$
$$p_v = 21\ \text{Pa}$$
$$p_t = 413\ \text{Pa}$$

AIR FLOW IN DUCTS

The flow of air at 20 °C d.b., 43% percentage saturation and a barometric pressure of 101 325 Pa through clean galvanized sheet metal ducts having joints made in accordance with good practice is given by

$$Q = -2.0278 \times dp^{0.5} \times d^{2.5} \times \log\left(\frac{4.05 \times 10^{-5}}{d} + \frac{2.933 \times 10^{-5}}{dp^{0.5} \times d^{1.5}}\right)$$

(CIBSE 1986c, adapted for fluid properties stated; Chadderton, 1997)
where Q = air flow rate (m^3/s); dp = pressure loss rate (Pa/m); and d = duct internal diameter (m).

EXAMPLE 7.3

A 700 mm internal diameter galvanized sheet steel duct is carrying air at 20 °C d.b. and has a design pressure loss rate of 0.8 Pa/m. Calculate the maximum air volume flow rate that can be passed and the air velocity. A worksheet is not used for this example.

$$d = 0.70\ \text{m}$$
$$dp = 0.8\ \text{Pa/m}$$
$$Q = -2.0278 \times 0.8^{0.5} \times 0.7^{2.5} \times \log\left(\frac{4.05 \times 10^{-5}}{0.7} + \frac{2.933 \times 10^{-5}}{0.8^{0.5} \times 0.7^{1.5}}\right)$$
$$= 2.932\ \text{m}^3/\text{s}$$
$$v = 2.932\ \frac{\text{m}^3}{\text{s}} \times \frac{4}{\pi \times 0.7^2}\ \text{m}^2$$
$$= 7.62\ \text{m/s}$$

EXAMPLE 7.4

Find a suitable diameter for a galvanized sheet steel duct that is to carry 500 l/s of air at 20 °C d.b. at a maximum velocity of 5.0 m/s when the pressure loss rate is not to exceed 0.6 Pa/m. Duct diameters are in increments of 50 mm. A worksheet is not used for this example.

An iterative procedure is needed; try $d = 350$ mm.

$$\text{air flow } Q = 0.5\,\text{m}^3/\text{s}$$

$$v = 0.5\,\frac{\text{m}}{\text{s}} \times \frac{4}{\pi \times 0.35^2}\,\text{m}^2 = 5.2\,\text{m/s}$$

This is over the limit. Try $d = 0.4$ m at $Q = 0.5\,\text{m}^3/\text{s}$:

$$v = 3.98\,\text{m/s}$$

A 400 mm diameter duct will be used, provided that the frictional pressure loss rate does not exceed $dp = 0.6$ Pa/m. Insert $dp = 0.6$ Pa/m and $d = 0.4$ m into the duct formula and calculate the maximum carrying capacity, irrespective of limiting velocity:

$$Q = -2.0278 \times 0.6^{0.5} \times 0.4^{2.5} \times \log\left(\frac{4.05 \times 10^{-5}}{0.4} + \frac{2.933 \times 10^{-5}}{0.6^{0.5} \times 0.4^{1.5}}\right)$$

$$= 0.572\,\text{m}^3/\text{s}$$

This is greater than required, so the actual pressure loss rate will be less than 0.6 Pa/m, around 0.47 Pa/m.

VARIATIONS OF PRESSURE ALONG A DUCT

Pressure gradients are produced along the length of a duct, as shown in Fig. 7.1. The gradient caused along the constant-diameter duct is calculated from the pressure drop rate due to friction. The gradients (slope) of the total and static pressure lines are equal. Friction losses can be calculated as total or static pressure drops. Pressure drops will be treated as reductions in the available total pressure.

Duct fittings cause loss of pressure through surface friction and additional turbulence because of the shape of the enclosure. Such losses are found from experiment, and require the use of unique factors that are multiplied by the velocity pressure. Data for commonly used duct fittings are given in Figs 7.14–7.16 (CIBSE, 1986c).

To produce a pressure gradient diagram similar to Fig. 7.1, draw the ductwork to be analysed directly above a graph as shown. Clearly mark each point where a change of section takes place, and identify each with a number. These points are the nodes that will be used for the worksheet pressure analysis and for finding the sizes of the ducts. This method is common to all duct-sizing programs. The sequence is:

1. Find the total pressure at 1, p_{t1}, and plot it to an appropriate scale above, or below, atmospheric pressure p_a Pa.
2. Calculate the pressure drop between 1 and 2 in the straight duct, dp_{1-2}. Note that dp_{1-2} means the pressure drop due to friction between nodes 1 and 2.
3. Subtract dp_{1-2} from p_{t1} to find the total pressure at point 2, p_{t2} Pa.
4. Draw the total pressure straight line from p_{t1} to p_{t2}.
5. Find the velocity pressure loss factor for the duct fitting, k, from Figs 7.14–7.16 or the reference.
6. Find the velocity pressure to be used for the frictional resistance calculation for the fitting. In this case, it is the velocity pressure in the smaller duct, p_{v3}.
7. Multiply the fitting k factor by the velocity pressure p_{v3} to calculate the pressure drop through the fitting, dp_{2-3}.
8. Subtract pressure drop dp_{2-3} from the total pressure p_{t2} to find the total pressure at 3, p_{t3}.
9. Plot p_{t3} and connect the total pressure straight line from p_{t2}.
10. Calculate the pressure drop between 3 and 4 in the straight duct, dp_{3-4}.
11. Subtract dp_{3-4} from p_{t3} to find the total pressure at point 4, p_{t4} Pa.
12. Draw the total pressure straight line from p_{t3} to p_{t4}.
13. Calculate the velocity pressure at 1, p_{v1} Pa. Note that p_{v1} is equal to p_{v2}, because the velocity remains constant along a uniform-diameter duct.
14. Subtract p_{v1} from both p_{t1} and p_{t2}. The differences are the static pressures at 1 and 2, p_{s1} and p_{s2}.
15. Plot p_{s1} and p_{s2}, and connect them with a straight line.
16. Subtract p_{v3} from p_{t3} and find static pressure at 3, p_{s3}.
17. Draw a straight line from p_{s2} to p_{s3}.
18. Subtract p_{v3} from p_{t4} and find p_{s4}. Note that p_{v3} equals p_{v4}.
19. Draw a straight line from p_{s3} to p_{s4}.
20. Mark the upper gradient as the total pressure line.
21. Mark the lower line as the gradient of static pressure.
22. Mark the differences between the two gradients as velocity pressures.
23. Identify each of the corresponding points of the duct diagram and the graph.
24. Mark on the graph all the calculated pressures.

This procedure applies to other air duct cases and to the changes of pressure across a fan.

Notice that for this increase in diameter of the duct, the static pressure rises. This is because the velocity pressure has reduced. The available total pressure has fallen because of friction losses. Such a regain of static pressure depends on the amount of frictional pressure drop that occurs. Static regain can also take place at branches because of a reduction in the air flow quantity and velocity. Excess static pressure in branch ducts can be absorbed by reducing the duct diameter, provided that the velocity limit is not exceeded, or by partial closure of a balancing damper. The worksheet shows a warning when there is excessive static pressure at a branch duct outlet. The user then enters new data until a satisfactory system balance is produced. This forces the user to understand the procedure of designing a correctly balanced air duct network. High-cost software will make such iterations automatically, leaving the designer unaware of how the design was balanced.

EXAMPLE 7.5

Air is blown at $2.5\,\text{m}^3/\text{s}$ into a 35 m long 700 mm diameter duct, which reduces to 500 mm diameter and remains at 500 mm for 22 m. The air is discharged into a room that is at atmospheric pressure. The air total pressure at the commencement of the 700 mm duct is 192 Pa above the atmospheric pressure. The reducer is the 30° concentric type. The air density is $1.23\,\text{kg/m}^3$. The frictional pressure loss rates are 0.6 Pa/m in the 700 mm diameter duct and 3.2 Pa/m in the 500 mm diameter duct. Calculate the air pressures in the duct system and draw them on a graph. Enter the data into a copy of file DUCTFIT.WK1 and save it as file DUCTFIT2.WK1.

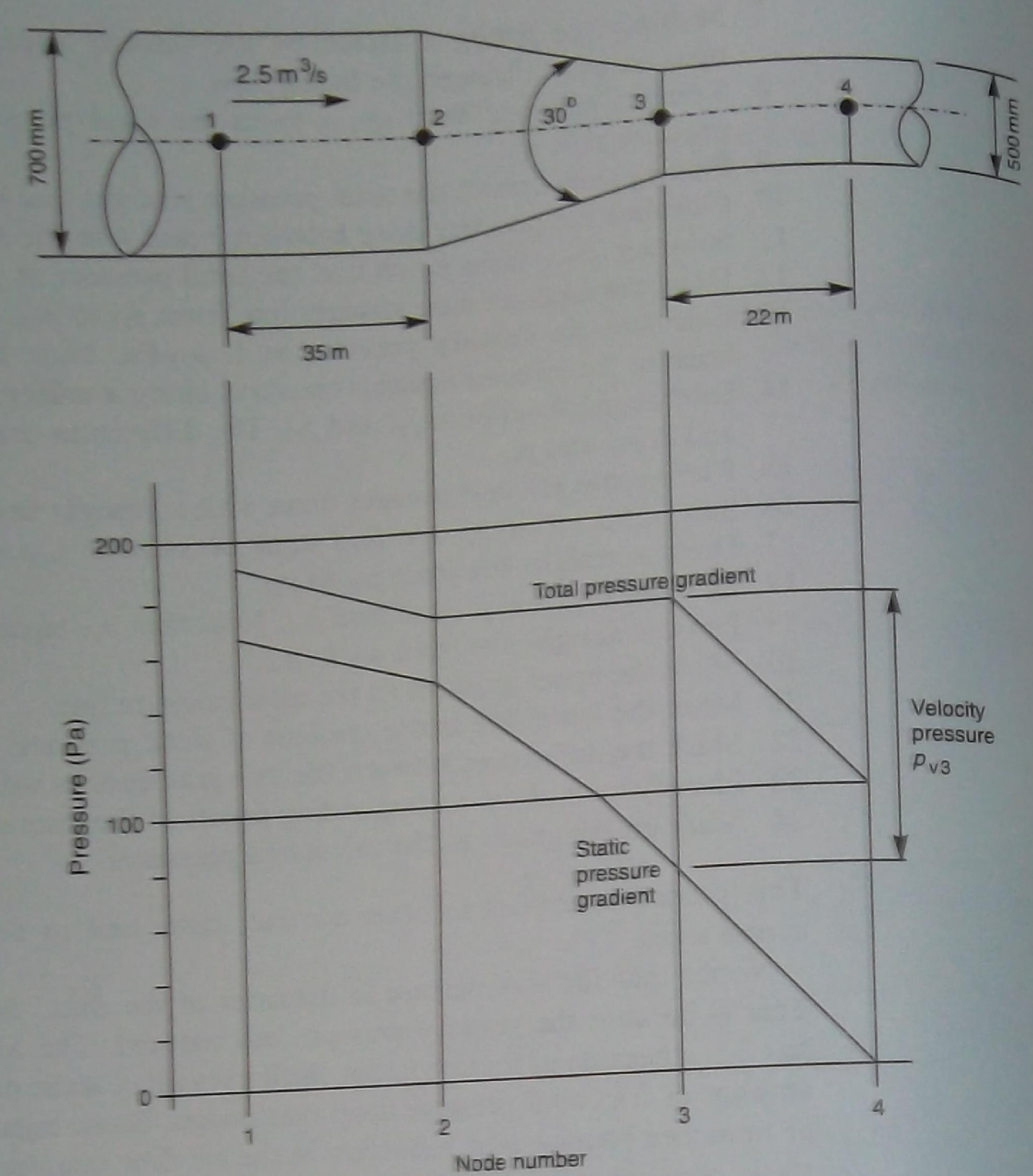


Fig. 7.2 Pressure changes through the contraction in Example 7.5.

Using the node numbers from Fig. 7.2 and the sequence numbers listed:

1. $p_{t1} = 192\,\text{Pa}$; plot graph axes.
2. $dp_{1-2} = 0.6\,\text{Pa/m} \times 35\,\text{m}$
   $= 21\,\text{Pa}$
3. $p_{t2} = p_{t1} - dp_{1-2}$
   $= 192 - 21\,\text{Pa}$
   $= 171\,\text{Pa}$
4. Draw the total pressure line 1 to 2.
5. The reducer has a velocity pressure loss factor k of 0.02, and this is multiplied by the smaller duct velocity pressure, p_{v3}.
6. $v_3 = 2.5\dfrac{\text{m}^3}{\text{s}} \times \dfrac{4}{\pi \times 0.5^2\,\text{m}^2}$
   $= 12.7\,\text{m/s}$
   $p_{v3} = 0.5 \times 1.23 \times 12.7^2\,\text{Pa}$
   $= 99\,\text{Pa}$

7. Pressure drop through reducer:
   $dp_{2-3} = \text{factor} \times p_{v3}$
   $= 0.02 \times 99\,\text{Pa}$
   $= 2\,\text{Pa}$
8. $p_{t3} = p_{t2} - dp_{2-3}$
   $= 171 - 2\,\text{Pa}$
   $= 169\,\text{Pa}$
9. Plot total pressure line 2 to 3.
10. Pressure drop through 500 mm duct:
   $dp_{3-4} = 22\,\text{m} \times 3.2\,\text{Pa/m}$
   $= 70\,\text{Pa}$
11. $p_{t4} = p_{t3} - dp_{3-4}$
   $= 169 - 70\,\text{Pa}$
   $= 99\,\text{Pa}$
12. Draw total pressure line 3 to 4.
13. $v_1 = 2.5\dfrac{\text{m}^3}{\text{s}} \times \dfrac{4}{\pi \times 0.7^2}\,\text{m}^2$
   $= 6.5\,\text{m/s}$
   $p_{v1} = 0.5 \times 1.23 \times 6.5^2\,\text{Pa}$
   $= 26\,\text{Pa}$
   $p_{v2} = 26\,\text{Pa}$
14. $p_{s1} = p_{t1} - p_{v1}$
   $= 192 - 26\,\text{Pa}$
   $= 166\,\text{Pa}$
   $p_{s2} = p_{t2} - p_{v2}$
   $p_{v2} = p_{v1}$
   $p_{s2} = 171 - 26\,\text{Pa}$
   $= 145\,\text{Pa}$
15. Plot static pressures 1 and 2, and join them with a line.
16. $p_{s3} = p_{t3} - p_{v3}$
   $= 169 - 99\,\text{Pa}$
   $= 70\,\text{Pa}$
17. Draw static pressure line from 2 to 3.
18. $p_{s4} = p_{t4} - p_{v4}$
   $p_{v3} = p_{v4}$
   $p_{s4} = 99 - 99\,\text{Pa}$
   $= 0\,\text{Pa}$
   This is atmospheric pressure.
19. Draw static pressure line from 3 to 4.

20–24. Figure 7.2 shows these answers plotted to scale.

PRESSURE CHANGES AT A FAN

The fan produces a rise of total pressure in the whole system. This rise is the fan total pressure, *FTP* Pa. The fan total pressure rise is equal to the drop of total pressure due to friction in the ductwork system plus the increase of air velocity pressure that has been generated by the fan. The fan total pressure is calculated from the total pressure in the fan outlet duct, node 2, minus the total pressure in the fan inlet duct, node 1:

$$FTP = p_{t2} - p_{t1}\ \text{Pa}$$

Fan velocity pressure, *FVP*, is defined as the velocity pressure in the discharge area from the fan, p_{v2}:

$$FVP = p_{v2}\ \text{Pa}$$

Fan static pressure, *FSP*, is defined as fan total pressure minus fan velocity pressure. Note that this is not necessarily the same as the change in static pressure across the fan connections:

$$FSP = FTP - FVP\ \text{Pa}$$

> EXAMPLE 7.6
>
> The 1200 mm × 1200 mm air-handling plant shown in Fig. 7.3 supplies 3 m³/s of air at 30 °C into a room. The axial flow fan has a diameter of 550 mm. Outdoor air at 5 °C is drawn through a louvre and filter. The room is at atmospheric pressure. The fixed pressure drops are: filter 450 Pa, heater coil 125 Pa, and discharge grille 45 Pa. The pressure drop rate in the plant ductwork is estimated to be 0.1 Pa/m. Calculate the pressures at each node. Enter the data into a copy of the file DUCT1.WK1 and save it as file DUCT7%6.WK1. The node numbers to be used are shown in Fig. 7.3.

From Figs 7.14–7.16 the velocity pressure loss factors are as follows.
Air intake louvre for entry, louvre and wire mesh:

$$k = 0.5 + 4.5 + 1.7$$
$$= 6.7$$

45° gradual contraction:

$$k = 0.04$$

30° gradual enlargement:

$$\frac{A_1}{A_2} = \frac{\pi \times 0.55^2\,\text{m}^2}{4} \times \frac{1}{1.2\,\text{m} \times 1.2\,\text{m}}$$
$$= 0.17$$

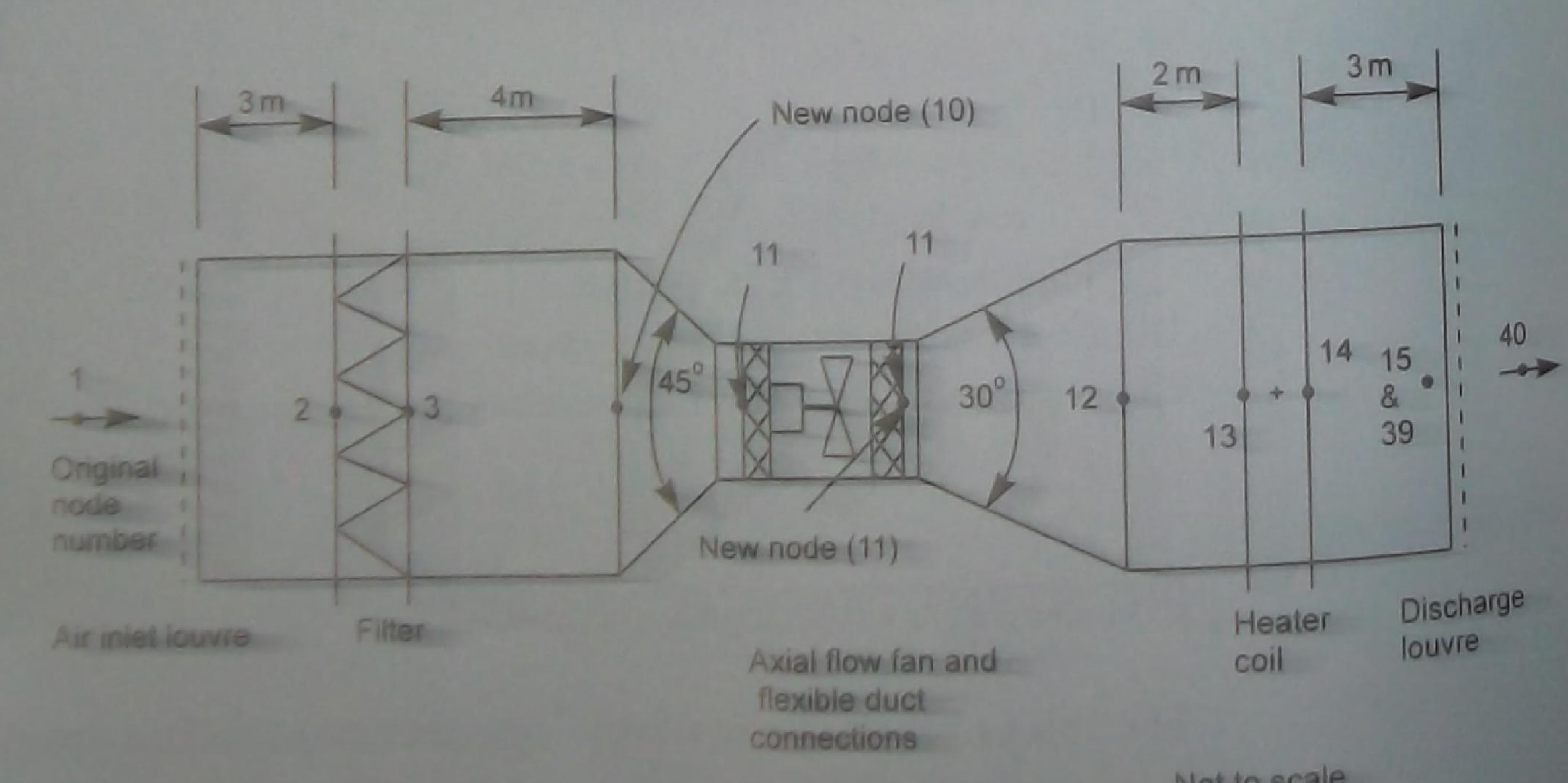


Fig. 7.3 Air-handling plant in Example 7.6.

$$k_1 = 0.7 \text{ estimated}$$
$$k_2 = 0.8$$
$$k = 0.7 \times 0.8$$
$$= 0.56$$

Duct exit:

$$k = 1$$

The fan air volume flow at 5 °C is

$$Q_1 = 3\frac{\text{m}^3}{\text{s}} \times \frac{273 + 5}{273 + 30}$$
$$= 2.753\,\text{m}^3/\text{s}$$

After the heater coil:

$$Q_{14} = 3\,\text{m}^3/\text{s}$$

The circular equivalent of the rectangular plant duct is found from (CIBSE, 1986c)

$$d_2 = 1.265 \times \left[\frac{(WD)^3}{W + D}\right]^{0.2} \text{mm}$$
$$= 1.265 \times \left[\frac{(1200 \times 1200)^3}{1200 + 1200}\right]^{0.2} \text{mm}$$
$$= 1322\,\text{mm}$$

$$v_2 = 2.753\frac{\text{m}^3}{\text{s}} \times \frac{4}{\pi \times 1.322^2\,\text{m}^2}$$
$$= 2\,\text{m/s}$$

$$v_{11} = 2.753\frac{\text{m}^3}{\text{s}} \times \frac{4}{\pi \times 0.55^2\,\text{m}^2}$$
$$= 11.59\,\text{m/s}$$

$$v_{14} = 3\frac{\text{m}^3}{\text{s}} \times \frac{4}{\pi \times 1.322^2\,\text{m}^2}$$
$$= 2.19\,\text{m/s}$$

$$\rho_2 = 1.1906\frac{\text{kg}}{\text{m}^3} \times \frac{273 + 20}{273 + 5}$$
$$= 1.255\,\text{kg/m}^3$$

$$\rho_{14} = 1.1906\frac{\text{kg}}{\text{m}^3} \times \frac{273 + 20}{273 + 30}$$
$$= 1.151\,\text{kg/m}^3$$

$$p_{v2} = 0.5 \times \rho_2 \times v_2^2\,\text{Pa}$$
$$= 0.5 \times 1.255 \times 2^2\,\text{Pa}$$
$$= 2.5\,\text{Pa}$$

$$p_{v11} = 0.5 \times \rho_2 \times v_{11}^2\,\text{Pa}$$
$$= 0.5 \times 1.255 \times 11.59^2\,\text{Pa}$$
$$= 84.3\,\text{Pa}$$

$$
\begin{aligned}
p_{v14} &= 0.5 \times \rho_{14} \times v_{14}^2 \,\text{Pa} \\
&= 0.5 \times 1.151 \times 2.19^2 \,\text{Pa} \\
&= 2.8 \,\text{Pa} \\
dp_{1-2} &= kp_{v2} + L\,\text{m} \times dp \frac{\text{Pa}}{\text{m}} \\
&= 6.7 \times 2.5 + 3 \times 0.1 \,\text{Pa} \\
&= 17 \,\text{Pa} \\
dp_{2-3} &= 450 \,\text{Pa} \\
dp_{3-10} &= 4 \times 0.1 \,\text{Pa} \\
&= 0.4 \,\text{Pa} \\
dp_{10-11} &= kp_{v11} \,\text{Pa} \\
&= 0.04 \times 84.3 \,\text{Pa} \\
&= 3.4 \,\text{Pa} \\
dp_{11-12} &= kp_{v11} \,\text{Pa} \\
&= 0.56 \times 84.3 \,\text{Pa} \\
&= 47.2 \,\text{Pa} \\
dp_{12-40} &= \text{duct} + \text{heater coil} + \text{grille} + \text{discharge } k \\
&= 5 \times 0.1 + 125 + 45 + kp_{v39} \,\text{Pa} \\
&= 170.5 + 1 \times 2.5 \,\text{Pa} \\
&= 173 \,\text{Pa}
\end{aligned}
$$

$$
\begin{aligned}
\text{drop of total pressure, } dp_{1-40} &= 17 + 450 + 0.4 + 3.4 + 47.2 + 173 \,\text{Pa} \\
&= 691 \,\text{Pa} \\
\text{fan total pressure rise, } FTP &= dp_{1-10} + p_{v40} - p_{v1} \,\text{Pa} \\
p_{v40} &= p_{v2} = 2.5 \,\text{Pa} \\
p_{v1} &= p_{v2} = 2.5 \,\text{Pa} \\
\text{fan total pressure rise, } FTP &= 691 + 2.5 - 2.5 \,\text{Pa} \\
&= 691 \,\text{Pa} \\
\text{fan velocity pressure, } FVP &= p_{v11} \\
&= 84 \,\text{Pa} \\
\text{fan static pressure, } FSP &= 691 - 84 \,\text{Pa} \\
&= 607 \,\text{Pa}
\end{aligned}
$$

$$
\begin{aligned}
p_{t1} &= 0 \,\text{Pa, atmospheric pressure} \\
p_{s1} &= 0 \,\text{Pa, atmospheric pressure} \\
p_{t2} &= p_{t1} - dp_{1-2} \,\text{Pa} \\
&= 0 - 17 \,\text{Pa} \\
&= -17 \,\text{Pa} \\
p_{s2} &= p_{t2} - p_{v2} \,\text{Pa} \\
&= -17 - 2.5 \,\text{Pa} \\
&= -20 \,\text{Pa}
\end{aligned}
$$

$$
\begin{aligned}
p_{t3} &= -17 - 450 \,\text{Pa} \\
&= -467 \,\text{Pa} \\
p_{s3} &= -467 - 2.5 \,\text{Pa} \\
&= -470 \,\text{Pa} \\
p_{t10} &= -467 - 0.4 \,\text{Pa} \\
&= -468 \,\text{Pa} \\
p_{s10} &= -468 - 2.5 \,\text{Pa} \\
&= -471 \,\text{Pa} \\
p_{t11} &= -468 - 3.4 \,\text{Pa} \\
&= -471 \,\text{Pa} \\
p_{s11} &= -471 - 84.3 \,\text{Pa} \\
&= -555 \,\text{Pa} \\
p_{t11} &= p_{t11} + \text{fan total pressure rise Pa} \\
&= -471 + 691 \,\text{Pa} \\
&= 220 \,\text{Pa} \\
p_{s11} &= 220 - 84.3 \,\text{Pa} \\
&= 136 \,\text{Pa} \\
p_{t12} &= 220 - 47.2 \,\text{Pa} \\
&= 173 \,\text{Pa} \\
p_{s12} &= 173 - 2.5 \,\text{Pa} \\
&= 171 \,\text{Pa} \\
p_{t13} &= 173 - 2 \times 0.1 \,\text{Pa} \\
&= 173 \,\text{Pa} \\
p_{s13} &= 173 - 2.5 \,\text{Pa} \\
&= 171 \,\text{Pa} \\
p_{t14} &= 173 - 125 \,\text{Pa} \\
&= 48 \,\text{Pa} \\
p_{s14} &= 48 - 2.5 \,\text{Pa} \\
&= 45 \,\text{Pa} \\
p_{t40} &= p_{t14} - \text{duct } dp_{14-15} - \text{grille } dp - \text{discharge } k \times p_{v15} \\
&= 48 - 3 \times 0.1 - 45 - 2.8 \,\text{Pa} \\
&= 0 \,\text{Pa}
\end{aligned}
$$

This is the pressure in the room due to the fan.

Also, as the air velocity leaving the discharge grille has reduced to zero within the room:

$$
\begin{aligned}
p_{s40} &= p_{t40} \\
&= 0 \,\text{Pa}
\end{aligned}
$$

This example is typical of other cases where there are fewer nodes in the ductwork system than are available on the worksheet. The user is required to edit the problem to fit the worksheet and ensure that the worksheet has the relevant data. This will involve renumbering some of the nodes in the example and ignoring some of the facilities of the worksheet. The fan is located between node numbers that need to remain as they are displayed on the worksheet. This is because the formula for calculating the total pressure at the fan outlet is not the same as those for other sections of duct. The user can change the fan location provided that the formulae are also changed. It may be simpler to leave the formulae where they are and change the node numbers on the drawing of the ductwork to suit the worksheet. This is the recommended method. The procedure for entering Example 7.6 into the worksheet file DUCT7%6.WK1 is as follows.

1. The node numbers in Fig. 7.3 have been arranged to suit the worksheet numbers.
2. Describe each duct section in column B.
3. Enter 5 °C into cells C50, C51, C52, C59, C60, C68 and C69.
4. Enter 30 °C into cells C70, C71 and C47. The other temperature cells are not needed in this example, and their values need not be changed.
5. Enter the duct lengths into column E. Make sure that zero length is in the cells where there should not be any duct.
6. The air flow quantity at 5 °C is found from the known value at 30 °C by multiplying by the ratio of the absolute temperatures:

$$Q_1 = Q_{10} \times \frac{273+5}{273+30}\,\frac{\mathrm{m}^3}{\mathrm{s}}$$
$$= 3 \times \frac{273+5}{273+30}\,\frac{\mathrm{m}^3}{\mathrm{s}}$$
$$= 2.753\ \mathrm{m}^3/\mathrm{s}$$

7. Enter 2.753 m^3/s (only the number and not the units) into cells F50, F51, F52, F59, F60, F68 and F69.
8. Enter 3 m^3/s into cells F70, F71 and F110.
9. Rows that have zero duct length and zero fittings velocity pressure loss factor k will always generate zero pressure drop in that duct section.
10. Enter 2.5 m/s for the maximum air velocity in the sections of plant duct and 15 m/s for the fan. Other values may be tried after the first complete calculation.
11. Enter 0.1 Pa/m for the plant duct pressure drop rate.
12. Ignore the Q error in column K, likely duct diameter in column L and velocity in column M, as the duct sizes and pressure drop rate are known.
13. Enter 1200 mm for both the width and depth of the plant ducts in columns N and O.
14. Enter 500 mm for both the width and depth of the fan duct connections, because this produces the correct circular equivalent diameter of 551 mm.
15. Enter the fittings velocity pressure loss factors in the cell range S50 to S110.
16. Enter the plant pressure drops in cell range V50 to V110.
17. This completes the data entry, and the results can be compared with the calculated solution.

DUCTWORK SYSTEM DESIGN

The worksheet can be used for various applications. Each application, worked example and question requires that the ductwork system is identified with node numbers, known air flow rates, and a limiting air velocity for each duct section. Typical velocity limits are given in Table 7.1.

Table 7.1 Limiting air velocities in ducts for low-velocity system design

Application	Main duct v (m/s)	Branch duct v (m/s)
Hospital, concert hall, library, sound studio	5.0	3.5
Cinema, restaurant, hall	7.5	5.0
General office, dance hall, shop, exhibition hall	9.0	6.0
Factory, workshop, canteen	12.5	7.5

The user is required to enter a duct pressure drop rate that produces the least error between the maximum flow capacity of the duct and the design air flow. This is done iteratively, and can produce a satisfactory result quickly. The user selects the appropriate dimensions for circular and rectangular ducts. These are normally in increments of 50 mm.

> EXAMPLE 7.7
>
> Calculate the fan total pressure for the supply air duct system shown in Fig. 7.4. The design air flow is 1 m^3/s with the air at 20 °C d.b. and 101 325 Pa. The heating and cooling coils are in place, but they are not operational. The fresh air intake louvre has a 50% free area, and it is backed with wire mesh. The plant pressure drops are: filter 250 Pa, heater 100 Pa, cooler 100 Pa, and discharge diffuser 25 Pa. The limiting air velocities are: intake and discharge grilles 2.5 m/s, plant 2.5 m/s, and distribution ductwork 5 m/s. Duct and plant dimensions are available in increments of 50 mm. The limiting air velocities are not to be exceeded. Find the smallest sizes of duct that can be achieved within the limits. Use only the dimensions and duct construction shown; do not add contractions or enlargements on the connections to the fan or other items in the system. Investigate the effects of realistic winter and summer operating conditions on the fan total pressure.

The original file DUCT1.WK1 shows the data and solution for this example. The air temperature remains at 20 °C throughout the system. Winter and summer air intake temperatures can range from −5 °C to 40 °C. Operation of the heating and cooling coils will produce supply air temperatures that will range from 30 °C in winter to 15 °C during summer. The nodes in Fig. 7.4 coincide with those in the original file. This leaves a gap between the end of the ductwork at node 12 and the next duct point at node 40. The user may alter the node numbering to suit the application or leave them as shown. The duct fitting velocity pressure loss factors are:

fresh air intake louvre 0.5 free area, $k = 4.5$;
wire mesh, $k = 1.7$;
fresh air intake duct entry, $k = 0.5$;
flat blade damper, $k = 0.5$;

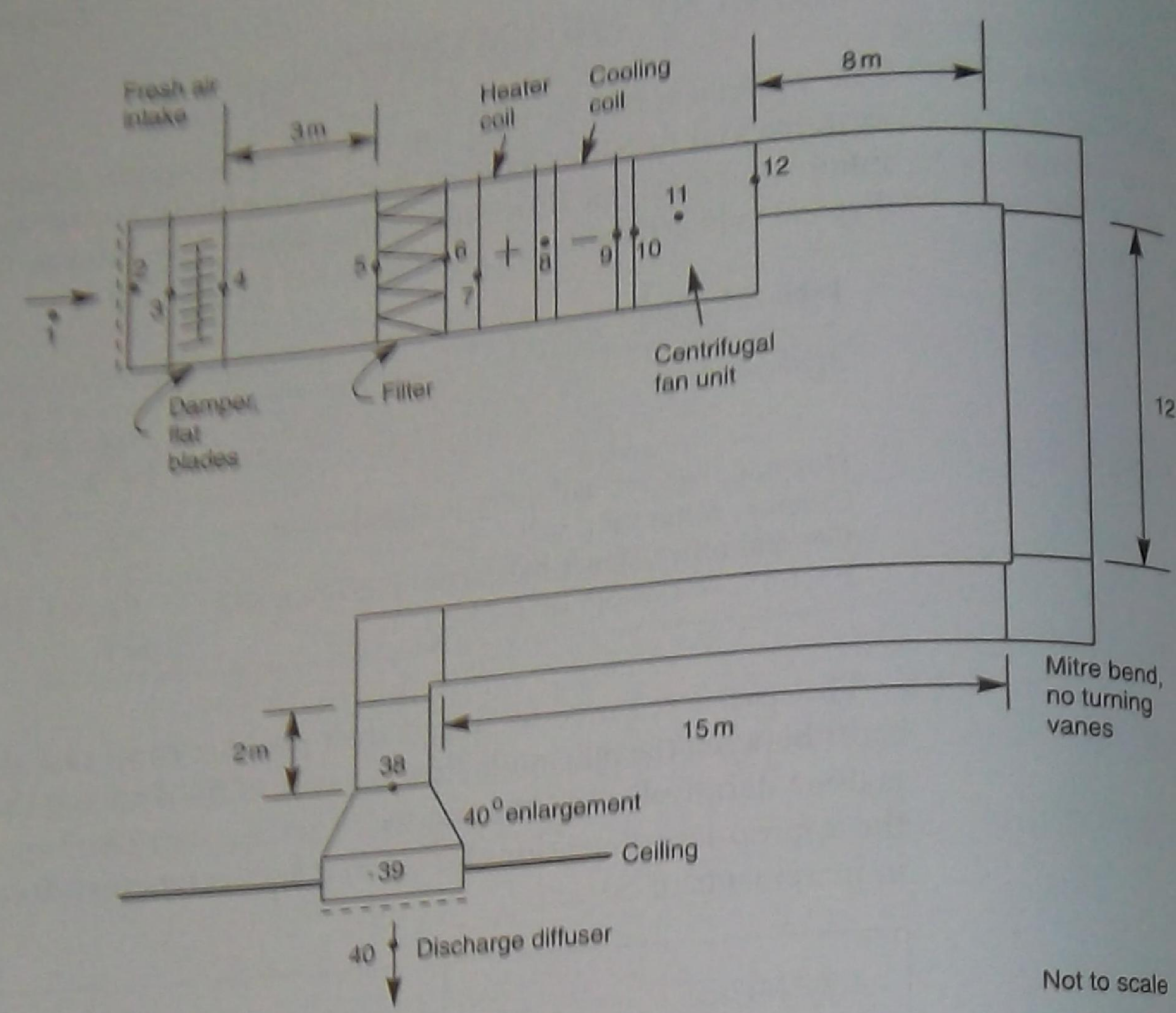


Fig. 7.4 Supply fan ductwork for Example 7.7.

mitre bend with no turning vanes, $k = 1.25$;
40° enlargement from the duct to the discharge diffuser; this will not be known until the duct sizes have been determined because the area ratio between the two ducts is needed.

The density of air at 20 °C d.b. and 101 325 Pa atmospheric pressure is 1.1906 kg/m^3.

An air-handling plant of 650 mm × 650 mm cross-section will have an equivalent circular diameter of

$$d_2 = 1.265 \times \left[\frac{(WD)^3}{W + D}\right]^{0.2} \text{mm}$$

$$= 1.265 \times \left[\frac{(650 \times 650)^3}{650 + 650}\right]^{0.2} \text{mm}$$

$$= 716\,\text{mm}$$

$$= 0.716\,\text{m}$$

The air velocity produced is from

$$Q\frac{\text{m}^3}{\text{s}} = \frac{\pi(d\,\text{m})^2}{4} \times v\frac{\text{m}}{\text{s}}$$

$$v_2 = \frac{4Q}{\pi d_2^2}\frac{\text{m}}{\text{s}}$$

$$= \frac{4 \times 1}{\pi \times 0.716^2}\frac{\text{m}}{\text{s}}$$

$$= 2.5\,\text{m/s}$$

The air velocity pressure is

$$p_{v2} = 0.5\rho v_2^2\,\text{Pa}$$

$$= 0.5 \times 1.1906 \times 2.5^2\,\text{Pa}$$

$$= 3.7\,\text{Pa}$$

The overall velocity pressure loss factor for the outside air inlet louvre is

$$k = 0.5 + 4.5 + 1.7$$

$$= 6.7$$

The pressure drop through the outside air intake louvre is

$$dp_{1-2} = kp_{v2}\,\text{Pa}$$

$$= 6.7 \times 3.7\,\text{Pa}$$

$$= 24.6\,\text{Pa}$$

Pressure drop through the damper is

$$dp_{3-4} = kp_{v3}\,\text{Pa}$$

$$= 0.5 \times 3.7\,\text{Pa}$$

$$= 1.8\,\text{Pa}$$

A pressure drop rate of 0.1 Pa/m of straight duct is appropriate, because it produces an air flow rate of

$$Q = -2.0278 \times dp^{0.5} \times d^{2.5} \times \log\left(\frac{4.05 \times 10^{-5}}{d} + \frac{2.933 \times 10^{-5}}{dp^{0.5} \times d^{1.5}}\right)$$

$$= -2.0278 \times 0.1^{0.5} \times 0.716^{2.5} \times \log\left(\frac{4.05 \times 10^{-5}}{0.716} + \frac{2.933 \times 10^{-5}}{0.1^{0.5} \times 0.716^{1.5}}\right)$$

$$= 1.023\,\text{m}^3/\text{s}$$

The duct pressure drop is

$$dp_{4-5} = \frac{dp}{L}\frac{\text{Pa}}{\text{m}} \times L\,\text{m}$$

$$= 0.1\frac{\text{Pa}}{\text{m}} \times 3\,\text{m}$$

$$= 0.3\,\text{Pa}$$

The total length of duct from the fan outlet to the discharge diffuser is

$$L = 8 + 12 + 15 + 2\,\text{m}$$

$$= 37\,\text{m}$$

The three mitre bends have an overall velocity pressure loss factor of

$$k = 3 \times 1.25$$
$$= 3.75$$

The expected diameter of the distribution duct is

$$d = 1000 \times \left(\frac{Q\,\mathrm{m}^3}{\mathrm{s}} \times \frac{4}{\pi} \times \frac{1}{v}\frac{\mathrm{s}}{\mathrm{m}}\right)^{0.5} \mathrm{mm}$$
$$= 1000 \times \left(\frac{1\,\mathrm{m}^3}{\mathrm{s}} \times \frac{4}{\pi} \times \frac{1}{5}\frac{\mathrm{s}}{\mathrm{m}}\right)^{0.5} \mathrm{mm}$$
$$= 505\,\mathrm{mm}$$

A pressure drop rate of 0.48 Pa/m in the duct between nodes 12 and 38 produces an air flow rate of

$$Q = -2.027 \times 0.48^{0.5} \times 0.505^{2.5} \times \log\left(\frac{4.5 \times 10^{-5}}{0.505} + \frac{2.933 \times 10^{-5}}{0.48^{0.5} \times 0.505^{1.5}}\right)$$
$$= 0.943\,\mathrm{m}^3/\mathrm{s}$$

A 500 mm × 450 mm rectangular duct has an equivalent diameter of

$$d = 1.265 \times \left[\frac{(500 \times 450)^3}{500 + 450}\right]^{0.2} \mathrm{mm}$$
$$= 522\,\mathrm{m}$$
$$v_{12} = 1\,\frac{\mathrm{m}^3}{\mathrm{s}} \times \frac{4}{\pi \times 0.522^2}\,\mathrm{m}^2$$
$$= 4.7\,\mathrm{m/s}$$
$$p_{v12} = 0.5 \times 1.1906 \times 4.7^2\,\mathrm{Pa}$$
$$= 13.2\,\mathrm{Pa}$$

The distribution duct pressure drop is

$$\text{duct } dp_{12-38} = 0.48\,\frac{\mathrm{Pa}}{\mathrm{m}} \times 37\,\mathrm{m}$$
$$= 17.8\,\mathrm{Pa}$$
$$\text{fittings } dp_{12-13} = 3.75 \times 13\,\mathrm{Pa}$$
$$= 48.8\,\mathrm{Pa}$$

The duct enlarges to the same dimensions as the plant, 650 mm × 650 mm, to reduce the air velocity to 2.5 m/s prior to connection to the diffuser.

$$p_v = 3.7\,\mathrm{Pa} \text{ as before.}$$

For an abrupt enlargement:

$$\frac{A_1}{A_2} = \frac{450 \times 450}{650 \times 650}$$
$$= 0.48$$

k_1 may be taken as 0.36, and this is multiplied by the factor for a 40° gradual enlargement, k_2, of 1.

$$\text{Enlargement } k = k_1 k_2$$
$$= 0.36 \times 1$$
$$= 0.36$$

Add the velocity pressure loss factor for the exit from the duct, and the overall becomes

$$k = 1 + 0.36$$
$$= 1.36$$

Pressure drop through the discharge diffuser at the design air flow rate is 25 Pa.

$$\text{fitting } dp_{38-40} = 1.36 \times 3.7 + 25\,\mathrm{Pa}$$
$$= 5 + 25\,\mathrm{Pa}$$
$$= 30\,\mathrm{Pa}$$

The overall drop of pressure through the duct and air-handling plant system is

$$dp_{1-40} = \Sigma dp_{1-40}\,\mathrm{Pa}$$
$$= 24.6 + 1.8 + 0.3 + 250 + 100 + 100 + 17.8 + 48.8 + 5 + 25\,\mathrm{Pa}$$
$$= 573\,\mathrm{Pa}$$

The fan total pressure rise is

$$FTP = dp_{1-40} + p_{v40} - p_{v1}$$
$$= 573 + 3.7 - 3.7\,\mathrm{Pa}$$
$$= 573\,\mathrm{Pa}$$

The fan velocity pressure FVP is, as far as we know in this case, p_{v12} Pa:

$$FVP = 13\,\mathrm{Pa}$$
$$FSP = FTP - FVP\,\mathrm{Pa}$$
$$= 573 - 13\,\mathrm{Pa}$$
$$= 560\,\mathrm{Pa}$$

Check the data on a working copy of file DUCT1.WK1, and save it as file name DUCT7%7.WK1 to represent Example 7.7. Make whatever changes seem appropriate to validate the answers given. There will be other possible solutions. It may be possible to reduce the fan total pressure by selecting other duct sizes. Find out what happens when the winter and summer air temperatures are used for the relevant sections of ductwork. (In summer the fan total pressure becomes 572 Pa, and in winter it is 574 Pa; the differences are not significant.)

The worksheet is arranged for a supply air duct system that has up to five branch ducts. The user may decide to alter the worksheet to suit an extract duct system with five branches. The contents of line numbers 60 and 67, which relate to the fan data, need to be moved down the worksheet until the correct number of suction and discharge branches are either side of the fan.

When a supply and extract duct system is to be analysed, enter the supply air duct system onto the worksheet and make a paper record of the results. This will include the room air pressure and the grille from the outside air. Then enter the data for the extract air duct system and fan from the room to the exhaust air grille. This will overwrite the previous data, which will be lost unless a separate file copy was made. The starting air pressure for the extract system will be the room static air pressure that was created by the supply system. The results for the extract system are then found.

Duct dimensions are quoted as width × depth. Note that the equivalent diameter equation produces a different result when these dimensions become reversed.

If the original spreadsheet does not have enough sections, copy rows down the screen to make more. Edit the new lines to ensure that the correct cell references are maintained, particularly for the air density. New users should beware when making such changes until they fully understand the worksheet.

DATA REQUIREMENT

The user will enter new data for:

1. Your name in cell B4.
2. Name of the job or assignment in cell B5.
3. Reference number or job details in cell B6.
4. Name of the file in cell B9.
5. Duct air temperature in cell C14.
6. Duct air pressure in cell C15.
7. Number all the system nodes on a schematic layout of the ductwork using the numbers that are on the worksheet.
8. Ensure that the section node numbers in column A of the worksheet agree with those on the drawing of the ductwork. There will be occasions when you have to adapt the worksheet node numbers to suit the drawing.
9. Enter the air temperature t °C d.b. in cell C50, duct length L m in cell E50, air flow Q m^3/s in cell F50, maximum air velocity v m/s in cell G50, and the maximum allowable pressure loss rate dp Pa/m in cell H50, and subsequent rows, for each section of duct.
10. The estimated duct diameter and its maximum carrying capacity are displayed in cells I50 and J50.
11. The difference between the maximum carrying capacity and the required Q m^3/s is calculated and expressed as Q error % in cell K50. If this is more than, say, ±5%, enter values for maximum dp Pa/m in cell H50 until close agreement is reached. Use only the first two decimal places for pressure drop rate, because this is sufficiently accurate.
12. Enter the likely duct diameter using increments of 50 mm in cell L50.
13. Enter the dimensions of the rectangular duct to be used in millimetres in cells N50 and O50. The circular equivalent diameter and actual air velocity are displayed in cells P50 and Q50. Change the duct dimensions until a satisfactory result is obtained. Check that the duct dimensions being selected will fit into the building. This is a crucial part of the design.
14. The velocity pressure p_v Pa is displayed in cell R50.
15. Enter the sum of the velocity pressure loss factors k for all the fittings in the duct section in cell S50.
16. Enter the pressure drop through the air-handling plant, such as filters, heating and cooling coils, in the plant dp Pa column in cell V50.

17. Enter the starting total pressure p_t Pa for node 1 in cell X50. This will normally be atmospheric pressure, 0 Pa gauge pressure.
18. Each line is self-contained, and calculations take place whenever new data is entered.
19. The total pressure at each node is displayed in cells X50 and Y50.
20. The static pressure at each node is displayed in cells AA50 and AB50.
21. Enter a line of data for each section of duct.
22. Check that zero is present in the cells for duct length L, air flow quantity Q and fittings velocity pressure loss factor k, where there is no ductwork in that row. Check that zero pressure drop is displayed for that section of ductwork.
23. Save the spreadsheet into a unique filename such as EX003. This preserves the original worksheet for future use.
24. Make the supply air outlet for the main duct route at node number 40 at line 110.
25. Enter the data for branches A to E as for the main route.
26. If the original worksheet does not have enough sections, copy rows down the screen to make more. Edit the new rows to ensure that the correct cell references and formulae are maintained. New users should be careful when making such changes (to a copy of the original file).

OUTPUT DATA

1. The current date is given in cell B7.
2. The current time is given in cell B8.
3. The air density for the temperature and pressure is stated in cells C14 and C15.
4. The air density given in cell D50 is that to be used in line number 50.
5. The first estimate of duct diameter is given in cell I50.
6. The maximum carrying capacity of the estimated duct diameter is given in cell J50.
7. The difference between the design air flow and the maximum carrying capacity of the duct is expressed as a percentage of the design flow in cell K50. This percentage is to be minimized by changing the maximum pressure drop rate in cell H50 until either zero error has been achieved, or as small an error as possible is obtained while using only two decimal places for the pressure drop. An error of less than ±1% is normally achievable. An error of ±5% should not be exceeded.
8. The air velocity that will be produced in the likely duct diameter from cell L50 is shown in cell M50.
9. The equivalent diameter of the rectangular duct selected in cells N50 and O50 is shown in cell P50.
10. The air velocity of the duct selected is shown in cell Q50.
11. The air velocity pressure is given in cell R50.
12. The air pressure drop through the straight duct is given in cell T50.
13. The air pressure drop through the duct fittings is given in cell U50.
14. The frictional resistance of the section of duct is given in cell W50.
15. The total air pressure at the commencement of the duct is given in cell X50.
16. The total air pressure at the end of the duct is given in cell Y50.
17. The static air pressure at the commencement of the duct is given in cell AA50.

18. The static air pressure at the end of the duct is given in cell AB50.
19. The air velocity pressure is repeated in cell AC50 from cell R50 for information only.
20. The node numbers and the description of the duct section are repeated in cells AE50 and AF50 for information.
21. The same information is produced for each line of duct section and branch.
22. The pressure drop through the main run of ductwork between node 1 and node 40 is shown in cell W112. It is not necessary to read this cell.
23. The air pressure drop through each section of duct is displayed in the cell range C228 to C238.
24. The pressure drop through the duct route from nodes 1 to 40 is displayed in cell D233.
25. The pressure drop from node 1 to the node at the end of each branch duct is shown in the cell range D234 to D238.
26. When the pressure drop through a branch duct is less than that which is available at the commencement of the branch, the excess pressure is displayed in cells E233 to E38. This is the pressure drop that is needed across the volume control damper in order to remove the excess air pressure.
27. Cells F233 to F238 display the duct route that has the highest resistance to air flow. This is the index route. The word 'index' is displayed.
28. The air pressure drop through the index duct route is given in cell D240.
29. The index route of duct nodes is given in cell F240.
30. The fan total pressure is given in cell E245.
31. The fan velocity pressure is given in cell E246.
32. The fan static pressure is given in cell E247.
33. The air volume flow rate through the fan is given in cell E248.
34. The air temperature at the fan is given in cell E249.
35. The air density at the fan is given in cell E250.

FORMULAE

Cell B7

@TODAY

The cell is formatted for the @TODAY function to display the current date.

Cell B8

@TODAY

The cell is formatted for the @TODAY function to display the current time.

Cell C17

1.1906*293*C15/101325/(273+C14)

The density of the air is calculated. This density is for reference only.

$$\text{density, } \rho = 1.1906\,\frac{\text{kg}}{\text{m}^3} \times \frac{(273+20)\text{K}}{(273+t)\text{K}} \times \frac{b\,\text{Pa}}{101\,325\,\text{Pa}}$$

where b = barometric air pressure (Pa), and t = air temperature (°C d.b.).

$$\text{C17} = 1.1906\,\frac{293}{273+\text{C14}} \times \frac{\text{C15}}{101325}\,\frac{\text{kg}}{\text{m}^3}$$

Cell D50

1.1906*293*$C$15/101325/(273+C50)

The air density to be used in that section of duct is calculated for each line of the worksheet. The standard atmospheric pressure is taken from cell C15, because this would normally be common to all the duct system.

$$\text{D50} = 1.1906\,\frac{293}{273+\text{C50}} \times \frac{\$\text{C}\$15}{101325}\,\frac{\text{kg}}{\text{m}^3}$$

Cell I50

1000*@SQRT(F50*4/G50/@PI)

Finds the duct diameter to carry the design air flow.

$$d = 1000 \times \sqrt{\left(\frac{4Q}{\pi v}\right)}\ \text{mm}$$

$$\text{I50} = 1000 \times \sqrt{\left(\frac{4 \times \text{F50}}{\pi \times \text{G50}}\right)}\ \text{mm}$$

> EXAMPLE 7.8
>
> Calculate the diameter of an air duct that will pass $2.25\,\text{m}^3/\text{s}$ of air at a velocity that does not exceed 5 m/s.

$$d = 1000 \times \sqrt{\left(\frac{4Q}{\pi v}\right)}\ \text{mm}$$

$$Q = 2.25\,\text{m}^3/\text{s}$$

$$v = 5\,\text{m/s}$$

$$d = 1000 \times \sqrt{\left(\frac{4 \times 2.25}{\pi \times 5}\right)}\ \text{mm}$$

$$= 757\,\text{mm}$$

Cell J50

−2.027*SQRT(H50)*(I50)^2.5*LOG(((4.05*10^−5)/I50)+(2.933*10^−5)/SQRT(H50)/I50^1.5)

This equation calculates the maximum carrying capacity of an air duct from the duct diameter d m, and the pressure drop rate dp Pa/m. The flow of air at 20 °C d.b., 43% percentage saturation and a barometric pressure of 1013.25 mbar through clean galvanized sheet metal ducts having joints made in accordance with good practice is given by

$$Q = -2.0278 \times dp^{0.5} \times d^{2.5} \times \log\left(\frac{4.05 \times 10^{-5}}{d} + \frac{2.933 \times 10^{-5}}{dp^{0.5} \times d^{1.5}}\right)$$

where Q = air flow rate (m^3/s); dp = pressure loss rate (Pa/m) (cell H50); and d = duct internal diameter (m) (cell I50).

$$J50 = -2.0278 \times (H50)^{0.5} \times (I50)^{2.5} \times \log\left[\frac{4.05 \times 10^{-5}}{I50} + \frac{2.933 \times 10^{-5}}{(H50)^{0.5} \times (I50)^{1.5}}\right]$$

> EXAMPLE 7.9
>
> Find the maximum air-carrying capacity and air velocity in a 350 mm diameter galvanized sheet steel duct at 20 °C d.b. when the pressure loss rate is 0.85 Pa/m.

$$d = 0.35\,\text{m}$$
$$dp = 0.85\,\text{Pa/m}$$
$$Q = \text{air flow (m}^3\text{/s)}$$
$$Q = -2.0278 \times dp^{0.5} \times d^{2.5} \times \log\left(\frac{4.05 \times 10^{-5}}{d} + \frac{2.933 \times 10^{-5}}{dp^{0.5} \times d^{1.5}}\right)$$
$$= -2.0278 \times 0.85^{0.5} \times 0.35^{2.5} \times \log\left(\frac{4.05 \times 10^{-5}}{0.35} + \frac{2.933 \times 10^{-5}}{0.85^{0.5} \times 0.35^{1.5}}\right)$$
$$= -0.1355 \times \log(2.6935 \times 10^{-4})$$
$$= -0.1355 \times -3.57$$
$$= 0.484\,\text{m}^3\text{/s}$$

> EXAMPLE 7.10
>
> A 500 mm internal diameter galvanized sheet steel duct is carrying air at 20 °C d.b. at a design pressure loss rate of 1.0 Pa/m. Calculate the air volume flow rate being passed and the air velocity.

$$d = 0.50\,\text{m}, \quad dp = 1.0\,\text{Pa/m}$$
$$Q = -2.0278 \times 1.0^{0.5} \times 0.50^{2.5} \times \log\left(\frac{4.05 \times 10^{-5}}{0.50} + \frac{2.933 \times 10^{-5}}{1.0^{0.5} \times 0.50^{1.5}}\right)$$
$$= 1.357\,\text{m}^3\text{/s}$$

$$\text{Air velocity } v = 1.357\frac{\text{m}^3}{\text{s}} \times \frac{4}{\pi \times 0.5^2\,\text{m}^2}$$
$$= 6.91\,\text{m/s}$$

Cell K50

(F50−J50)*100/F50

Calculates the percentage error between the design air flow Q m^3/s and the maximum carrying capacity of the duct Q_1 m^3/s at the current air pressure drop rate dp/l Pa/m. The user enters new values for the duct air pressure drop rate to minimize the error. When the user is satisfied with the error, the current value of air pressure drop rate is used to calculate the air pressure drop along the straight ducts.

$$\text{error} = \frac{Q - Q_1}{Q} \times 100\%$$
$$K50 = \frac{F50 - J50}{F50} \times 100\%$$

Cell M50

+F50*4/@PI/(L50/1000)^2

Calculates the air velocity that will be produced for the duct diameter shown in cell L50.

$$\text{air velocity } v = Q\frac{\text{m}^3}{\text{s}} \times \frac{4}{\pi d^2\,\text{m}^2}$$
$$M50 = F50\frac{\text{m}^3}{\text{s}} \times \frac{4}{\pi} \times \left(\frac{1000}{L50}\right)^2 \frac{1}{\text{m}^2}$$

Cell P50

1.265*((N50*O50)^3/(N50+O50))^0.2

Calculates the equivalent circular diameter of the rectangular duct selected in cells N50 and O50 for equal volume, pressure drop rate and surface roughness (CIBSE, 1986c).

$$d = 1.265 \times \left[\frac{(ab)^3}{a+b}\right]^{0.2}\,\text{mm}$$

where a = duct width (mm); b = duct depth (mm); and d = equivalent duct diameter (mm).

$$P50 = 1.265 \times \left[\frac{(N50 \times O50)^3}{N50 + O50}\right]^{0.2}\,\text{mm}$$

Cell Q50

+F50*4/@PI/(P50/1000)^2

Calculates the air velocity in the circular equivalent of the rectangular duct selected.

$$\text{air velocity } v = Q\frac{m^3}{s} \times \frac{4}{\pi d^2\,m^2}$$

$$M50 = F50\frac{m^3}{s} \times \frac{4}{\pi} \times \left(\frac{1000}{P50}\right)^2 \frac{1}{m^2}$$

EXAMPLE 7.11

A 700 mm wide and 600 mm deep galvanized steel air duct is to carry 2200 l/s of air at 20 °C d.b. at a maximum velocity of 6 m/s and at a pressure loss rate not exceeding 0.72 Pa/m. Calculate the equivalent diameter of the duct, the air velocity, and the error between the design air flow and the maximum air-carrying capacity of the duct.

$$\text{design air flow } Q = 2.2\,m^3/s$$

The equivalent diameter of a 700 mm × 600 mm duct is

$$d = 1.265 \times \left[\frac{(ab)^3}{a+b}\right]^{0.2} \text{mm}$$

$$= 1.265 \times \left[\frac{(700 \times 600)^3}{700 + 600}\right]^{0.2} \text{mm}$$

$$= 713\,\text{mm}$$

The air velocity in the duct is

$$v = Q\frac{m^3}{s} \times \frac{4}{\pi d^2\,m^2}$$

$$= 2.2\frac{m^3}{s} \times \frac{4}{\pi 0.713^2\,m^2}$$

$$= 5.51\,m/s$$

The maximum air-carrying capacity of a 713 mm diameter duct at a pressure drop rate of 0.72 Pa/m is

$$Q_1 = -2.0278 \times dp^{0.5} \times d^{2.5} \times \log\left(\frac{4.05 \times 10^{-5}}{d} + \frac{2.933 \times 10^{-5}}{dp^{0.5} \times d^{1.5}}\right)$$

$$= -2.0278 \times 0.72^{0.5} \times 0.713^{2.5} \times \log\left(\frac{4.05 \times 10^{-5}}{0.713} + \frac{2.933 \times 10^{-5}}{0.72^{0.5} \times 0.713^{1.5}}\right)$$

$$= -0.7386 \times \log(1.1422 \times 10^{-4})$$

$$= 2.912\,m^3/s$$

$$\text{error} = \frac{Q - Q_1}{Q} \times 100\%$$

$$= \frac{2.2 - 2.91}{2.2} \times 100\%$$

$$= -32.3\%$$

It can be seen that the duct is carrying 32.3% less than its maximum capacity; this is indicated by the negative result. A pressure drop rate of 0.43 Pa/m produces a maximum carrying capacity of 2.22 m³/s and an error of −0.9%; this is as close as can be found from an accuracy of two decimal places.

Cell R50

0.5*D50*Q50*Q50

The air velocity pressure p_v Pa is given by

$$p_v = 0.5\rho v^2 \text{ Pa}$$

$$R50 = 0.5 \times D50 \times Q50 \times Q50 \text{ Pa}$$

D50 density kg/m³

Q50 velocity m/s

Cell T50

+E50*H50

Air pressure drop dp_1 Pa in the straight duct length is

$$dp_1 = dp\frac{Pa}{m} \times L\,m$$

$$T50 = H50\frac{Pa}{m} \times E50m$$

Cell U50

+S50*R50

Air pressure drop through the duct fittings dp_2 Pa is

$$dp_2 = kp_v \text{ Pa}$$

$$R59 = S50 \times R50 \text{ Pa}$$

Cell W50

+T50+U50+V50

The sum of the duct pressure drop dp_1, duct fittings resistance dp_2 and the plant pressure drop dp_3 for that section of duct is

$$dp = dp_1 + dp_2 + dp_3 \text{ Pa}$$

$$W50 = T50 + U50 + V50 \text{ Pa}$$

Cell X51

+Y50

The total air pressure at the commencement of the duct section. This is normally zero for node 1. Node 2 has a total pressure that is equal to the total air pressure at the end of section 1. The content of the previous duct end total pressure is copied into cell X51.

Cell Y50

+X50−W50

The air total pressure at the end of this section of duct is equal to the total pressure at the beginning of the duct less the frictional resistance of the section:

$$p_{t2} = p_{t1} - dp_{1-2} \text{ Pa}$$
$$\text{Y50} = \text{X50} - \text{W50 Pa}$$

Cell AA50

+AB50

The static air pressure at the start of the duct section, p_s Pa, is either:

1. zero, if this is the first node in the duct system and is at atmospheric air pressure; or
2. the room air static pressure, for example up to ±50 Pa depending upon how the space is pressurized; or
3. equal to the static air pressure at the end of the preceding section of duct.

Cell AB50

+Y50−R50

The static air pressure p_s Pa at the end of the duct is

$$p_{s2} = p_{t2} - p_{v2} \text{ Pa}$$
$$\text{AB50} = \text{Y50} - \text{R50 Pa}$$

Cell AC50

+R50

A copy of the velocity pressure in the section for information only.

Cell AE50

+A50

Copies the node numbers that are used for the duct section.

Cell AF50

+B50

Copies the section description for reference only.

Cell Y60

+X60+D240

The total air pressure at the air discharge from the fan p_{t2} is the total pressure at the fan inlet node p_{t1} plus the fan total pressure rise FTP.

p_{t1} = total pressure at fan inlet Pa
FTP = fan total pressure rise Pa
= loss of total pressure through the ductwork d_p Pa

$$p_{t2} = p_{t1} + FTP \text{ Pa}$$
$$\text{Y60} = \text{X60} + \text{D240 Pa}$$

Cell Y110

+X110−W110+R110

The total pressure of the air that is discharged from the air outlet grille or diffuser at the end of the duct route into the air conditioned or ventilated space is equal to the total pressure at the final duct node minus the pressure drop through the grille and minus one velocity pressure. There is one velocity pressure loss factor for the discharge of air from a duct. This produces a final total pressure of zero in the ventilated space; this is not correct, because the discharge air velocity pressure is produced by the total pressure. It is the final static air pressure that is zero, or whatever is held within the ventilated space. The final total air pressure must contain the discharge air velocity and zero, or other value, static pressure. The final velocity pressure is added back into the total pressure because it was taken out by the velocity pressure loss factor.

$$p_{t2} = p_{t1} - d_{p1-2} + p_{v2} \text{ Pa}$$
$$\text{Y110} = \text{X110} - \text{W110} + \text{R110 Pa}$$

> EXAMPLE 7.12
>
> The total pressure at node 1 in a duct system is 235 Pa above the atmospheric air pressure. The duct is 25 m long, 600 mm in diameter, and carries 1400 l/s of air at a density of 1.2 kg/m^3. The air pressure loss rate through the straight duct is 0.5 Pa/m, and the duct fittings have a combined velocity pressure loss factor of 3.6. The duct has an air filter and a heating coil, which have a combined resistance to air flow of 125 Pa. Calculate the total and static air pressures at the end of the duct.

$$p_{t1} = 235 \text{ Pa}$$

The air velocity in the duct is

$$\begin{aligned} v_1 &= Q\frac{\text{m}^3}{\text{s}} \times \frac{4}{\pi d^2\ \text{m}^2} \\ &= 1.4\frac{\text{m}^3}{\text{s}} \times \frac{4}{\pi\, 0.6^2\ \text{m}^2} \\ &= 4.95 \text{ m/s} \\ p_{v1} &= 0.5\rho v_1^2 \text{ Pa} \\ &= 0.5 \times 1.2 \times 4.95^2 \text{ Pa} \\ &= 15 \text{ Pa} \end{aligned}$$

$$dp_{1-2} = dp\frac{\text{Pa}}{\text{m}} \times L\,\text{m}$$
$$= 0.5\frac{\text{Pa}}{\text{m}} \times 25\,\text{m}$$
$$= 13\,\text{Pa}$$

Air pressure drop through the duct fittings:

$$dp = kp_{v1}\,\text{Pa}$$
$$= 3.6 \times 15\,\text{Pa}$$
$$= 54\,\text{Pa}$$

Loss of total pressure through the duct:

$$dp_{1-2} = (\text{duct} + \text{fittings} + \text{plant})\,\text{Pa}$$
$$= 13 + 54 + 125\,\text{Pa}$$
$$= 192\,\text{Pa}$$

Total pressure at node 2:

$$p_{t2} = p_{t1} - dp_{1-2}\,\text{Pa}$$
$$= 235 - 192\,\text{Pa}$$
$$= 43\,\text{Pa}$$
$$p_{s1} = p_{t1} - p_{v1}\,\text{Pa}$$
$$= 235 - 15\,\text{Pa}$$
$$= 220\,\text{Pa}$$
$$p_{s2} = p_{t2} - p_{v2}\,\text{Pa}$$
$$= 43 - 15\,\text{Pa}$$
$$= 28\,\text{Pa}$$

EXAMPLE 7.13

Part of a ventilation duct system has three nodes. Node 1 is within the duct system, some distance before node 2, which is at the entry into a discharge air diffuser. The ventilated room is maintained at atmospheric air pressure. Node 3 is in the ventilated room after the diffuser. The total pressure at node 1 in the duct system is 115 Pa above the atmospheric air pressure. The duct has pressure losses that amount to 80 Pa between nodes 1 and 2. The air velocity at node 1 is 6 m/s and that at node 2 is 3 m/s. The velocity pressure loss factor for the discharge of air from the end of the duct, k, is 1, and the diffuser has a resistance of 30 Pa. The air velocity in the room is zero. The air density is $1.2\,\text{kg/m}^3$. Calculate the total and static air pressures at the end of the duct.

Sketch the duct system, identify the nodes and sketch the pressure gradients.

$$p_{t1} = 115\,\text{Pa}$$
$$v_1 = 6\,\text{m/s}$$
$$p_{v1} = 0.5 \times 1.2 \times 6^2\,\text{Pa}$$
$$= 22\,\text{Pa}$$

$$dp_{1-2} = 80\,\text{Pa}$$
$$p_{t2} = p_{t1} - dp_{1-2}\,\text{Pa}$$
$$= 115 - 80\,\text{Pa}$$
$$= 35\,\text{Pa}$$
$$p_{s1} = p_{t1} - p_{v1}\,\text{Pa}$$
$$= 115 - 22\,\text{Pa}$$
$$= 93\,\text{Pa}$$
$$v_2 = 3\,\text{m/s}$$
$$v_3 = 3\,\text{m/s}$$
$$p_{v2} = 0.5 \times 1.2 \times 3^2\,\text{Pa}$$
$$= 5\,\text{Pa}$$
$$p_{v3} = 5\,\text{Pa}$$
$$p_{s2} = p_{t2} - p_{v2}\,\text{Pa}$$
$$= 35 - 5\,\text{Pa}$$
$$= 30\,\text{Pa}$$

Air pressure drop through the diffuser and due to the discharge of air from the end of the duct is

$$dp_{2-3} = 30 + kp_{v2}\,\text{Pa}$$
$$= 30 + 1 \times 5\,\text{Pa}$$
$$= 35\,\text{Pa}$$

The final air velocity on the discharge side, ignoring the diffusing air stream velocity within the room, is that through the diffuser:

$$v_3 = 3\,\text{m/s}$$
$$p_{v3} = 5\,\text{Pa}$$

The air pressure loss from node 2 to node 3 includes the discharge air velocity pressure. The total pressure of the discharge air is the velocity pressure p_{v3}, and it is not zero, so:

$$p_{t3} = p_{t2} - dp_{2-3} + p_{v3}\,\text{Pa}$$
$$= 35 - 35 + 5\,\text{Pa}$$
$$= 5\,\text{Pa}$$
$$p_{s3} = p_{t3} - p_{v3}\,\text{Pa}$$
$$= 5 - 5\,\text{Pa}$$
$$= 0\,\text{Pa}$$

Cell C228

@SUM(W50..W79)

The air pressure drop from node 1 to node 23 is found by the addition of the drop of total pressure in each section. This calculation is repeated for the other sections of duct in the main route from node 23 to the final discharge of air into the ventilated space at node 40, in cells C229 to C233.

8 Water pipe sizing

INTRODUCTION

This chapter uses the worksheet file PIPE.WKS to find the sizes of pipes and the pump specification for two-pipe heating hot water and chilled water circulation systems. The user assesses which is the index circuit by inspection of the pipework layout on the scale drawing of the building. The heat load and pipe length are entered onto the worksheet for each section of the index route and then for the branches. A few iterations of the value for the pressure drop rate, as is done during manual design, produce a suitable pipe diameter for the section.

Data for a range of different pipe materials is provided on the worksheet. The user copies the required data into a staging area on the worksheet. Additional pipe data may be entered by the user for any material or range of diameters. An initial estimate is entered for the pipe heat emission. The worksheet lists the lengths of all the specified pipe diameters, and calculates the actual heat emission and the total installed cost of the pipe system. The designer can then change the input data to try different options to minimize the cost of the pipework. Such redesign would often be avoided during manual design because of the time involved in recalculation. The worksheet is a powerful design tool, which finds pipe sizes and the pump duty with the minimum of data entry, and also generates a list of pipe lengths and cost for the estimator.

LEARNING OBJECTIVES

Study and use of this chapter will enable the user to:

1. understand how to find the sizes of a two-pipe water heating or cooling system;
2. know how to calculate the pump performance specification for a two-pipe system;
3. know the use of the term index circuit;
4. calculate the water flow rates that are required for a two-pipe system;
5. find the heat emission from a pipe system;
6. use the hydraulic resistance of items of plant;
7. use the equations for water flow in pipes;
8. use water density, dynamic viscosity, pipe surface roughness and pipe internal diameter in pipe sizing calculations;

9. use velocity pressure loss factors in the calculation of pump pressure rise;
10. know what is meant by the equivalent length of a pipeline system;
11. know how to maintain a convenient source of pipe sizing data on disk;
12. know how to calculate pipe systems by manual methods;
13. calculate the installed cost of 1 m of a pipeline system from updated price data;
14. find the total installed cost of a two-pipe system;
15. use the total installed cost of the pipe system as part of the design decision work.

Key tems and concepts

TWO-PIPE CIRCULATION SYSTEM

Most heating hot water and chilled water circulation systems are of the two-pipe flow and return design. This worksheet is not used for single-pipe systems. A typical two-pipe reticulation system is shown in Fig. 8.1. A numbering or lettering code is used for each section of pipe. A format block is shown in Fig. 8.1 to label each pipe section and to allow for the recording of the output data on the schematic drawing.

The worksheet has lines of data for 20 pipes along the index circuit and for 20 branch pipes. More pipe sections can be added by the user to enlarge the worksheet when necessary for a particular project. The worksheet should not be enlarged without a good reason. Move the cursor down a screen page at a time using the Page Down, Page Up and Home cursor control keys.

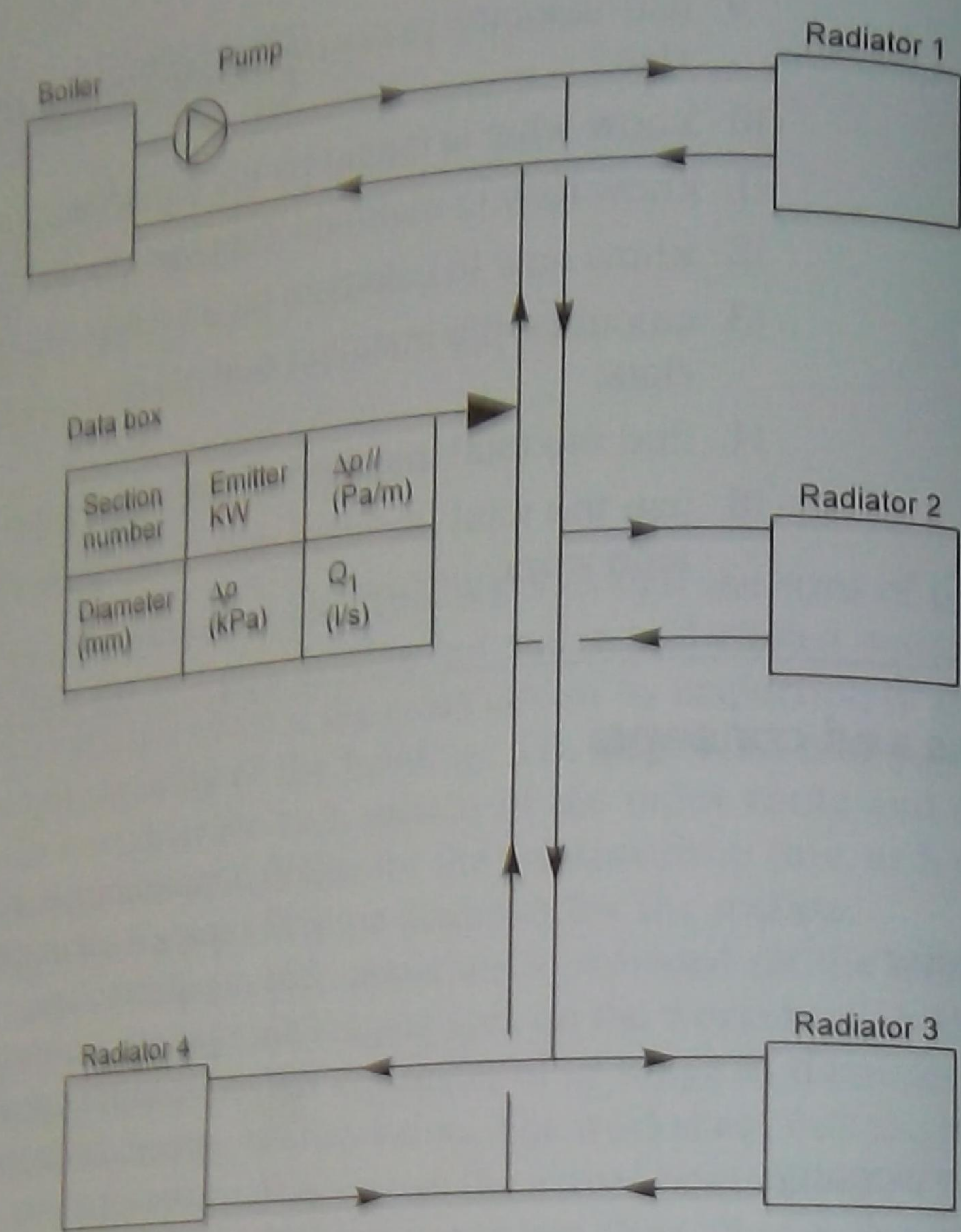


Fig. 8.1 Typical two-pipe system schematic.

The heat load at each terminal heat emitter is found from the heat loss and heat gain worksheets in conjunction with the designer's further calculations and system decisions. The heat loads are summed from the heat emitters, along the pipe routes, back to the heat source. The heat source will be a heat exchanger, boiler or water-chilling machine. The worksheet calculates the water flow rate required for each section of pipe, and adds the specified percentage for the heat lost from the pipework. Designers will usually overestimate the pipe heat emission. The worksheet calculates the percentage heat emission that is produced by the design. The user enters the correct value of percentage heat emission and checks the pipe and pump sizes again. The hydraulic resistance of all the plant and pipeline fittings is entered, as would be done with manual calculation methods. For cooling systems, the term 'heat emission' refers to the chilled water cooling load at the terminal unit and heat gains to the chilled water circulation system.

WATER FLOW EQUATIONS

The water flow rate that is required to provide the heat load is given by

$$Q_1 = H\,\text{kW} \times \frac{1}{SHC}\frac{\text{kg K}}{\text{kJ}} \times \frac{1}{T_f - T_r} \times \frac{100 + L\%}{100} \times \frac{1\,\text{kJ}}{1\,\text{kW s}}$$

$$= \frac{H(100 + L)}{SHC(T_f - T_r) \times 100}\,\frac{\text{kg}}{\text{s}}$$

$$= \frac{H(100 + L)}{SHC(T_f - T_r) \times 100}\,\frac{\text{kg}}{\text{s}} \times \frac{1}{\rho}\,\frac{\text{m}^3}{\text{kg}} \times \frac{10^3\,\text{litre}}{1\,\text{m}^3}$$

$$= \frac{H(100 + L) \times 10}{SHC(T_f - T_r) \times \rho}\,\frac{1}{\text{s}}$$

The pipe diameter that is needed for a specified water velocity is found from

$$Q_1 = \frac{\pi d^2}{4}\,\text{m}^2 \times v\,\frac{\text{m}}{\text{s}} \times \frac{10^3\,\text{litre}}{1\,\text{m}^3}$$

$$d^2 = \frac{4Q_1}{\pi v \times 1000}\,\text{m}^2$$

$$d = \left(\frac{4Q_1}{\pi v \times 1000}\right)^{0.5}\text{m} \times \frac{10^3\,\text{mm}}{1\,\text{m}}$$

where H = heat emitter load (kW); Q_1 = water flow rate required (l/s); SHC = specific heat capacity of water, 4.186 kJ/kg K; T_f = water flow temperature at heat emitter (°C); T_r = water return temperature leaving heat emitter (°C); L = estimated pipe heat emission (%); ρ = density of water (kg/m^3); and d = pipe diameter (m).

The turbulent flow of water in a pipe is given by (CIBSE, 1986c)

$$M = -\pi\rho\left(\frac{dp_1 \times d^5}{2\rho}\right)^{0.5} \times \log_{10}\left[\frac{k_s}{3.7d} + \frac{1.255\mu}{\rho} \times \left(\frac{2\rho}{dp_1 \times d^3}\right)^{0.5}\right]$$

where M = water flow rate (kg/s); dp_1 = water pressure drop rate (Pa/m); d = pipe internal diameter (m); μ = dynamic viscosity of water (Pa s); and k_s = absolute surface roughness of pipe (m).

This equation is the basis of the published data tables. It would not normally be used by designers, but is easily handled on a spreadsheet. An example is given here, and also a question that requires its use by manual calculation. These are the only occasions on which the user sees the equation. The water density and viscosity data used on the worksheet are given in Table 8.1. The absolute roughness for pipes is given in Table 8.2. Table 8.3 shows the range of nominal pipe diameters and actual internal diameters for copper pipe Table X, BS 2871 : Part 1.

Table 8.1 Density and viscosity for water (Rogers and Mayhew, 1987)

Temperature (°C)	*Density* (kg/m^3)	*Dynamic viscosity* (mPa s)
10	999.7	1.300
60	983.3	0.463
70	977.5	0.400
75	974.7	0.374
80	971.8	0.351
85	969.0	0.330
90	965.3	0.311
100	957.9	0.279
120	943.4	0.230
130	934.6	0.211
140	925.9	0.195
150	916.6	0.181

Table 8.4 shows the similar data for medium black mild steel pipes to BS 1387. Table 8.5 shows typical values for the velocity pressure loss factors for pipe fittings in the range of diameters from 32 mm to 50 mm. Further data may be added to this table as required. This, and other, reference data is repeated in the data bank section in the worksheet.

Table 8.2 Absolute roughness for pipes

Material	*Roughness, k_s* (mm)
Copper	0.0015
Plastic	0.0030
New black mild steel	0.0460
Rusted black mild steel	2.5000
Galvanized mild steel	0.1500

Table 8.3 Copper pipe Table X, BS 2871 : Part 1

Nominal diameter, d (mm)	*Internal diameter, d* (mm)
15	13.60
22	20.22
28	26.22
35	32.63
42	39.63
54	51.63
67	64.27
76	73.22
108	105.12
133	130.38
159	155.38

Table 8.4 Medium black mild steel BS 1387

Nominal diameter, d (mm)	*Internal diameter, d* (mm)
15	16.10
20	21.60
25	27.30
32	36.00
40	41.90
50	53.00
65	68.70
80	80.70
90	93.15
100	105.10
125	129.95
150	155.40

Table 8.5 Velocity pressure loss factors for pipe fittings

Pipe fitting	*k factor*
Copper pipe tee branch	1.35
25 mm branch equal sweep tee	1
Mild steel tee branch	1.05
25 mm screwed mild steel bend	0.7
25 mm copper pipe elbow	1
Angle radiator valve	5
Steel panel radiator	2.5
Cast iron boiler	2.5
Fire tube horizontal boiler	8
25 mm open gate valve	0.3

EXAMPLE 8.1

A 35 mm Table X copper pipe passes water at 75 °C at a pressure drop rate of 200 Pa/m. Calculate the maximum water flow rate that the pipe can carry.

$$M = -\pi\rho\left(\frac{dp_1 \times d^5}{2\rho}\right)^{0.5} \times \log_{10}\left[\frac{k_s}{3.7d} + \frac{1.255\mu}{\rho} \times \left(\frac{2\rho}{dp_1 \times d^3}\right)^{0.5}\right]$$

From Table 8.1, at 75 °C, $\rho = 974.85\ \text{kg/m}^3$, $\mu = 0.378$ mPa s. From Table 8.2, for a 35 mm Table X copper pipe, $d = 32.63$ mm. From Table 8.3, for copper pipe $k_s = 0.0015$ mm.

$$\begin{aligned}
\rho &= 974.85\ \text{kg/m}^3\\
\mu &= 0.378\ \text{mPa s}\\
&= 0.378 \times 10^{-3}\ \text{Pa s}\\
d &= 32.63\ \text{m}\\
&= 0.032\,63\ \text{m}\\
dp_1 &= 200\ \text{Pa/m}\\
k_s &= 0.0015\ \text{mm}\\
&= 0.0015 \times 10^{-3}\ \text{m}
\end{aligned}$$

$$M = \rho\pi\left(\frac{dp_1 \times d^5}{2\rho}\right)^{0.5} \times \log_{10}\left[\frac{k_s}{3.7d} + \frac{1.255\,\mu}{\rho} \times \left(\frac{2\rho}{dp_1 \times d^3}\right)^{0.5}\right]$$

Arrange as

$$M = -\rho\pi\left(\frac{dp_1 \times d^5}{2\rho}\right)^{0.5} \times L$$

where

$$L = \log_{10}\left[\frac{k_s}{3.7d} + \frac{1.255\,\mu}{\rho} \times \left(\frac{2\rho}{dp_1 \times d^3}\right)^{0.5}\right]$$

$$= \log_{10}\left[\frac{0.0015 \times 10^{-3}}{3.7 \times 0.03263} + \frac{1.255 \times 0.378 \times 10^{-3}}{974.85} \times \left(\frac{2 \times 974.85}{(200 \times 0.03263)^3}\right)^{0.5}\right]$$

$$= \log_{10}(1.242 \times 10^{-5} + 4.866 \times 10^{-7} \times 529.716)$$
$$= \log_{10}(2.7018 \times 10^{-4})$$
$$= -3.568$$

$$M = \rho\pi \left(\frac{dp_1 \times d^5}{2\rho} \right)^{0.5} \times L$$

$$M = -974.85 \times \pi \times \left(\frac{200 \times 0.032\,63^5}{2 \times 974.85} \right)^{0.5} \times -3.568 \text{ kg/s}$$
$$= 0.673 \text{ kg/s}$$

This is the maximum carrying capacity of the pipe at the specified pressure drop rate of 200 Pa/m. The required water flow rate is compared with the maximum carrying capacity, Q_2 l/s, to establish the difference, or error:

$$\text{flow error } E = \frac{Q_1 - Q_2}{Q_1} \times 100\%$$

The user enters successive values for the water pressure drop rate until the error is less than ±1%. The final value of the water pressure drop rate, dp_1 Pa/m, is that used to calculate the system pressure drop and the pump pressure increase. The error percentage should be as small as can be achieved. Increments of 5 Pa/m for the pressure drop rate should be the minimum.

The volume flow rate of water through the pipe system is taken as the maximum value. This is because the user has found the nearest pressure drop rate to satisfy the design criteria. This can be done by manual calculation and inspection of published pipe sizing tables. The water volume flow rate is

$$Q_2 = M \frac{\text{kg}}{\text{s}} \times \frac{10^3 \text{ litre}}{1 \text{ m}^3} \times \frac{\text{m}^3}{\rho \text{ kg}}$$
$$= \frac{M \times 10^3}{\rho} \frac{\text{l}}{\text{s}}$$

and the water velocity is

$$Q_2 = \frac{\pi d^2}{4} \text{ m}^2 \times v \frac{\text{m}}{\text{s}} \times \frac{10^3 \text{ litre}}{1 \text{ m}^3}$$
$$v = \frac{4Q_2}{\pi d^2 \times 10^3} \frac{\text{m}}{\text{s}}$$

The equivalent length of the pipe is

$$l_e = \frac{0.81 M^2}{dp_1 \rho d^4} \text{ m}$$

(CIBSE, 1986c)

Each pipe fitting provides a hydraulic resistance to the flow of water. This resistance is characterized by the velocity pressure loss factor k, as shown in Table 8.5. The overall equivalent length of the index pipe circuit is the sum of the straight pipe lengths and the total equivalent length of all the pipe fittings:

$$EL = L + kl_e \text{ m}$$

The pressure drop through the index circuit is found from the resistance of the equivalent length of the pipework and fittings, plus the hydraulic resistance of

other known plant or valves. The manufacturer of heating and cooling coils, heat exchangers, flow control valves, balancing valves and boilers provides the known pressure drop to add to the overall figure:

$$\text{total } dp = EL \text{ m} \times dp_1 \frac{\text{Pa}}{\text{m}} \times \frac{1 \text{ kPa}}{10^3 \text{ Pa}} + \text{plant kPa}$$

EXAMPLE 8.2

An air heater coil provides 45 kW of heating to a ducted ventilation system. The total length of flow plus return pipe is 30 m. The water flow and return temperatures are 82 °C and 71 °C. The water velocity is not to exceed 1 m/s. The specific heat capacity of water is 4.186 kJ/kg K. The density of water at the mean water temperature is 974.85 kg/m³. The dynamic viscosity of water at the mean water temperature is 0.378 mPa s. Copper Table X BS 2871 pipe is to be used. The absolute surface roughness of copper is 0.0015 mm.

The designer estimates that the heat emission from the pipes will be 8% of the installed heat load. A cast iron boiler is the heat source, and the pipe system has 12 bends, 4 gate valves and 4 branch tees. The hydraulic resistance of the heating coil is 6 kPa. The two-port flow control valve has a hydraulic resistance of 10 kPa. Calculate the required water flow rate in kg/s and l/s, the estimated pipe diameter, the maximum carrying capacity of a selected pipe diameter, the flow error, the flow occurring in the pipe, the water velocity, the equivalent length of the circuit, and the frictional resistance pressure drop through the pipework system.

The required water flow is

$$Q_1 = \frac{H(100 + E)}{SHC(T_f - T_r) \times 100} \frac{\text{kg}}{\text{s}}$$
$$= \frac{45 \times (100 + 8)}{4.186 \times (82 - 71) \times 100} \frac{\text{kg}}{\text{s}}$$
$$= 1.0555 \text{ kg/s}$$

Also:

$$Q_1 = \frac{45 \times (100 + 8) \times 10}{4.186 \times (82 - 71) \times 974.85} \frac{\text{litre}}{\text{s}}$$
$$= 1.0827 \text{ litre/s}$$

The estimated pipe diameter is

$$d = \left(\frac{4Q_1}{\pi v \times 1000} \right)^{0.5} \times 1000 \text{ mm}$$
$$= \left(\frac{4 \times 1.0827}{\pi \times 1 \times 1000} \right)^{0.5} \times 1000 \text{ mm}$$
$$= 37 \text{ mm}$$

The next pipe diameter larger than this is 42 mm with an internal diameter of 39.63 mm. For a pressure drop rate dp_1 of 180 Pa/m in a 42 mm nominal diameter copper Table X BS 2871 pipe, the maximum water-carrying capacity is found, as before:

$$M = -\rho\pi\left(\frac{dp_1 \times d^5}{2\rho}\right)^{0.5} \times L$$

where

$$L = \log_{10}\left[\frac{k_s}{3.7d} + \frac{1.255\mu}{\rho} \times \left(\frac{2\rho}{dp_1 \times d^3}\right)^{0.5}\right]$$

$$= \log_{10}\left[\frac{0.0015 \times 10^{-3}}{3.7 \times 0.039\,63} + \frac{1.255 \times 0.378 \times 10^{-3}}{974.85} \times \left(\frac{2 \times 974.85}{175 \times (0.039\,63)^3}\right)^{0.5}\right]$$

$$= -3.6653$$

$$M = -\rho\pi\left(\frac{dp_1 \times d^5}{2\rho}\right)^{0.5} \times L$$

$$= -974.85\pi \times \left(\frac{175 \times (0.039\,63)^5}{2 \times 974.85}\right)^{0.5} \times -3.6653 \text{ kg/s}$$

$$= 1.0515 \text{ kg/s}$$

$$\text{flow error } E = \frac{Q_1 - M}{Q_1} \times 100\%$$

$$= \frac{1.0555 - 1.0515}{1.0555} \times 100\%$$

$$= 0.4\%$$

$$Q_2 = \frac{M \times 10^3}{\rho} \frac{\text{litre}}{\text{s}}$$

$$= \frac{1.0515 \times 10^3}{974.85} \frac{\text{litre}}{\text{s}}$$

$$= 1.0786 \text{ litre/s}$$

$$v = \frac{4Q_2}{\pi d^2 \times 10^3} \frac{\text{m}}{\text{s}}$$

$$= \frac{4 \times 1.0786}{\pi \times (0.039\,63)^2 \times 10^3} \frac{\text{m}}{\text{s}}$$

$$= 0.874 \text{ m/s}$$

The equivalent length of the pipe is

$$l_e = \frac{0.81M^2}{dp_1 \rho d^4} \text{ m}$$

$$= \frac{0.81 \times 1.0515^2}{175 \times 974.85 \times (0.039\,63)^4} \text{ m}$$

$$= 2.13 \text{ m}$$

The pipe fittings velocity pressure loss factors have a total of

$$\Sigma k = 2.5 + 12 \times 1 + 4 \times 0.3 + 4 \times 1.5$$

$$= 21.7$$

$$EL = L + kl_e \text{ m}$$

$$= 30 + 21.7 \times 2.13 \text{ m}$$

$$= 76.2 \text{ m}$$

$$\text{total } dp = EL \text{ m} \times dp_1 \frac{\text{Pa}}{\text{m}} \times \frac{1 \text{ kPa}}{10^3 \text{ Pa}} + \text{plant kPa}$$

$$= 76.2 \text{ m} \times 175 \frac{\text{Pa}}{\text{m}} \times \frac{1 \text{ kPa}}{10^3 \text{ Pa}} + 6 + 10 \text{ Pa}$$

$$= 29.339 \text{ kPa}$$

PIPE HEAT EMISSION

The heat lost from heating hot water pipes, in watts per metre run of pipe per degree Kelvin difference between the mean water and ambient air temperatures, having 25 mm thick fibreglass rigid insulation, is given in Table 8.6 (CIBSE, 1986c).

Table 8.6 Heat emission from pipes insulated with 25 mm fibreglass

Nominal diameter (mm)	*Heat emission* (W/m K)
15	0.19
22	0.23
28	0.25
35	0.29
42	0.32
54	0.37
67	0.44
80	0.5
100	0.61
125	0.75
150	0.88

PIPELINE COST

The installed cost of a two-pipe system is evaluated from the cost of the pipe, thermal insulation, pipe fittings, supporting brackets, labour, overheads and profit. Costs are to be used for comparison, so that alternative routes, diameters and lengths may be tested quickly by the designer. After having found the relative cost of the accepted pipework design, the final costing can be completed. Such final costing includes fixed cost items that are not strongly influenced by the pipe layout. Pipe lengths and diameters that are accumulated from this worksheet can be used for the final costing. Typical pipe cost data is available either from within the user's company or from regularly published price books (Spon, 1992).

To provide a reasonable comparison of the installed cost of different pipe diameters, the following data is used. The data is for copper pipe to BS 2871, Table X, and an example is shown for a 35 mm diameter pipe. The user can construct similar tables of costs for other pipe materials, and use other sources of cost data as required.

The measured installed cost of 35 mm copper pipe is £12.92 per metre length. A 15% discount is normally applied and then an addition of 24.3% for waste, overheads and profit. The typical installation of a 5 m length of pipe in a heating system would have pipe fittings of a tee, two elbows, two pipe brackets and 25 mm thick thermal insulation. The tee costs £12.89, discount 35%, waste, overheads and profit 18.39%. An elbow costs £7.74, discount 35%, waste overheads and profit 18.39%. A single-pipe ring bracket costs £6.69, discount 25%, waste, overheads and profit 18.39%. Preformed glass fibre 30 mm thick thermal insulation for concealed service areas costs £4.73 per metre length, 0% discount, waste, overheads and profit 18.39%. The total installed cost of a 5 m length of 35 mm diameter copper Table X pipe is given by

$$\begin{aligned}\text{cost} &= (5 \times £12.92, -15\%, +24.3\%) + (£12.89, -35\%, +18.39\%)\\ &\quad (2 \times £7.74, -35\%, +18.39\%) + (2 \times £6.69, -25\%, +18.39\%)\\ &\quad + (5 \times £4.73, -0\%, +18.39\%)\\ &= £(68.25 + 9.92 + 11.91 + 11.88 + 28.00)\\ &= £129.96\end{aligned}$$

The average cost of one metre of 35 mm diameter installed copper pipe is

$$\begin{aligned}\text{cost} &= \frac{£129.96}{5\,\text{m}}\\ &= £26.00 \text{ per metre}\end{aligned}$$

Copper pipe diameters of 15–67 mm are jointed with capillary fittings. Pipes of 76–159 mm have weldable joints. Typical costs are listed in Table 8.7.

Table 8.7 Installed costs of copper Table X pipework

Diameter (mm)	*Installed cost* (£/m)
15	13.70
22	16.10
28	19.45
35	26.00
42	29.55
54	37.00
67	52.00
80	77.40
100	110.85
125	172.40
150	216.60

EXAMPLE 8.3

A two-pipe heating system has copper Table X pipework. The lengths are 150 m of 15 mm, 80 m of 20 mm, 60 m of 28 mm, 40 m of 35 mm and 10 m of 42 mm. The heat emitters provide an output of 65 kW to the building. The flow and return temperatures used are 82 °C and 70 °C. The ambient air temperature around the pipes is 18 °C. Calculate the pipe heat emission as a percentage of the installed heat load, and the relative cost of the pipe system.

$$\begin{aligned}\text{temperature difference} &= \frac{82 + 70}{2} - 18\,\text{K}\\ &= 58\,\text{K}\end{aligned}$$

The solution is presented in Table 8.8.

Table 8.8 Solution to Example 8.3

Diameter	*Pipe length,*	*Heat emission*		*Installed cost*	
(mm)	*L* (m)	(W/m K)	(kW)	(£/m)	(£)
15	150	0.19	1.653	13.70	2055
22	80	0.23	1.067	16.10	1288
28	60	0.25	0.870	19.45	1167
35	40	0.29	0.673	26.00	1040
42	10	0.32	0.186	29.55	295
		Total	4.449		5845

$$\begin{aligned}\text{pipe emission} &= \frac{4.449\,\text{kW}}{65\,\text{kW}} \times 100\%\\ &= 6.8\%\end{aligned}$$

total cost of the pipe system = £5845

DATA REQUIREMENT

The user will enter new data for:

1. Your name in cell B4.
2. Job title in cell B5.
3. Job reference number in cell B6.
4. Filename in cell B9.
5. Water flow temperature in cell C44.
6. Water return temperature in cell C45.
7. Water density in cell C47.
8. Water dynamic viscosity in cell C48.
9. Specific heat capacity of the water in cell C49.
10. Temperature of the ambient air surrounding the pipework system in cell C50. This is the temperature that acts as the sink for the pipe heat emission. This may be the temperature within a builder's work duct or a ceiling space.
11. Roughness of the pipe internal surface in cell C51.
12. First estimate of the heat emission from the pipework system in cell C52. This can be left at the original value of 5%.
13. Total heat output from the heating system, *H* kW, in cell C54. This is the sum of the heat emitter outputs.
14. Type of pipe, nominal and internal diameters of the pipe to be used in cell range A69 to C79. COPY this block of data from the data bank that commences at line 161; the cell range from A69 to C79 must contain the correct range of pipe diameters to be used. The data that is entered here is automatically copied to other parts of the worksheet. This information is critical to the running of the worksheet.

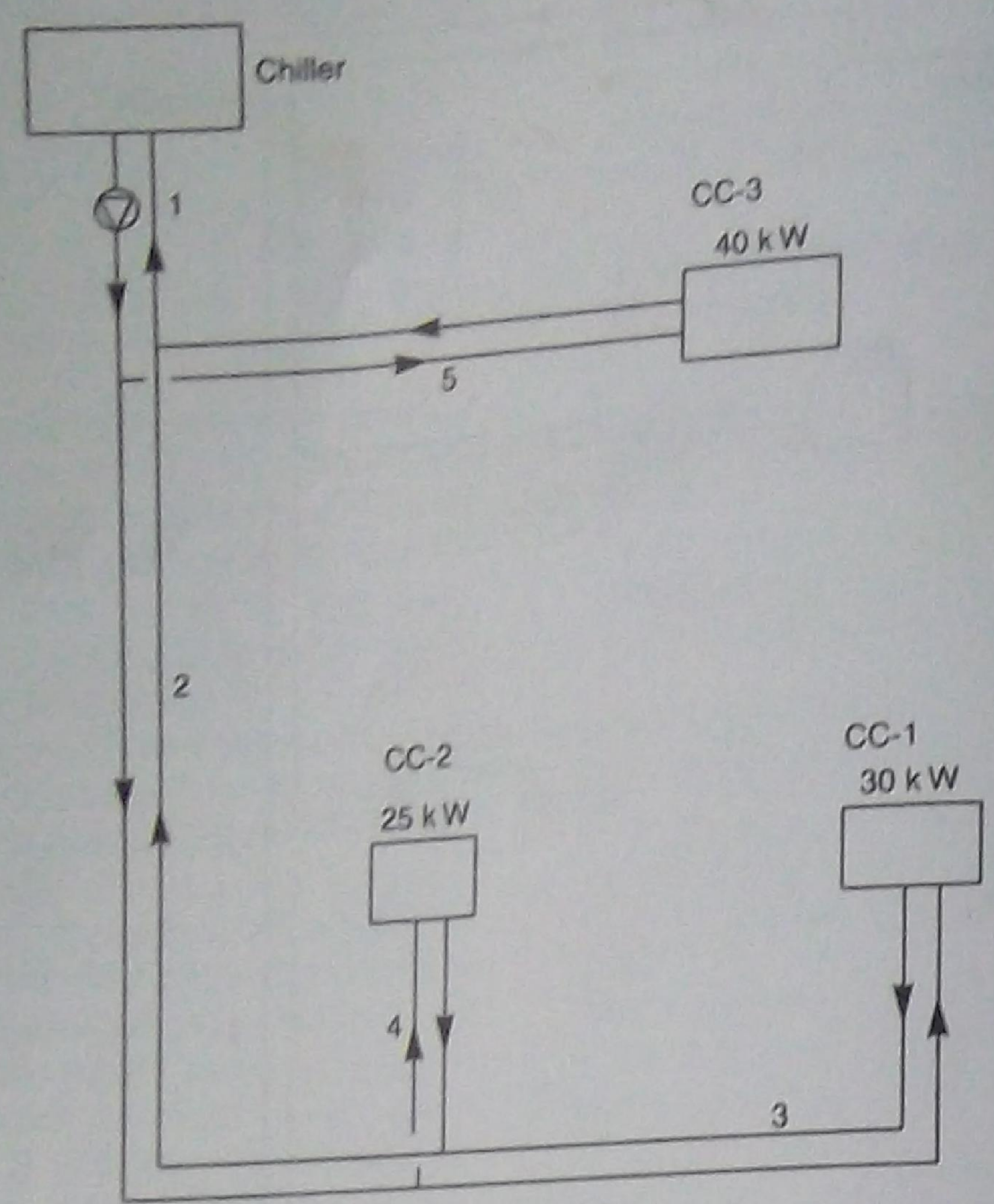
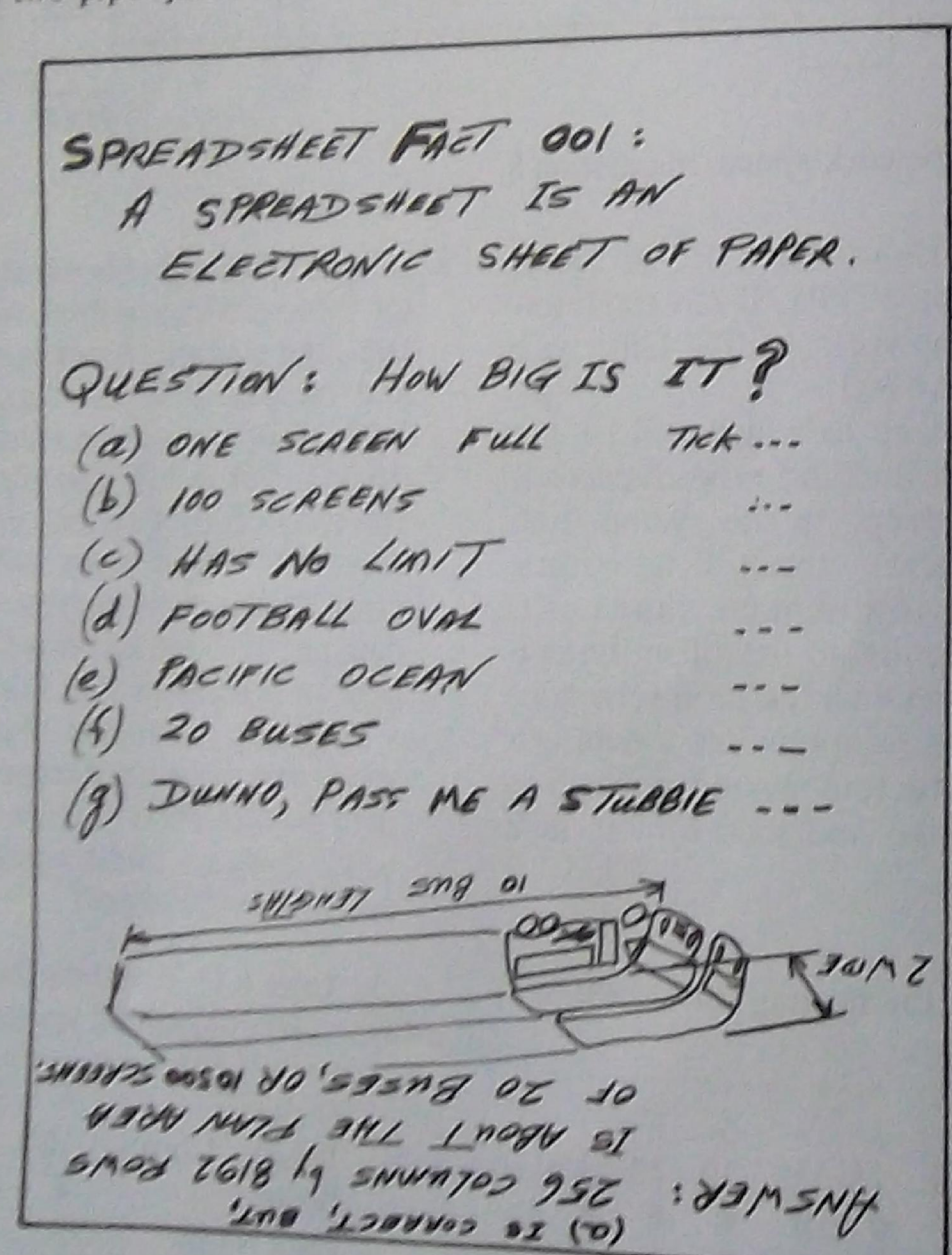


Fig. 8.6 Chilled water two-pipe system to air cooling coils in question 9.



Spreadsheet fact 001.

9 Lighting

INTRODUCTION

The general artificial lighting of interiors is usually to provide a uniform illumination with fluorescent lamps. Desk level is frequently the illuminated working plane. This worksheet provides the lumen design method for general lighting of a single rectangular space. The design for additional areas and rooms is accomplished with repeated copies of this file. The user will save each room file under a unique name. The effect of localized lighting and glare is not calculated. Sufficient reference data is provided on the worksheet for the examples and questions and general design cases. The user will enter the required design illuminance and current photometric data for professional applications.

LEARNING OBJECTIVES

Study and use of this chapter will enable the user to:

1. know the principles of the artificial lighting of interiors with the lumen method;
2. know the photometric data to be used for design;
3. use the utilization factor for installed lamp, luminaire and room combinations;
4. know the use of light loss factor;
5. calculate the number of luminaires and lamps for a lighting system;
6. know how to use luminaire space-to-height ratio;
7. design the layout of the luminaires;
8. calculate the installed electrical power for the lighting system;
9. know the electrical power load per square meter of floor area.

Key terms and concepts

LUMEN DESIGN METHOD

The lumen design method for the artificial lighting of the interiors of rooms provides a uniform distribution of illuminance over the working plane. The working plane is frequently the same area as the floor, but at a raised height, typically at desk level. The intention of the method is to locate luminaires at a spacing between the rows and height above the working plane that will not create shadows or poorly illuminated areas (Chadderton, 1995, Ch. 11). Design values for illuminance vary from 50 to 2000 lx (lm/m^2), depending upon the detail to be viewed and the energy economy required (CIBSE, 1986a; Pritchard, 1985, Ch. 7).

The daylighting of interiors is accompanied by winter heat losses through the areas of glazing and the admission of solar heat gains during the warmer weather. Taken together with the consumption of electrical energy to provide the artificial illumination, it is clear that lighting is an energy-intensive service. The engineering designer aims to optimize the architectural, heating, air conditioning and electrical power components of the overall scheme in order to minimize the use of energy in the provision of an acceptable standard of service.

A selection of reference data is provided in the worksheet. The user will update the reference information and extend the range data in the worksheet in accordance with design office requirements. Lamp photometric data is on the reference screen in the same arrangement as is used in the calculation area of the worksheet. The user can copy the relevant data directly from the reference area to the calculation area with one COPY RANGE command. The data on the worksheet is for

1. illuminance levels in lux;
2. luminance factors for the room surfaces;
3. utilization factors for a typical combination of luminaire and room configuration (Philips, 1986);
4. fluorescent lamp data (GE, 1992).

The geometric configuration of the room is represented by the room index, RI:

$$RI = \frac{L_1 W_1}{H_3(L_1 + W_1)}$$

where RI = room index (dimensionless); L_1 = room length (m); W_1 = room width (m); H_3 = height from working plane to luminaire $= H_1 - H_2$ (m); and H_1 = height of the room (m), H_2 = height of working plane above the floor (m).

The surface finishes of the ceiling, walls and floor reflect the available light to create an overall illuminance. A luminance factor is attributed to each surface (BS 4800 : 1972). Typical values are 70% for a white ceiling, 50% for coloured walls and 10% for flooring. The user can enter any value that is appropriate for the design.

Utilization factor is the ratio of the luminous flux received at the working surface to the installed flux. This ratio should be as high as possible for energy-efficient lighting. It is a combination of the lamp type, luminaire construction, room surface luminance factors and room dimensions. The designer normally refers to published utilization factors from manufacturers' data sheets. Sample values are provided in the worksheet. The data provided is for use within this chapter. The user will acquire current information for professional design.

The number of luminaires that are needed to provide the design illuminance are found from

$$\text{no. of luminaires} = \frac{\text{design lux} \times \text{working plane area } A_1 \text{ m}^2}{\text{luminaire lumens} \times UF \times LLF}$$

where A_1 = area of the working plane (m^2); UF = utilization factor (dimensionless); LLF = overall light loss factor (dimensionless); and MF = maintenance factor (dimensionless).

$$LLF = \text{lamp lumen } MF\% \times \frac{\text{luminaire } MF\%}{100} \times \frac{\text{room surface } MF\%}{100}$$

Lamp lumen maintenance factor represents the performance of the lamp at 2000 h of use; this is typically 92% of the initial lumens (Pritchard, 1985). Frequent cleaning of the room surfaces and luminaires, at three-or six-monthly intervals, will ensure that the room surface and luminaire maintenance factors within clean environments have an average value of about 95%. Each design of luminaire has a recommended maximum spacing-to-height ratio. This defines the maximum allowable spacing between the rows of luminaires to provide a shadow-free distribution of light on the working plane:

$$SHR = \frac{S_1}{H_3}$$

where SHR = maximum spacing-to-height ratio (dimensionless); S_1 = space between rows of luminaires (m); and S_2 = space between luminaire perimeter row and wall (m).

When the number of luminaires has been calculated, the designer decides the distribution of the luminaires on a reflected ceiling plan. The lighting system is integrated with the position of the working planes, the air conditioning supply air diffusers and return air grilles, smoke and fire detectors, fire sprinkler heads, emergency and exit lighting, equipment being suspended from the ceiling, and architectural features. The number of luminaires in a row and the number of rows is determined by iteration of the options on the reflected ceiling plan. These options are entered into the worksheet until a satisfactory solution is obtained. The worksheet shows the illuminance that will be provided by each option, the row spacing and the electrical power consumption by the lighting system. The illuminance produced is found from

$$\text{illuminance} = \frac{\text{no. of luminaires} \times \text{luminaire lumens} \times UF \times \text{LLF}}{A_1 \text{ m}^2}$$

The electrical power consumption of the lighting system is

$$\text{power} = \text{luminaires} \times \text{lamps per luminaire} \times (W_1 + W_2)\ \text{W}$$

where W_1 = lamp power (W) and W_2 = lamp control gear power (W).

The lighting system electrical power consumption per square metre of room floor area is

$$\text{power} = \frac{\text{luminaires} \times \text{lamps per luminaire} \times (W_1 + W_2)}{A_1\ \text{m}^2}\ \text{W}$$

EXAMPLE 9.1

A uniform illuminance of 400 lx is to be provided in the first floor general office shown in Fig. 9.1. The working plane is to be taken as the same as the floor area. The lighting system will be permanently switched on from 8.00 a.m. to 6.00 p.m., 5 days per week for 52 weeks per year. The building has a complete internal cleaning twice per year. The ceiling has white acoustic tiles, luminance 70%, the walls are painted cream, luminance 50%, and the light grey carpet has a luminance of 10%. The maintenance factors are to be taken as 95% for the room surfaces and 95% for the luminaires. Fluorescent triphosphor Polylux 840 lamps are to be selected. The lamp lumen maintenance factor is 92% at 2000 h of use. Each lamp consumes 36 W plus 20 W by the control gear, is 1200 mm long, has an initial luminous flux of 3450 lm, a colour rendering index of 80 and a corrected colour temperature of 4000 K. The luminaires have two lamps each, and are 1300 mm long open-bottom reflectors that are surface mounted beneath the ceiling tiles. The maximum spacing-to-height ratio for the luminaires is 1.5. The manufacturer's utilization factor for the combination of lamp, luminaire and room data is 62%.

$$
\begin{aligned}
H_1 &= 3\,\text{m} \\
H_2 &= 0.8\,\text{m} \\
L_1 &= 10\,\text{m} \\
W_1 &= 10\,\text{m} \\
H_3 &= H_1 - H_2\,\text{m} \\
&= 3 - 0.8\,\text{m} \\
&= 2.2\,\text{m} \\
RI &= \frac{L_1 W_1}{H_3(L_1 + W_1)} \\
&= \frac{10 \times 10}{2.2 \times (10 + 10)} \\
&= 2.273 \\
A_1 &= 10\,\text{m} \times 10\,\text{m} \\
&= 100\,\text{m}^2
\end{aligned}
$$

$UF = 62\%$
lamp lumen $MF = 92\%$
luminaire $MF = 95\%$
room surface $MF = 95\%$

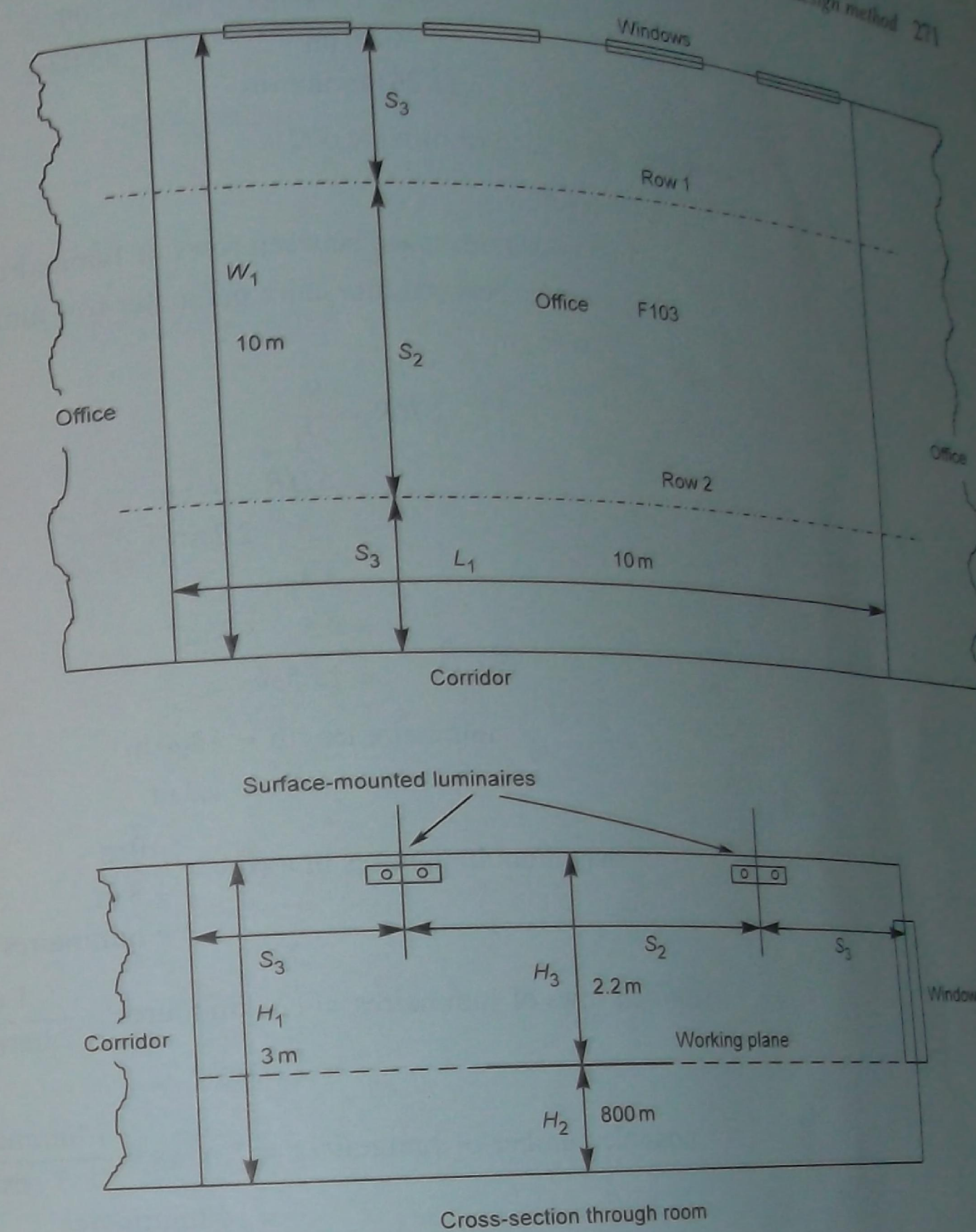


Fig 9.1 Office in Example 9.1

$$
\begin{aligned}
LLF &= \text{lamp lumen } MF\% \times \frac{\text{luminaire } MF\%}{100} \times \frac{\text{room surface } MF\%}{100} \\
&= 92\% \times \frac{95\%}{100} \times \frac{95\%}{100} \\
&= 83.03\% \\
\text{luminaire lumens} &= \frac{2\ \text{lamps}}{\text{luminaire}} \times \frac{3450\,\text{lm}}{\text{lamp}} \\
&= 6900\,\text{lm} \\
\text{no. of luminaires} &= \frac{\text{design lux} \times \text{working plane area } A_1\,\text{m}^2}{\text{luminaire lumens} \times UF \times LLF}
\end{aligned}
$$

$$= \frac{400\,\text{lx} \times 100\,\text{m}^2}{6900\,\text{lm}} \times \frac{100}{62} \times \frac{100}{83.03}$$
$$= 11.26\,\text{luminaires}$$

12 luminaires are needed to provide 400 lx.

$SHR = 1.5$

S_1 = maximum space between rows of luminaires (m)

S_2 = space between luminaire perimeter row and wall (m)

$= 0.5 S_1$ m

$$SHR = \frac{S_1}{H_3}$$
$$S_1 = SHR \times H_3\,\text{m}$$
$$= 1.5 \times 2.2\,\text{m}$$
$$= 3.3\,\text{m}$$
$$S_2 = 0.5 \times 3.3\,\text{m}$$
$$= 1.65\,\text{m}$$

$$\text{luminaire length} = 1300\,\text{m}$$
$$= 1.3\,\text{m}$$

$$\text{maximum luminaires in a row} = \frac{10\,\text{m}}{1.3\,\text{m}}$$
$$= 7\,\text{luminaires}$$

$$\text{required rows of luminaires} = 12\,\text{luminaires} \times \frac{1\,\text{row}}{7\,\text{luminaires}}$$
$$= 2\,\text{rows}$$

$$\text{possible number of luminaires} = 2\,\text{rows} \times \frac{7\,\text{luminaires}}{1\,\text{row}}$$
$$= 14\,\text{luminaires}$$

The spacing of two rows will be

$$S_1 = \frac{10\,\text{m}}{2\,\text{rows}}$$
$$= 5\,\text{m}$$

The proposed spacing is greater than the maximum allowable distance, 3.3 m, for the correct distribution of light. Try three rows of four luminaires per row, to provide the required 12 luminaires:

$$S_1 = \frac{10\,\text{m}}{3\,\text{rows}}$$
$$= 3.333\,\text{m}$$
$$S_2 = 0.5 \times 3.333\,\text{m}$$
$$= 1.667\,\text{m}$$

The row spacing is marginally wider than the maximum, and the luminaires are spread longitudinally along each row. The designer may be able to decide

that this is an acceptable solution. The average illuminance upon the working plane is

$$\text{illuminance} = \frac{\text{no. of luminaires} \times \text{luminaire lumens} \times UF \times LLF}{A_1\,\text{m}^2}$$
$$= \frac{12 \times 6900\,\text{lm}}{100\,\text{m}^2} \times \frac{62}{100} \times \frac{83.03}{100}$$
$$= 426\,\text{lx}$$

The illuminance provided exceeds the design value by

$$\text{excess illuminance} = \frac{426 - 400}{400} \times 100\%$$
$$= 6.5\%$$

This is certainly acceptable. The designer may allow up to, say, 25% excess illuminance, 500 lx in this case. The lighting system electrical power consumption is

$$\text{power} = \text{luminaires} \times \text{lamps per luminaire} \times (W_1 + W_2)\,\text{W}$$

where W_1 = lamp power (W), and W_2 = lamp control gear power (W).

$$W_1 = 36\,\text{W}$$
$$W_2 = 20\,\text{W}$$
$$\text{power} = 12 \times 2 \times (36 + 20)\,\text{W}$$
$$= 1344\,\text{W}$$
$$\text{power} = \frac{1344\,\text{W}}{100\,\text{m}^2}$$
$$= 13.44\,\text{W/m}^2$$

DATA REQUIREMENT

The user will enter new data for:

1. Your name in cell B4.
2. Job title in cell B5.
3. Job reference in cell B6 and C6.
4. Filename in cell B10.
5. Room name in cell D25.
6. Room number in cell D26.
7. Room location in cell D27.
8. Room use in cell D28.
9. Hours of lighting use per day in cell D29.
10. Days of lighting use per week in cell D30.
11. Weeks of lighting use per year in cell D31.
12. Number of weeks between lamp and luminaire cleaning in cell D33.
13. Luminance of the ceiling in cell D34.
14. Luminance of the walls in cell D35.
15. Luminance of the floor in cell D36.
16. Maintenance factor for the room surfaces in cell D37.

10 Electrical cable sizing

INTRODUCTION

This worksheet finds the size of cable that is required for single-phase, three-phase and extra low voltage single-phase electrical power and lighting circuits. The user enters the power load, power factor and the measured length of the conductor. The worksheet evaluates the phase current, the maximum allowable length of the conductor, the voltage drop, and whether each cable size will meet the resistance criteria. Cable data from any source can be entered and used. Sufficient reference data is provided in the worksheet for the examples and questions and general design cases. The user will enter the required reference data for professional applications.

LEARNING OBJECTIVES

Study and use of this chapter will enable the user to:

1. know the data needed to calculate the size of an electrical cable;
2. know the voltage drop limits for sizing cable;
3. know the current limits for cables;
4. know the cable sizes used;
5. calculate the phase current for single-and three-phase circuits;
6. calculate the maximum allowable length of cable to meet the voltage drop limitation;
7. calculate the voltage drop in a cable;
8. calculate the percentage voltage drop in a cable.

Key terms concepts

CABLE DESIGN METHOD

The current in a cable is found from the electrical power consumption of the load that is supplied, the voltage that is applied to the circuit, and the power factor of the load. This design current is compared with the current capacity of cables to find a suitable cross-sectional area. The voltage drop along the length of the proposed cable is calculated from the known resistance. The voltage drop in a cable from the supply point to the final load is limited, by regulation, to 4% of the applied voltage. The cable support method and how the cables are bunched together are taken into account when using the published cable data (IEE, 1991). Table 10.1 shows values of current rating and voltage drop for a selection of cables. A larger number of cables are listed on the worksheet for design applications. The simple approach (Jenkins, 1991) to the selection of cables is used, with the inclusion of the power factor (Chadderton, 1995, Ch. 13). Note that mV/A m is the resistance of 1 m of cable in milliohms (mΩ).

Table 10.1 Electrical cable capacities for unenclosed copper cables, twin-sheathed in PVC, clipped to the surface of the building

Nominal cross-sectional area of conductor (mm^2)	*Maximum current rating, I_b* (A)	*Voltage drop* (mV/A m)
1	15	44
1.5	19.5	29
2.5	27	18
4	36	11
6	46	7.3
10	63	4.4
16	85	2.8

(IEE, 1991)

The phase current in a three-phase 415 V circuit is

$$\text{current} = \frac{\text{power W}}{415\,\text{V} \times PF \times \sqrt{3}}\ \text{A}$$

The current in a single-phase 240 V circuit is

$$\text{current} = \frac{\text{power W}}{240\,\text{V} \times PF}\ \text{A}$$

The current in a single-phase 12 V circuit is

$$\text{current} = \frac{\text{power W}}{12\,\text{V} \times PF}\ \text{A}$$

where power = output electrical power (W), and PF = power factor (dimensionless).

The maximum allowable reduction in voltage in a circuit is 4% of the incoming voltage (IEE, 1991). The voltage drops are:

1. 415 V three phase, $V = \frac{4}{100} \times 415\ \text{V}$
   $= 16.6\ \text{V}$
2. 240 V single phase, $V = \frac{4}{100} \times 240\ \text{V}$
   $= 9.6\ \text{V}$
3. 12 V single phase, $V = \frac{4}{100} \times 12\ \text{V}$
   $= 0.48\ \text{V}$

The maximum length of cable that can be used so that the voltage drop limit is not exceeded is

$$L_1 = \frac{\text{maximum voltage drop allowed V} \times 10^3\,\text{m}}{\text{load current A} \times \text{voltage drop mV/A m}}$$

When the length of the conductor is known, from measurement on the building drawings or along the cable route, the voltage drop produced in that circuit is

$$\text{voltage drop} = L_1\,\text{m} \times I_b\,\text{A} \times \text{voltage drop}\,\frac{\text{mV}}{\text{A m}} \times \frac{1}{10^3}\,\text{V}$$

where I_b = design phase current (A).

> EXAMPLE 10.1
>
> A three-phase 415 V fan motor has an electrical power rating of 7.5 kW. The motor power factor is 0.75. The conductor from the main distribution board has a measured length of 60 m. The cable is installed in cable trays with other circuits. Use the cable data in Table 10.1 to find a suitable cable size.

The solution to this example is shown in the original copy of the file A:CABLE.WKS. The cable data in the worksheet is from the 16th edition of the reference regulation.

Allowable voltage drop is

$$415\text{ V three phase, } V = \frac{4}{100} \times 415\ \text{V} = 16.6\ \text{V}$$

Phase current is

$$I_b = \frac{\text{power W}}{415\ \text{V} \times PF \times \sqrt{3}}\ \text{A} = \frac{7.5 \times 10^3\ \text{W}}{415\ \text{V} \times 0.75 \times \sqrt{3}}\ \text{A} = 13.9\ \text{A per phase}$$

Try a 1 mm^2 cable.

allowable voltage drop rate = 44 mV/A m
maximum current rating = 15 A
maximum cable length is

$$L_1 = \frac{\text{maximum voltage drop allowed V} \times 10^3}{I_b\,\text{A} \times \text{voltage drop mV/A m}}\ \text{m} = \frac{16.6\ \text{V} \times 10^3}{13.9\ \text{A} \times 44\ \text{mV/A m}}\ \text{m} = 27\ \text{m}$$

The resistance of this cable is too high, because the allowable voltage drop will allow only 27 m of cable, whereas 60 m is required. If this cable was used, the voltage drop would be

$$\text{volt drop} = L_1\,\text{m} \times I_b\,A \times \text{voltage drop}\,\frac{\text{mV}}{\text{A m}} \times \frac{1}{10^3}\,\text{V} = 60\ \text{m} \times 13.9\ \text{A} \times 44\,\frac{\text{mV}}{\text{A m}} \times \frac{1}{10^3}\,\text{V} = 36.7\ \text{V}$$

This would exceed the allowable drop of 16.6 V. Try the 2.5 mm^2 cable; allowable drop is 18 mV/A m.

$$L_1 = \frac{\text{maximum voltage drop allowed V} \times 10^3}{I_b\,\text{A} \times \text{voltage drop mv/A m}}\ \text{m} = \frac{16.6\ \text{V} \times 10^3}{13.9\ \text{A} \times 18\ \text{mV/A m}}\ \text{m} = 66.3\ \text{m}$$

$$\text{voltage drop} = L_1\,\text{m} \times I_b\,\text{A} \times \text{voltage drop}\,\frac{\text{mV}}{\text{A m}} \times \frac{1}{10^3}\,\text{V} = 60\ \text{m} \times 13.9\ \text{A} \times 18\,\frac{\text{mV}}{\text{A m}} \times \frac{1}{10^3}\,\text{V} = 15\ \text{V}$$

The 2.5 mm^2 cable has a maximum current capacity of 27 A and it produces a voltage drop that is within the 4% allowed. The percentage volt drop is

$$\text{voltage drop} = \frac{15}{415} \times 100\% = 3.6\%$$

The proposed design complies with the allowable voltage drop and the current capacity.

DATA REQUIREMENT

The user will enter new data for:

1. Your name in cell B4.
2. Job title in cell B5.

3. Job reference in cells B6 and C6.
4. Filename in cell B9.
5. The electrical power load in cell C27.
6. Power factor in cell C29.
7. Conductor length to be installed in cell C31.
8. Circuit voltage in cell C33.
9. Cable number in column A. A consecutive series of numbers are in the worksheet, and these can remain unaltered.
10. A description of the cable installation in cells C35, D35, E35, F35 and G35. Enter a description such as: grouped cable, clipped direct. Enter the correct number of characters in each cell.
11. The normal range of cable cross-sectional areas is provided in column B from cell B53. New ranges of cable sizes can be entered as required.
12. The maximum current rating of each cable size is provided in column C from cell C53. New data can be entered in these cells as required.
13. The voltage drop in mV per A per m length of cable is provided in column D from cell D53. New data can be entered in these cells as required.

OUTPUT DATA

1. The current date is given in cell B7.
2. The current time is given in cell B8.
3. The voltage in a three-phase supply is given in cell D43.
4. The maximum allowable percentage voltage drop in a circuit is given in cell D44.
5. The maximum allowable voltage drop in a circuit is given in cell D45.
6. The power load carried by the cable is given in cell E53.
7. The power factor of the connected load is given in cell F53.
8. The phase current is given in cell G53.
9. The maximum length of the conductor is given in cell H53.
10. The measured length of the conductor is given in cell I53.
11. The voltage drop in the conductor is given in cell J53.
12. The voltage drop expressed as a percentage of the voltage that is applied to the circuit is given in cell K53.
13. The acceptability criteria for the voltage drop (either 'Allowed' or 'Too high') are shown in cell L53.
14. The current limit acceptability criteria for the current carried by the cable are given in cell M53. Either 'Allowed' or 'Too high' is shown.
15. The cell range E54 to M79 contains the equivalent output data to that in line 53.
16. The cell range A83 to M100 contains the equivalent output data to that on line 53 for 240 V single-phase circuits.
17. The cell range A103 to M117 contains the equivalent output data to that on line 53 for 12 V extra low voltage single-phase circuits.

FORMULAE

Representative samples of the formulae are given here. Each formula can be read by moving the cursor to the cell. The equation is presented in the form in

which it would normally be written and in the format that is used by the spreadsheet.

Cell B7

@TODAY

This function produces the serial number of the current day and time. The cell is formatted to display the date.

Cell B8

@TODAY

This function produces the serial number of the current day and time. The cell is formatted to display the time.

Cell D45

+D43*D44/100

The maximum voltage drop that is allowable in a cable is found from the allowable percentage drop from

$$\text{voltage} = 415\ \text{V} \times \frac{4}{100}\ \text{V}$$

$$= 16.6\ \text{V}$$

$$\text{D45} = \text{D43} \times \frac{\text{D44}}{100}\ \text{V}$$

Cell range C46 to G46

+$C$35

The description of the cable installation is copied from cell C35 into C46; this is repeated along the range.

Cell E53

+$C$27

The circuit power load is copied from cell C27 into cell E53.

Cell F53

+$C$29

The power factor of the load is copied from cell C29 into cell F53.

Cell G53

+E53/$D$43/F53/@SQRT(3)

The phase current is

$$I_b = \frac{\text{power W}}{\text{volts} \times PF \times \sqrt{3}}\ \text{A}$$

$$G53 = \frac{\text{E53 W}}{\text{D43 V} \times \text{F53} \times \sqrt{3}}\ \text{A}$$

Cell H53

+$D$45*1000/G53/D53

The maximum allowable length of the conductor in a 415 V three-phase circuit is

$$L_1 = \frac{\text{maximum voltage drop allowed V} \times 10^3}{I_b\,\text{A} \times \text{voltage drop mV/A m}}\ \text{m}$$

$$= \frac{16.6\ \text{V} \times 10^3}{I_b\,\text{A} \times \text{voltage drop mV/A m}}\ \text{m}$$

$$H53 = \frac{\text{D45 V} \times 10^3}{\text{G53 A} \times \text{D53 mV/A m}}\ \text{m}$$

Cell I53

+$C$31

The measured length of conductor is copied from cell C31 into cell 153.

Cell J53

+I53*G53*D53/1000

The actual voltage drop in the cable is

$$\text{voltage drop} = L_2\,\text{m} \times I_b\,\text{A} \times \frac{\text{mV}}{\text{A m}} \times \frac{1}{1000}\ \text{V}$$

$$J53 = \text{I53 m} \times \text{G53 A} \times \frac{\text{D53 mV}}{\text{A m}} \times \frac{1}{1000}\ \text{V}$$

Cell K53

+J53*100/$D$43

The percentage voltage drop that is produced is

$$\text{voltage drop} = \text{voltage drop} \times \frac{100}{\text{voltage}}\ \%$$

$$\text{voltage drop} = \text{J53 V} \times \frac{100}{\text{D43}}\ \%$$

Cell L53

@IF (K53<=$D$44,"Allowed","Too high")

This tests whether the voltage drop that is produced in the cable exceeds the maximum allowable voltage drop. If the actual voltage drop in cell K53 is less than or equal to the limit in cell D44, the cable volt drop is 'Allowed'. If the voltage drop exceeds the limit, the next message, 'Too high', is shown in cell L53.

Cell M53

@IF (G53<=C53,"Allowed","Too high")

This tests whether the current in the cable exceeds the maximum allowable current. If the circuit current in cell G53 is less than or equal to the limit in cell C53, the cable current is 'Allowed'. If the current exceeds the cable limit, the next message, 'Too high', is shown in cell M53.

Cell range E54 to M117

The formulae from line 53 are repeated in this cell range for the additional cables and single phase circuits.

> EXAMPLE 10.2
>
> A single-phase 240 V exhaust fan motor has an electrical power rating of 250 W. The motor power factor is 0.7. The conductor from the main distribution board has a measured length of 25 m. The cable is installed in cable trays with other circuits. Use the data in Table 10.1 to find a suitable cable size.

Allowable voltage drop is

$$240\ \text{V phase},\ V = \frac{4}{100} \times 240\ \text{V}$$
$$= 9.6\ \text{V}$$

Phase current is

$$I_b = \frac{250\ \text{W}}{240\ \text{V} \times 0.7}\ \text{A}$$
$$= 1.49\ \text{A}$$

The 1 mm^2 cable has an allowable drop of 44 mV/A m:

$$L_1 = \frac{9.6\ \text{V} \times 10^3}{1.49\ \text{A} \times 44\ \text{mV/A m}}\ \text{m} = 146\ \text{m}$$

$$\text{voltage drop} = 25\ \text{m} \times 1.49\ \text{A} \times 44\ \frac{\text{mV}}{\text{A m}} \times \frac{1}{10^3}\ \text{A}$$
$$= 1.64\ \text{V}$$

The 1 mm^2 cable has a maximum current capacity of 15 A, and it produces a voltage drop that is within the 4% allowed. The percentage voltage drop is

$$\text{voltage drop} = \frac{1.64}{240} \times 100\%$$
$$= 0.7\%$$

The proposed design complies with the allowable voltage drop and the current capacity.

> EXAMPLE 10.3
>
> A single-phase 12 V lamp has an electrical power rating of 50 W. The lamp power factor is 1.0. The conductor from the main distribution board has a measured length of 5 m. The cable is clipped onto the building surfaces alongside other cables. Use the data in Table 10.1 to find a suitable cable size to use.

Allowable voltage drop is

$$12\text{ V single phase}, \quad V = \frac{4}{100} \times 12\text{ V}$$
$$= 0.48\text{ V}$$

Phase current is

$$I_b = \frac{50\text{ W}}{12\text{ V} \times 1.0}\text{ A}$$
$$= 4.2\text{ A}$$

The 2.5 mm^2 cable has an allowable drop of 18 mV/A m:

$$L_1 = \frac{0.48\text{ V} \times 10^3}{4.2\text{ A} \times 18\text{ mV/A m}}\text{ m}$$
$$= 6.35\text{ m}$$

$$\text{voltage drop} = 5\text{ m} \times 4.2\text{ A} \times 18\frac{\text{mV}}{\text{A m}} \times \frac{1}{10^3}\text{ V}$$
$$= 0.38\text{ V}$$

The 2.5 m^2 cable has a maximum current capacity of 27 A, and it produces a voltage drop that is within the 4% allowed. The percentage voltage drop is

$$\text{voltage drop} = \frac{0.38}{12} \times 100\%$$
$$= 3.15\%$$

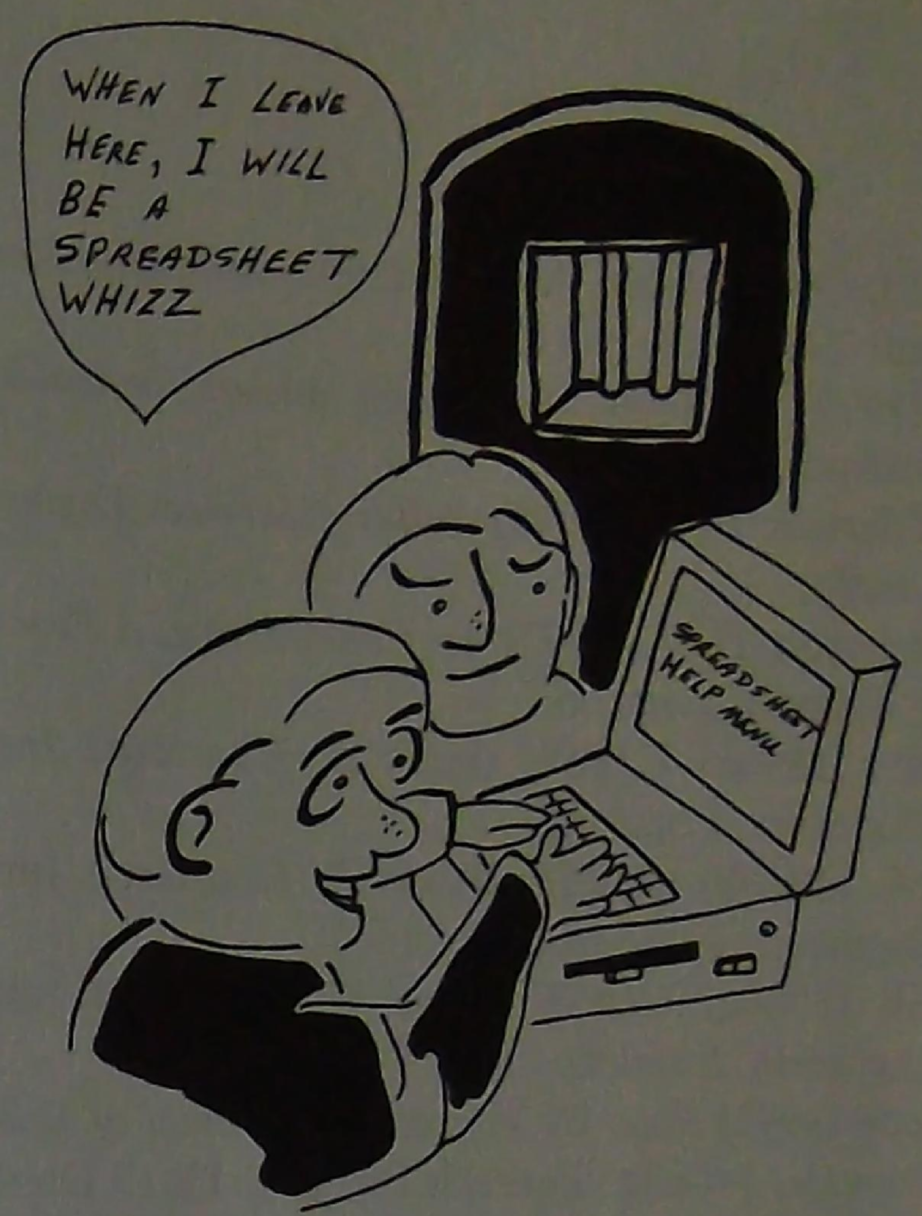
The proposed design complies with the allowable voltage drop and the current capacity.

Questions

Use the data on the worksheet to answer the questions.

1. A three-phase 415 V air-handling unit fan motor has an input electrical power rating of 9.6 kW. The motor power factor is 0.85. The conductor from the main distribution board has a measured length of 37 m. The cable is installed in cable trays with other circuits. Find a suitable cable size.
2. A single-phase 240 V fan coil unit motor has an input electrical power rating of 375 W. The motor power factor is 0.65. The conductor from the main distribution board has a measured length of 18 m. The cable is installed in cable trays with other circuits. Find a suitable cable size.

3. A single-phase 12 V lamp has an input electrical power rating of 35 W. The lamp power factor is 1.0. The conductor from the main distribution board has a measured length of 7.5 m. The cable is clipped onto the building surfaces alongside other cables. Find a suitable cable size.



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