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THE FREQUENCY OF OCCURENCE OF OUTDOOR AIR CONDITIONS IN THE UNITED KINGDOM AND THEIR USE IN PREDICTING ENERGY CONSUMPTION OF AIR CONDITIONING SYSTEMS

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The availability of frequency of occurence of outdoor air conditions for the UK are described. These include distributions of dry-bulb and wet-bulb temperatures, specific enthalpy and dry-bulb temperature in association with moisture content, all of which can be used as BIN data. The application of load and temperature analysis diagrams for the calculation of the annual energy consumption of air conditioning systems is illustrated by means of a number of examples.

FREQUENCY OF OCCURRENCE OF OUTDOOR AIR CONDITIONS

The Meteorological Office has published a set of reports(1) dealing with the combined frequency distribution of dry-bulb temperature and screen wet-bulb temperature for 28 stations throughout the United Kingdom. Details of these sites are given in Reference (1) and the locations are shown on Figure 1. The CIBSE Guide(2) presents data from eight of these stations in Section A2 (Weather and Solar Data), retaining the use of screen wet-bulb an air property not entirely convenient for use by air conditioning engineers.

For 23 of the stations the original data is held on magnetic tape and from this data Collingbourne and Legg(3) obtained percentage frequency distributions for the following air properties:

(i) dry-bulb temperature

(ii) wet-bulb temperature, sling

(iii) specific enthalpy

(iv) dry-bulb temperature in association with moisture content.

These frequency distributions, for the whole of the annual 8760 hourly observations, were for the following daily periods:

(a)	24 hours	
(b)	12 hours	07.00 - 18.00 GMT
	12 hours	19.00 - 06.00 GMT
(c)	8 hours	02.00 - 09.00 GMT
	8 hours	10.00 - 17.00 GMT
	8 hours	18.00 - 01.00 GMT

TABLE 1 Percentage frequency distribution of hourly values of outdoor dry-bulb temperature, annual 24 hour periods, Heathrow

dry-bulb temperature °C 1 K intervals	frequency f_1	dry-bulb temperature °C 1 K intervals	frequency f_1	dry-bulb temperature °C 1 K intervals	frequency f_1
		0.0 to 0.9	2.19	20.0 to 20.9	1.71
		1.0 to 1.9	2.78	21.0 to 21.9	1.27
		2.0 to 2.9	3.30	22.0 to 22.9	0.98
		3.0 to 3.9	3.85	23.0 to 23.9	0.63
		4.0 to 4.9	4.16	24.0 to 24.9	0.46
		5.0 to 5.9	5.12	25.0 to 25.9	0.31
<-13.1	-	6.0 to 6.9	5.46	26.0 to 26.9	0.20
-13.0 to-12.1	0.00	7.0 to 7.9	5.82	27.0 to 27.9	0.12
-12.0 to-11.1	0.00	8.0 to 8.9	6.21	28.0 to 28.9	0.07
-11.0 to-10.1	0.01	9.0 to 9.9	6.01	29.0 to 29.9	0.06
-10.0 to -9.1	0.01	10.0 to 10.9	6.08	30.0 to 30.9	0.04
-9.0 to -8.1	0.01	11.0 to 11.9	5.76	31.0 to 31.9	0.02
-8.0 to -7.1	0.02	12.0 to 12.9	5.63	32.0 to 32.9	0.02
-7.0 to -6.1	0.04	13.0 to 13.9	5.47	33.0 to 33.9	0.01
-6.0 to -5.1	0.08	14.0 to 14.9	5.08	34.0 to 34.9	0.00
-5.0 to -4.1	0.16	15.0 to 15.9	4.97	> 35.0	-
-4.0 to -3.1	0.30	16.0 to 16.9	4.40		
-3.0 to -2.1	0.51	17.0 to 17.9	3.57		
-2.0 to -1.1	0.83	18.0 to 18.9	2.82		
-1.0 to -0.1	1.38	19.0 to 19.9	2.09		

TABLE 2 Percentage frequency distribution of hourly values of outdoor sling wet-bulb temperature, annual 24 hour periods, Heathrow

wet-bulb temperature °C 1 K intervals	frequency f_2	wet-bulb temperature °C 1 K intervals	frequency f_2	wet-bulb temperature °C 1 K intervals	frequency f_2
		0.0 to 0.9	2.77	20.0 to 20.9	0.13
		1.0 to 1.9	3.46	21.0 to 21.9	0.04
		2.0 to 2.9	4.18	22.0 to 22.9	0.01
		3.0 to 3.9	4.76	>23.0	-
		4.0 to 4.9	5.63		
		5.0 to 5.9	5.89		
<-13.1	-	6.0 to 6.9	6.47		
-13.0 to-12.1	0.00	7.0 to 7.9	6.85		
-12.0 to-11.1	0.00	8.0 to 8.9	6.78		
-11.0 to-10.1	0.01	9.0 to 9.9	6.94		
-10.0 to -9.1	0.01	10.0 to 10.9	6.38		
-9.0 to -8.1	0.01	11.0 to 11.9	6.67		
-8.0 to -7.1	0.03	12.0 to 12.9	6.72		
-7.0 to -6.1	0.06	13.0 to 13.9	6.39		
-6.0 to -5.1	0.11	14.0 to 14.9	5.36		
-5.0 to -4.1	0.23	15.0 to 15.9	3.98		
-4.0 to -3.1	0.47	16.0 to 16.9	2.77		
-3.0 to -2.1	0.80	17.0 to 17.9	1.38		
-2.0 to -1.1	1.38	18.0 to 18.9	0.70		
-1.0 to -0.1	2.29	19.0 to 19.9	0.35		

(temperatures measured at a frequency <0.005% are listed as 0.00)

Typical 24 hour frequency distributions for Heathrow for dry-bulb temperature, sling wet-bulb temperature and specific enthalpy are given in Tables 1, 2 and 3 respectively. In each of these tables, the frequency of occurence of the measured air condition, for a range of the air property, is given as a percentage of the total hours in the year. The frequency of occurence of the dry-bulb temperature in association with moisture content is given on a psychrometric chart in Figure 2; in this case each 'box' (or bin) contains the (percentagex100) of the measured air conditions which were within ranges of 2/K dry-bulb temperature and 0.001 kg/kg_{da} moisture content. All the values given as 0.00% (zero in Figure 2) are the extreme conditions recorded with a frequency of less than 0.05% of the total annual hours for each daily period. This data may be used:

- to determine outdoor air design conditions
- to investigate system operating requirements at off-peak conditions
- as BIN data for annual energy consumption calculations.

TABLE 3 Percentage frequency distribution of hourly values of outdoor specific enthalpy, annual 24 hour periods, Heathrow

(enthalpies measured at a frequency <0.005% are listed as 0.00)

specific enthalpy 2 kJ/kg intervals	frequency f_3	specific enthalpy 2 kJ/kg intervals	frequency f_3
-10.0 to -8.1	**0.00**	30.0 to 31.9	**5.58**
-8.0 to -6.1	**0.01**	32.0 to 33.9	**5.54**
-6.0 to -4.1	**0.01**	34.0 to 35.9	**5.36**
-4.0 to -2.1	**0.03**	36.0 to 37.9	**5.06**
-2.0 to -0.1	**0.07**	38.0 to 39.9	**4.40**
0.0 to 1.9	**0.20**	40.0 to 41.9	**3.72**
2.0 to 3.9	**0.46**	42.0 to 43.9	**2.99**
4.0 to 5.9	**0.92**	44.0 to 45.9	**2.16**
6.0 to 7.9	**1.67**	46.0 to 47.9	**1.42**
8.0 to 9.9	**2.89**	48.0 to 49.9	**0.85**
10.0 to 11.9	**3.45**	50.0 to 51.9	**0.51**
12.0 to 13.9	**4.33**	52.0 to 53.9	**0.33**
14.0 to 15.9	**4.98**	54.0 to 55.9	**0.18**
16.0 to 17.9	**5.43**	56.0 to 57.9	**0.10**
18.0 to 19.9	**6.21**	58.0 to 59.9	**0.05**
20.0 to 21.9	**6.18**	60.0 to 61.9	**0.02**
22.0 to 23.9	**6.45**	62.0 to 63.9	**0.01**
24.0 to 25.9	**6.23**	64.0 to 65.9	**0.00**
26.0 to 27.9	**6.18**	>66.0	-
28.0 to 29.9	**6.01**		

Design Conditions from Frequency of Occurence Tables

The tables of frequency of occurrence of hourly values can be used to determine the outdoor design conditions, giving a more rational solution than the CIBSE(4) method using standard meteorological data. Though the frequencies of dry-bulb temperature, wet-bulb temperature and specific enthalpy are not coincident with each other they can be considered to be self-consistent since the air properties were derived from the same set of measurements.

When using the frequency tables, a precise calculation of an air condition will not normally be necessary; the nearest 0.5 K is sufficiently accurate for temperatures and 1 kJ/kg_{da} for enthalpies.

Summer design conditions. To determine a design condition for summer, a 'target' of a total accumulated percentage (percentile) of annual hours is set at which the outdoor air condition is reached or exceeded. This percentile is a considered judgement made by the design engineer. The individual frequencies of the appropriate air property are then summed from the extreme condition until the target is reached within reasonable accuracy.

Example 1 Determine the outdoor design conditions for an air conditioning system for a building near Heathrow, London. A 99.65% percentile (equivalent to the design condition being exceeded for 30 hours per year) is considered appropriate.

Solution: To obtain the summer design conditions, the frequencies are summed commencing from the maximum property in the table until the target of (100-99.65)% is reached. All 0.00% values are ignored. Considering specific enthalpy in Table 3; commencing at the extreme value of 64 kJ/kg_{da}, the calculation is tabulated as follows:

specific enthalpy kJ/kg_{da} intervals	frequency f_3
54.0 to 55.9	0.18
56.0 to 57.9	0.10
58.0 to 59.9	0.05
60.0 to 61.9	0.02
62.0 to 63.9	0.01
$\Sigma(f_3)$	0.36

design condition = 54 kJ/kg_{da}

In the same way calculations are made for other air properties and the summer design conditions will therefore be:

dry-bulb temperature	$27^{o}C$
wet-bulb temperature	$19.5^{o}C$
specific enthalpy	54 kJ/kg_{da}

Winter design conditions. A similar procedure may be adopted for winter design conditions. In this case a target percentile is set in which the air property is at or below the design condition. The individual frequencies of the air property being considered are then summed from the extreme minimum condition until the percentile is reached.

Example 2 Determine the outdoor design conditions for an air conditioning system which is being specified for a building near Heathrow, London. A percentile of 0.35% (equivalent to 30 hours per year) is considered suitable.

Solution: To obtain the winter design conditions, the frequencies are summed commencing from the minimum property in the table until the target percentile of 0.35% is reached. All 0.00% values are ignored. Considering dry-bulb temperature in Table 1; commencing at the extreme value -11°C the calculation is tabulated as follows:

dry-bulb temperature intervals	frequency f_1
-11.0 to -10.1	0.01
-10.0 to -9.1	0.01
-9.0 to -8.1	0.01
-8.0 to -7.1	0.02
-7.0 to -6.1	0.04
-6.0 to -5.1	0.08
-5.0 to -4.1	0.16
$\Sigma(f_1)$	0.33

design condition = -4°C

In the same way calculations are made for other air properties. The winter design conditions will therefore be:

dry-bulb temperature	-4.0°C
wet-bulb temperature	-4.5°C
specific enthalpy	2 kJ/kg_{da}

The choice of which air properties to use as the design condition(s) will depend on the application. For example, dry-bulb temperature may be used for sensible heating processes, wet-bulb temperatures for cooling tower selection and enthalpy for dehumidifying cooling coils.

Outdoor Air Condition Envelope

Air conditioning systems will usually be required to operate throughout the year against a whole range of outdoor air conditions. The extent of these conditions is most usefully shown on what may be termed the **outdoor air condition envelope**, based on the frequency distributions similar to that shown in Figure 2.

These envelopes provide a useful tool for summarizing plant operations. The design conditions of dry-bulb temperature and specific enthalpy, determined in Examples 1 and 2, are shown on the skeleton envelopes in Figure 3. The shaded areas on each envelope represent the **total** accumulated frequency in which the outdoor air condition falls above, or below, the design condition.

An example of the use of these envelopes is given in Example 4, Figure 8.

ANNUAL ENERGY CONSUMPTION OF AIR CONDITIONING SYSTEMS

The ability to calculate the predicted annual energy consumption and energy costs of air conditioning systems is required for the following reasons:

* to allow a comparison of systems on a total cost basis, before a selection is made.

* to optimize the design and control of the selected system to achieve minimum energy consumption.

* to inform the client of the expected costs for budgeting purposes.

* to provide a basis for the measurement of the energy performance of the installed system during its working life.

The method of determining the average annual energy consumption for heating systems using the concept of degree-days is well established. There is no such commonly accepted method for calculating the energy consumption of air conditioning systems. It is possible to use computer programs with detailed mathematical models of a building, the weather, the systems and hour-by-hour operation to assess energy consumption. This method produces custom-built models which involve a large amount of time and money for the model's construction. The approach described below is a relatively simple technique for predicting energy consumption, providing relevant data for cost comparisons between alternative designs. This method was first proposed by Robertson(5) and amplified by Legg(6). It is essentially the same as the BIN method given in the ASHRAE Guide(7). But whereas the BIN method limits itself to a base of dry-bulb temperature, the general method described below offers greater flexibility in the choice of outdoor conditions.

Basis for Calculation of Annual Energy Costs

The annual energy cost for the complete air conditioning system(s) is obtained through the summation of the energy demands of each individual plant item within the system demanding energy. These include the following:

- heaters and boiler plant.

- coolers and refrigerating plant.

- steam humidifiers.

- fans for central plant supply and extract from air conditioned and mechanically ventilated spaces, cooling towers and room units.

- pumps for heating, cooling and humidification water systems.

- compressors for unitary systems.

Some of the plant items will run at constant load in which case the annual consumption is simply the annual hours of operation multiplied by the installed load. Where plant load varies because of variations in outdoor conditions and internal heat gains then it will be necessary to analyse the load variations according to the annual frequency of occurrence of the load variations.

The method is as follows:

(a) establish the relationship between energy demand and an appropriate climate factor.

(b) integrate the variations in energy demand with the annual frequency distribution of the climate factor.

The technique is illustrated in Figure 4. The load on the plant item being considered is plotted against the outdoor climate factor for which there is a known frequency of occurrence. Since, in this case, the load is increasing with outdoor conditions, the diagram would represent an cooler battery in an air conditioning system.

Some of the features of this diagram are:

- The load profile is **ABCDE**.
- The plant is switched on at **A**.
- The load **AB** is constant and is termed a <u>base load</u>. A base load may be due to miscellaneous heat gains and losses to pipework. This occurs at a total accumulated frequency in the range **ab**.
- The load varies between **BC** in the frequency range **bc**. The mode of operation of the plant changes at **C** and the load varies between **CD** in the frequency range **cd**.
- At point **P** there is a load **Q** which occurs with frequency **f**.
- Between **DE** the load is constant at the design or installed load. This occurs at a total accumulated frequency in the range **de**.

Where both the load and its frequency of occurrence of the load are varying it is necessary to accumulate the sum of their product. The average annual energy consumption is then given by:

for cooling loads:

$$E_c = \frac{1}{100} \sum \frac{f\dot{Q}}{COP} T \qquad (1)$$

for heating loads:

$$E_h = \frac{1}{100} \sum \frac{f\dot{Q}}{\eta} T \qquad (2)$$

where η = heating system part-load efficiency

COP = cooling system part-load coefficient of performance.

From this calculation the annual energy costs may be obtained from the unit costs of the fuel.

This theoretical approach is particularly suitable to the analysis of the cost-benefit of additional equipment added to a system to make it more energy efficient. In this case only the analysis of the <u>difference</u> in energy costs would be required. With manual calculations a width of the BIN would be chosen to suit the calculation. Too small a range leads to a lengthy calculation; too large a range leads to reduced accuracy and flexibility.

The examples which follow illustrate the method of calculation.

Example 3 The air conditioning system (System 1) shown in Figure 5 maintains a constant ratio of fresh to recirculated air quantities and the off-coil condition, B, is controlled by the dew-point sensor, T_1. Calculate the energy consumption for the cooler for the following criteria:

mass air flow rate (assumed constant)	2.5 kg/s
air enthalpies:	
recirculation, h_R	42 kJ/kg
off-coil (T_1), h_B	30 kJ/kg
outdoor air, design condition	54 kJ/kg

Table 4 Calculation of average hourly energy demand of cooler operating in System 1

Range of outdoor air specific enthalpy (kJ/kg_{da})	mid-point of range h_o (kJ/kg_{da})	frequency f_3	Δh_1	$f_3\ \Delta h_1$	notes
< 2.0	-3	0.32	0.75	0.2	use Equation 3
2 - 5.9	2	1.38	2.5	3.5	
6 - 9.9	8	4.65	3.5	16.3	$\Delta h_1 = 0.25\ h_o + 1.5$
10 - 13.9	12	7.78	4.5	35.0	
14 - 15.9	16	10.41	5.5	57.3	
18 - 21.9	20	12.39	6.5	80.5	
22 - 25.9	24	12.68	7.5	94.7	
26 - 29.9	28	12.19	8.5	103.6	
30 - 33.9	32	11.12	9.5	105.6	
34 - 37.9	36	10.42	10.5	109.4	
38 - 41.9	40	8.12	11.5	93.4	
42 - 45.9	44	5.15	12.5	64.4	
46 - 49.9	48	2.27	13.5	30.6	
50 - 53.9	52	0.84	14.5	12.2	
> 54	-	0.36	15.0	5.4	design load

$\Sigma(f_3 \Delta h_1) = 812.1$

Table 5 Calculation of average hourly energy demand of cooler operating in System 2

Range of outdoor air specific enthalpy (kJ/kg_{da})	mid-point of range h_o (kJ/kg_{da})	frequency f_3	Δh	$f_3\ \Delta h$	notes
< 30	-	-	-	-	cooler off
30 - 33.9	32	11.12	2	22.2	cooler operates on
34 - 35.9	36	10.42	6	62.5	100% outdoor air
38 - 41.9	40	8.12	10	81.2	Use Equation 4
					$\Delta h_3 = h_o - 30$
42 - 45.9	44	5.15	12.5	64.4	cooler operates on
46 - 49.9	48	2.27	13.5	30.6	min. outdoor air
50 - 53.9	52	0.84	14.5	12.2	Use Equation 3
					$\Delta h_1 = 0.25\ h_o + 1.5$
>54	-	0.36	15.0	5.4	design load

$\Sigma(f\Delta h) = 278.5$

Table 6 Calculation of the difference in average hourly energy demand of cooler operation, System 1 compared to System 2

Range of outdoor air specific enthalpy (kJ/kg$_{da}$)	mid-point of range h_o (kJ/kg$_{da}$)	frequency f_3	Δh	$f_3\ \Delta h$	notes
< 2.0	-3	0.32	0.75	0.2	use Equation 3
2 - 5.9	2	1.38	2.5	3.5	$\Delta h_1 = 0.25\ h_o + 1.5$
6 - 9.9	8	4.65	3.5	16.3	
10 - 13.9	12	7.78	4.5	35.0	
14 - 17.9	16	10.41	5.5	57.3	
18 - 21.9	20	12.39	6.5	80.5	
22 - 25.9	24	12.68	7.5	95.1	
26 - 29.9	28	12.19	8.5	103.6	
30 - 33.9	32	11.12	7.5	83.4	use Equation 5
34 - 37.9	36	10.42	4.5	46.9	$\Delta h_3 = 31.5 - 0.75\ h_o$
38 - 41.9	40	8.12	1.5	12.2	
			$\Sigma(f_3\ \Delta h)$ =	534.0	

Table 7 Calculation of average hourly energy saving of heat recovery unit in system 3

Range of outdoor air dry-bulb temperature (°C)	mid-point of range t_o (°C)	frequency f_1	Δt_1	$f_1 \Delta t_{HRU}$	notes
< -4.1	-	0.33	16.8	5.5	design load*
-4.0 to -1.1	-2.5	1.64	15.8	25.8	HRU at full output
-1.0 to 2.1	0.5	6.53	13.7	89.1	use Equation 6
2.0 to 5.1	3.5	11.31	11.6	130.6	$\Delta t_{HRU} = 14 - 0.7t_o$
5.0 to 7.9	6.5	16.40	9.4	155.0	
8.0 to 9.9	9.0	12.22	7.7	94.1	
10.0 to 11.9	11.0	11.84	4.6	55.1	HRU at reduced load
12.0 to 13.9	13.0	13.11	1.8	22.9	use Equation 7
					$\Delta t_{HRU} = 19.6 - 1.35t_o$
			$\Sigma(f_1\ \Delta t_{HRU})$ =	578.1	

(Note: * Design load is that of a conventional heater)

Solution: A summary of the plant operation is given in Figure 8. With the outside air above the set point of C_1 the cooling load is the same as for System 1, i.e. use Equation 3. When the fresh air enthalpy is between the set points of T_1 and C_1 the system operates on 100% outdoor air (x = 1.0) and the cooling load becomes:

$$\dot{Q} = \dot{m}\,(h_O - h_B)$$

$$\text{or } \Delta h_2 = h_O - 30 \qquad (4)$$

The load profile for the operation of this system is given in Figure 9. The numerical integration is completed in Table 5.

The accumulated sum of the product $(f_3\,\Delta h_1) = 278.5$

Using Equation 1, the annual energy consumption is obtained:

$$E_c = \dot{m}\,\frac{1}{COP}\sum\frac{f\dot{Q}}{100}\,T$$

$$= 2.5\ \frac{1}{3.8}\ \frac{278.5}{100}\ 8760$$

$$= 16{,}776 \text{ kWh}$$

The two systems in Examples 3 and 4 are alternative methods of treating the recirculated air, the addition of the modulating dampers making System 2 more energy efficient than System 1. The difference between the energy consumption between the two systems is then given by:

$$46{,}826 - 16{,}050 = 30{,}776 \text{ kWh/annum}$$

The theoretical percentage energy saving is:

$$\frac{30{,}776}{46826} \times 100 = 66\%$$

The difference in annual energy consumption can be costed as part of a cost benefit calculation for the additional dampers, ductwork and controls.

Shorter Route to Determine the Energy Consumption Differences

If only the difference between the systems is required this can be obtained by a shorter route as shown in the following example, which compares the two systems in the previous examples.

Example 5. Determine the difference in energy consumption between Systems 1 and 2.

Solution: The load profiles for Systems 1 and 2 are shown in Figures 6 and 9 respectively. If these two diagrams are superimposed, Figure 10 is obtained. The difference in energy consumption is then represented by the area **BDC**. The intervals of enthalpy difference of this area are then integrated with the frequency of specific enthalpy in convenient ranges.

for $h_O < 30$ Equation 4 applies

In the enthalpy range 30 - 42 kJ/kg_{da} the enthalpy difference is given by:

$$\Delta h_3 = \Delta h_1 - \Delta h_2$$

$$\Delta h_3 = (0.25\ h_o + 1.5) - (h_o - 30)$$

$$\Delta h_3 = 31.5 - 0.75\ h_o \qquad (5)$$

The numerical integration is completed in Table 6.

The accumulated sum of the product $(f_3 \Delta h) = 534.0$

Using Equation 10.1, the annual energy consumption is obtained:

$$E_c = m \frac{1}{COP} \sum \frac{fQ}{100} T$$

$$= 2.5 \frac{1}{3.8} \frac{534}{100} 8760$$

$$= 30{,}776 \text{ kWh}$$

Which is the same as the energy saving calculated previously.

Load Diagrams

The variation of heat gains and losses to either a building, zone of a building or an individual room, can be plotted against outdoor air temperature and the resulting graph is known as a load diagram or load chart. These diagrams were originally(?) described in the classic book 'Buffalo Forge'(8), developed by Higham(9) and Appleby(10), and now accepted in the latest edition of the CIBSE Guide(10). Load profiles may be obtained from design calculations or from BIN data such as those published by Letherman and Dewsbury(12). The diagrams are used to assist with system design, planning the economic operation of the plant and for estimating annual energy demands.

Referring to Figure 11. The line **AB** is the transmission heat loss (or gain) $\Sigma(UA/\Delta t)$; this line may also include heat loss due to air infiltration. Note that **AB** passes through point **R** where the outdoor air temperature is equal to the room temperature t_R and where the transmission loss is zero. It is usual for the design total internal sensible heat gain from occupants, lights and electrical equipment to be considered constant throughout the year. When the total internal heat gain is added to the transmission line it produces line **CD** parallel to **AB**. To complete the diagram, the maximum solar heat gains calculated at the summer and winter design conditions are added to line **CD** to produce **EF** the line of maximum heat gain. The area **ABFE** represents the complete load variation for the room between the limits of the outdoor summer and winter design conditions.

Figure 11 is a load diagram suitable for a zone on a west face of a building. For most orientations **EF** will be a dog-leg **EFF'**, as in Figure 12 where the maximum solar gain does not coincide with the maximum outdoor temperature.

For constant air flow rate systems supply air temperature lines may be drawn in conjunction with the load diagram as shown in Figure 13. The minimum supply temperature, t_S, corresponds to the maximum heat gain at **A**, the temperature difference $(t_R - t_S)$ being the design cooling temperature differential used to determine the air flow rate. In a constant air flow rate system, as the room sensible heat gain decreases the supply air temperature increases to give the line **ST**. Similarly the maximum supply air temperature will correspond to the maximum heat loss at **A** to give the point **U**, while the supply temperature line **UV** corresponds to the transmission line at a particular value of t_o. The supply air temperature will therefore be at any point located within the area **STUV**, depending on the actual sensible heat gain or loss to the air conditioned space.

The energy consumption of most systems will depend on the frequency of occurrance of the loads within the load diagram. The BIN method would allow this to be done relatively easily by incorporating within the load diagram the predicted frequency of occurrence of the heat gains within each bin. The corresponding frequency of occurrence of the air supply temperatures for each of these loads can then be obtained. However, the method suggested by Legg(6) and in the ASHRAE Guide(7) is to produce a mean load line. To obtain this, the heat gains are multiplied by appropriate estimated load diversity factors based on occupancy levels, use of heat generating equipment and average levels of bright sunshine hours published by the Meteorological Office(13).

Scheduling

To improve the operating efficiency of some systems, temperatures within the plant can be scheduled to outdoor air dry-bulb temperatures. The schedule is determined from a temperature analysis diagram. The analysis can also include the calculation of the energy consumption and this is illustrated by the following example of the use of an air-to-air heat recovery unit in an 100% outdoor air system.

Example 6. The air conditioning system shown in Figure 14 is required to maintain two laboratories at 20^{o}C for 24 hours per day and for 7 days per week. The output of the heat recovery unit is controlled by sensor T_1, reset by sensor T_2, to maintain a scheduled temperature at point **B**. Using the design data, determine the annual heating energy saved by using a heat recovery unit in place of a conventional preheater.

Design data:

Mass air flow rate (assumed constant)	2.5 kg/s
Outdoor air design condition	- 4^{o}C
Load diagram for laboratories given in Figure 12	
Climate data for Heathrow	
Boiler firing efficiency (average)	75%

The temperature rise across the HRU related to the outdoor dry-bulb temperature, t_O, by the equation:

$$\Delta t_{HRU} = 14 - 0.7t_O \qquad (6)$$

Solution: The temperature analysis is prepared on Figure 15. Line **ST** is the variation of minimum supply temperatures required to meet maximum heat gains of the load diagram and is the 'ideal' schedule line. Line **XY** is the dry-bulb temperature of the outdoor air entering the system. Line **LNR** is the temperature after the heat recovery unit, based on the temperature rise expressed by Equation 6. To meet the requirements of the schedule line **STU**, the output of the heat recovery unit has to be reduced from point **N** until zero load at point **P**. The heating requirements of the HRU are then represented by the shaded area **XLNP** (also extends below the winter design condition). This will give the energy saving compared to a conventional preheater.

Since the mass flow rate, m_a, and the humid specific heat, c_{pas}, remain constant, they can be omitted temporarily from the calculations until the numerical integration has been completed. The load variations can therefore be expressed in terms of temperature differences.

At the outdoor design condition of -4^{o}C the temperature rise would be that for a conventional heater. Up to 10^{o}C Equation 6 applies. Between 10^{o}C and 14^{o}C the temperature rise across the heat recovery unit will be the difference between line **UNPT** and the outdoor temperature.

Equation of line **UNPT**:

$$t_s = 19.4 - 0.35\ t_o$$

$$\therefore \Delta t_{HRU} = 19.6 - 0.35\ t_o - t_o$$

$$\therefore \Delta t_{HRU} = 19.6 - 1.35\ t_o \qquad (7)$$

The numerical integration is completed in Table 7.

The accumulated sum of the product $(f_1 \Delta t) = 578$

Using Equation 2, the annual energy consumption is obtained:

$$E_c = \dot{m}\ c_{pas} \frac{1}{\eta} \sum \frac{fQ}{100}\ T$$

$$= 2.5 * 1.02 \frac{1}{0.75} \frac{578}{100}\ 8760$$

$$= 1.72 \times 10^5 \text{ kWh}$$

The annual energy saving can be costed and compared with the additional capital cost of the heat recovery unit and associated ductwork, filters and controls. Account must also be taken of increased fan energy and miscellaneous equipment energy consumption costs.

Variations in System Efficiency

In the examples given in this paper COP and boiler efficiency have been considered constant. If variation of these two parameters are known they can easily be associated with each bin and included in the calculations. The technique was used by Wong and Legg(14) to investigate the economic application of variable compressor speed for refrigeration system capacity control.

DISCUSSION

The frequency of occurrence of hourly values of outdoor air conditions in the United Kingdom has been illustrated with the tabulated data for Heathrow. Their use for determining design conditions, outdoor air condition envelope and annual energy consumption for air conditioning systems has been demonstrated with a number of examples.

A sense of proportion should be attached to these topics as they are only predictions based on historic data of long term averages. Any one year will differ from the average year and there is no guarantee that the long term averages will remain the same. Though total cost analysis has been likened to 'crystal-ball gazing', this doesn't absolve design engineers from planning for the future with the best techniques available. When comparing the total costs of alternative systems, or system components, it is unlikely that a difference of, say, six months in pay-back period will have a significant effect on the client's decision to go ahead with a particular scheme. Many approximations and assumptions have to be made and therefore high accuracy is not required in the calculations of energy consumption.

SYMBOLS USED

COP	coefficient of performance
E_c	annual energy consumption, cooler
E_h	annual energy consumption, heater
f	frequency of occurrence of outdoor air condition
h_B	specific enthalpy of air leaving cooler
h_M	specific enthalpy of mixed air
h_O	specific enthalpy of outdoor air
h_R	specific enthalpy of room air
$\dot{Q}$	heat flow rate
$\dot{m}$	air mass flow rate
T	hours of operation
t_O	dry-bulb temperature, outdoor air
t_S	dry-bulb temperature, supply air
x	fraction of outdoor air
Δh	enthalpy difference
Δt	temperature difference
η	boiler firing efficiency

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CM Number	Meteorological Station	Map Reference	Height above sea level m	Period Years
10	Croydon	51-312637	69	1946-55
11	Glasgow Airport (Renfrew)	26-508662	8	"
12	Driffield	54-004565	23	"
13	Boscombe Down	41-172403	126	"
14	Manchester Airport (Ringway)	33-818850	75	"
15	Stornoway	19-459332	3	"
16	Lympne	61-111352	104	1946-53
17	Birmingham (Elmdon)	42-176839	96	1949-57
18	Belfast Airport (Aldergrove)	33-147798	68	1946-55
19	Pembroke Dock	12-959039	10	"
20	Mildenhall	52-683779	5	1952-60
35	Edinburgh Airport	36-159739	35	1952-60
39	Kinloss	38-067627	5	1952-61
80	Birmingham Airport (Elmdon)	42-176839	96	1960-74
81	London Airport (Heathrow)	51-077767	25	"
82	Manchester Airport (Ringway)	33-821849	75	"
83	Dishforth/Leeming	44-307890	32	"
84	Waddington	43-988653	68	"
85	Mildenhall/Honington	52-887749	50	"
86	Boscombe Down	41-172403	126	"
87	Manston	61-335666	44	1961-74
88	Thorney Island	41-760026	4	1960-74
89	Valley	23-310758	10	"
90	Aberporth	22-242521	133	"
91	Glamorgan Airport (Rhoose)	31-064679	67	"
92	Plymouth (Mountbatten)	20-492529	27	"
93	Lerwick	41-453397	82	"
94	Wick	39-364522	36	"
95	Stornoway	19-459332	3	"
96	Kinloss	38-067627	5	"
97	Edinburgh Airport (Turnhouse)	36-159739	35	"
98	Tiree	07-999446	9	"
99	Glasgow Airport (Abbotsinch)	26-480667	5	"
100	Prestwick Airport	26-369261	16	"
101	Eskdalemuir	36-235026	242	"
102	Belfast Airport (Aldergrove)	33-147798	68	"

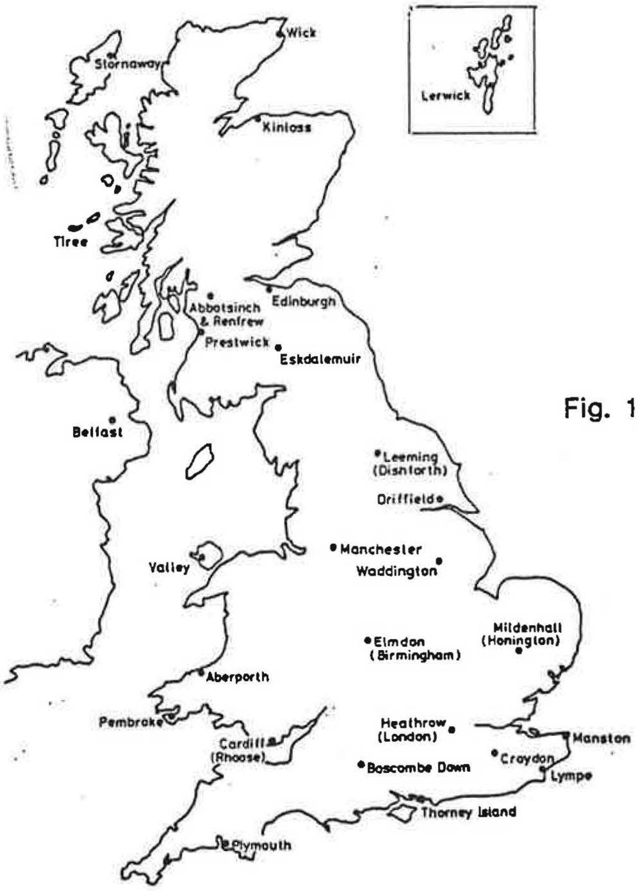

Fig. 1 Location of meteorological stations in the United Kingdom

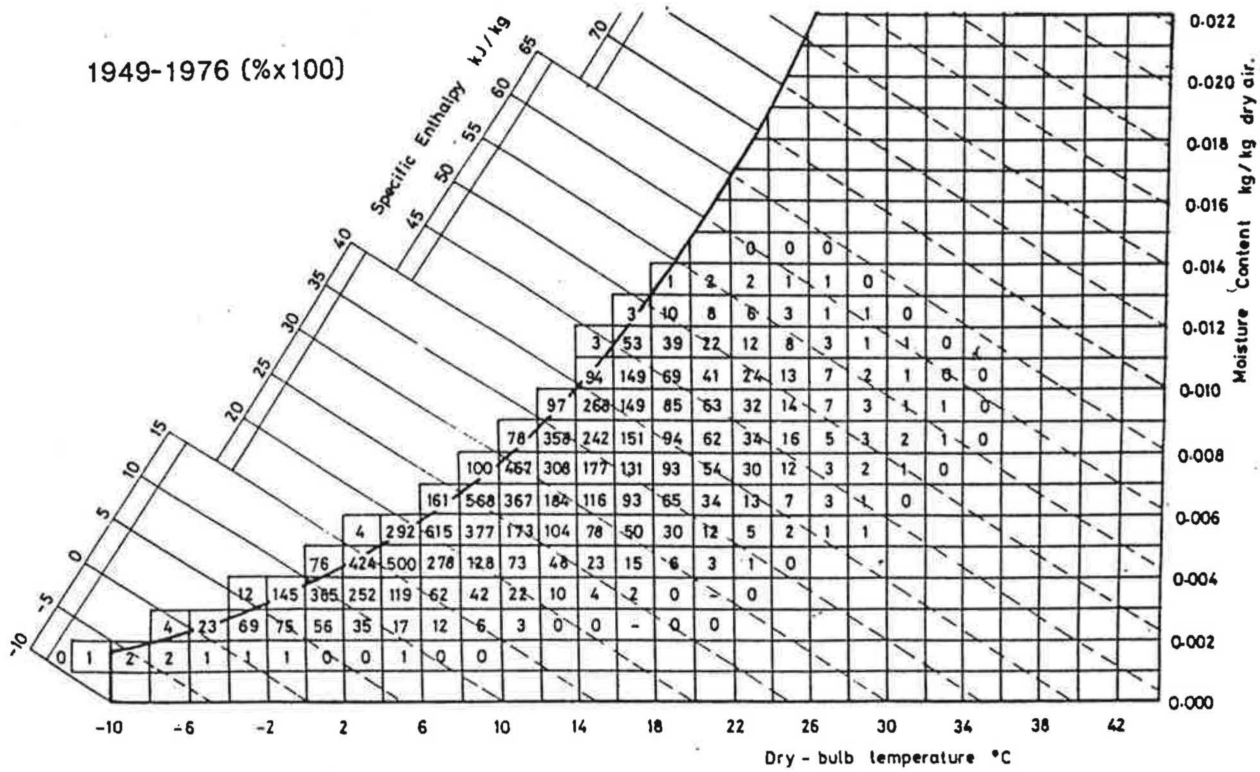

Fig. 2 Annual frequency of occurence of hourly values of outdoor air condition at Heathrow

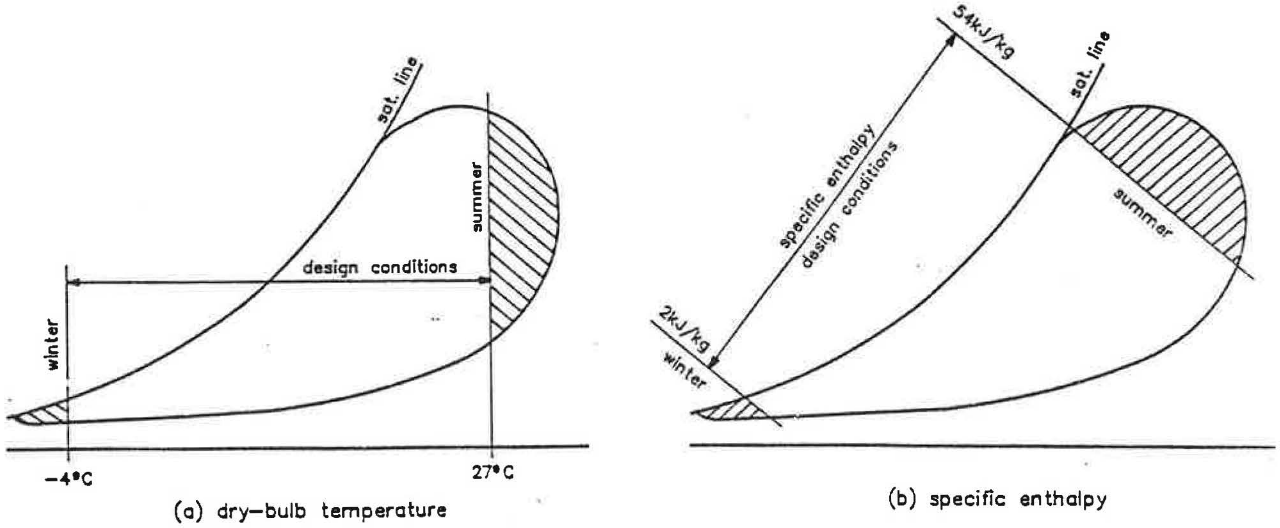

Fig. 3 Outdoor air condition envelope and design conditions, Heathrow

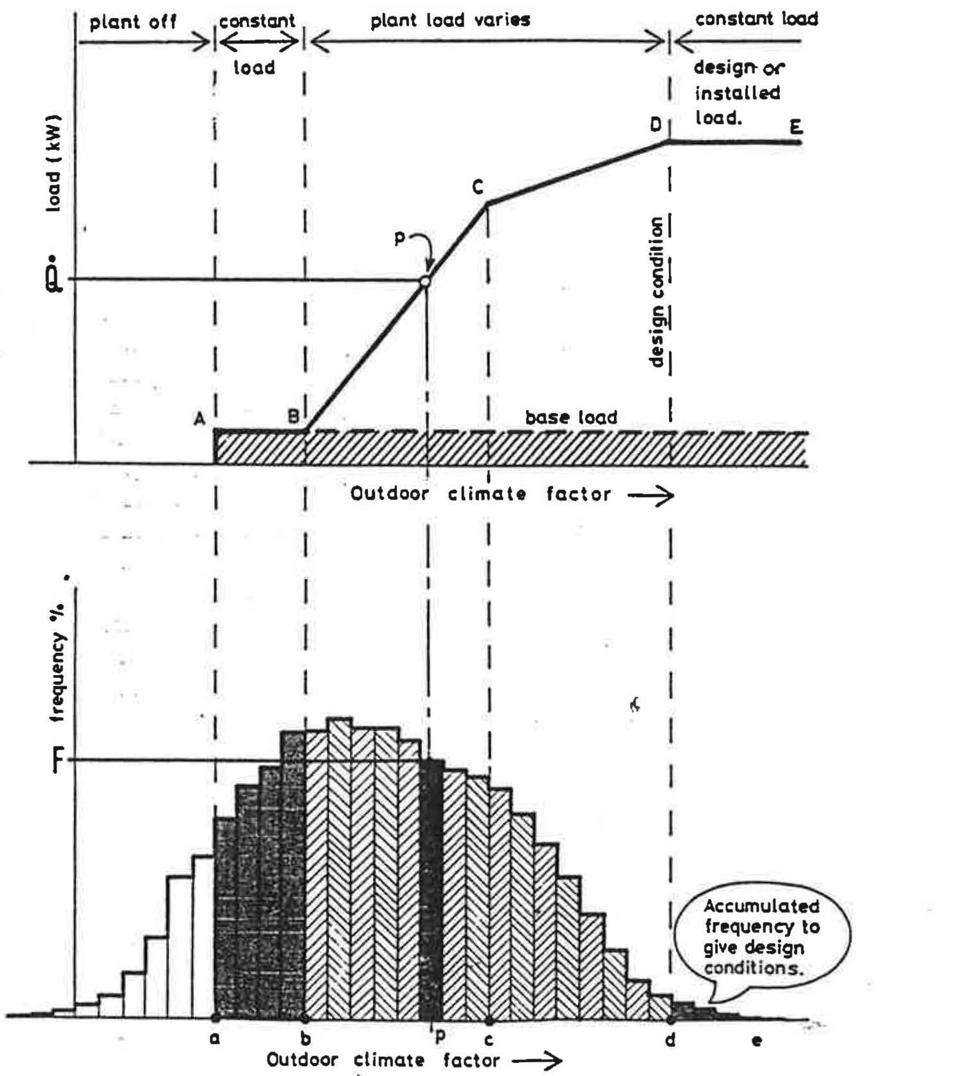

Fig. 4 Plant load versus outdoor climate factor with known frequency of occurrence

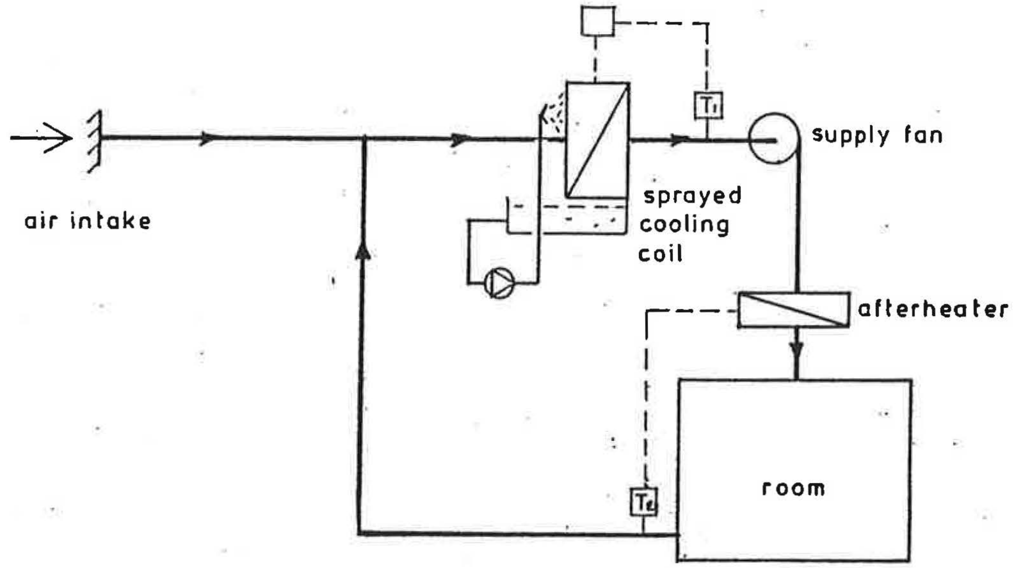

Fig. 5 System 1

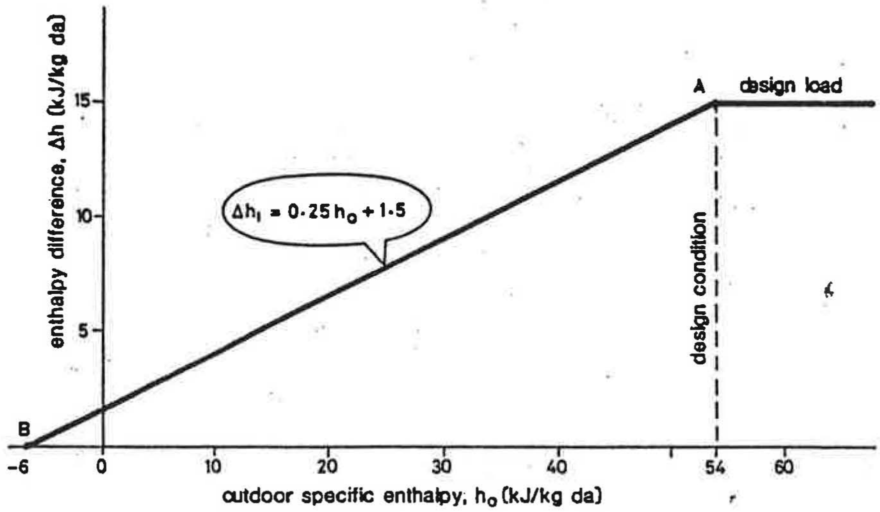

Fig. 6 Load profile, System 1

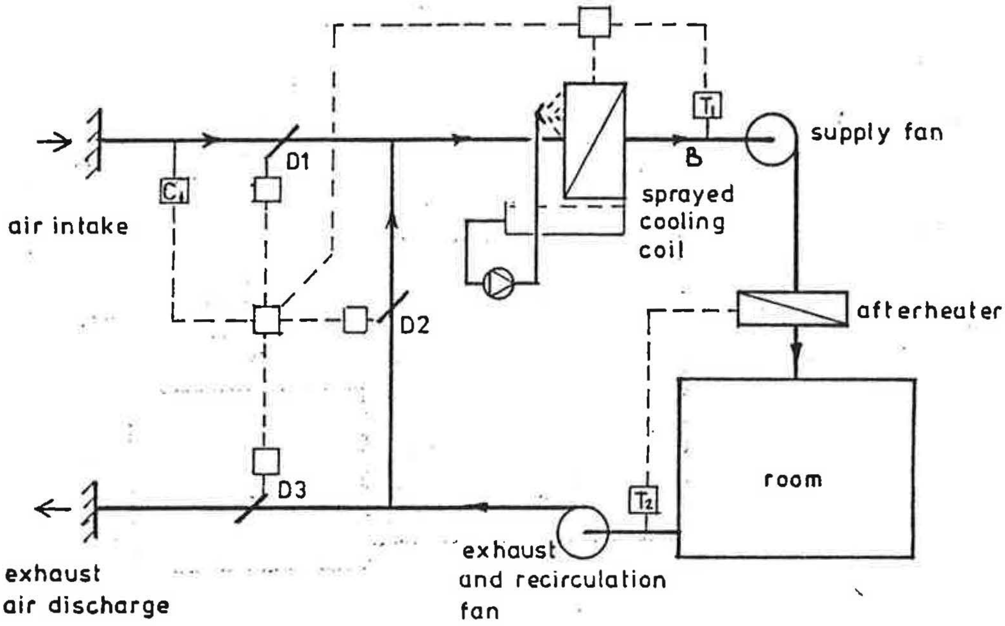

Fig. 7 System 2

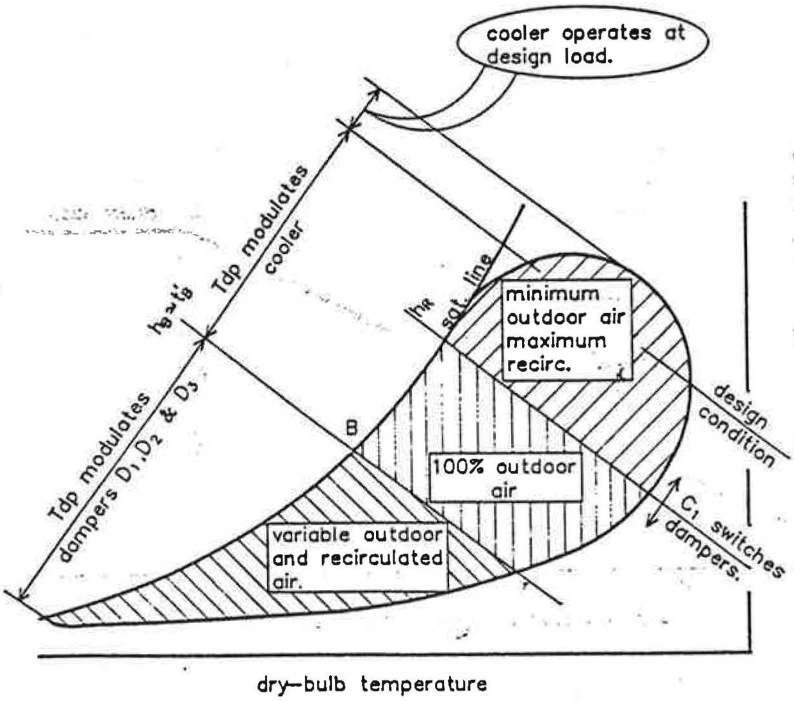

Fig. 8 Summary of plant operation, System 2

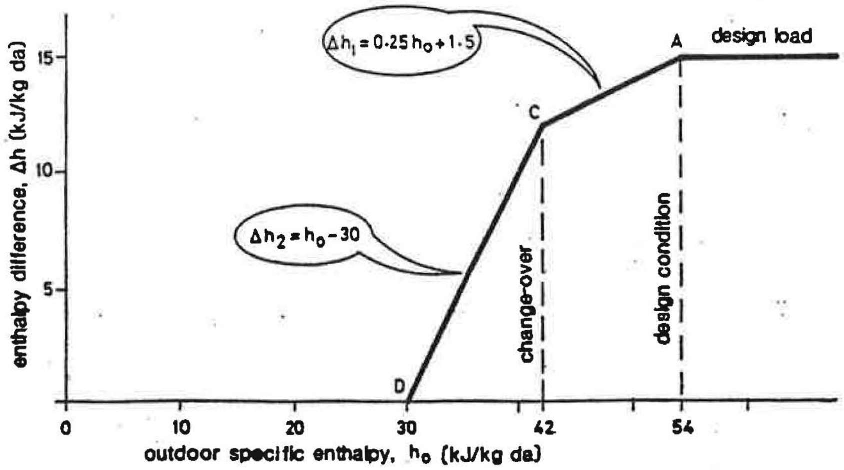

Fig. 9 Load profile, System 2

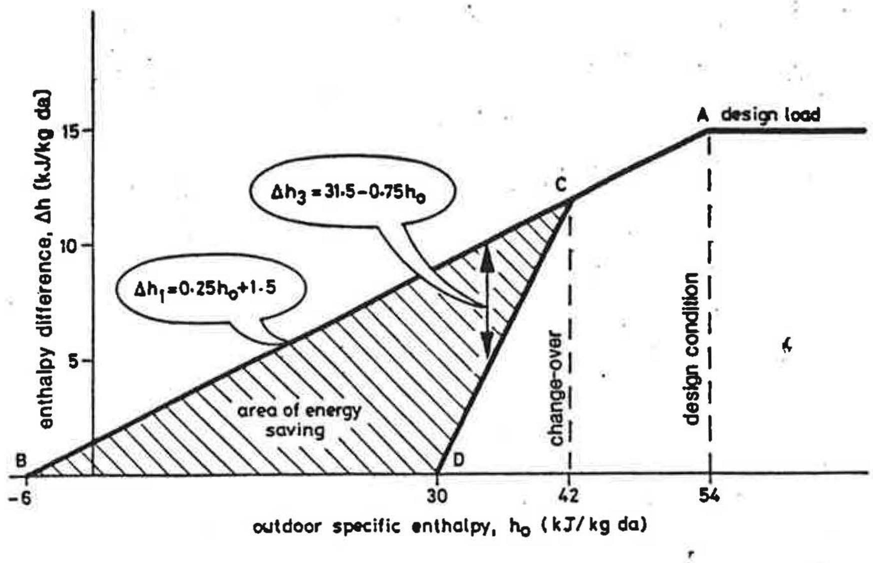

Fig. 10 Combined load profiles, Systems 1&2

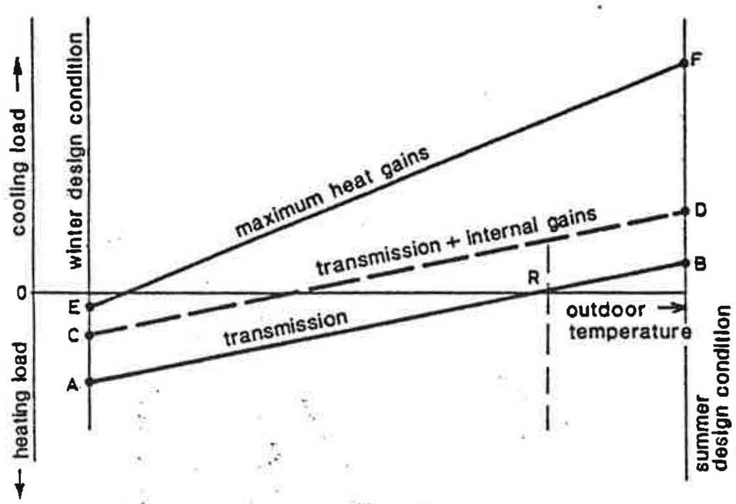

Fig. 11 Load diagram for west orientation

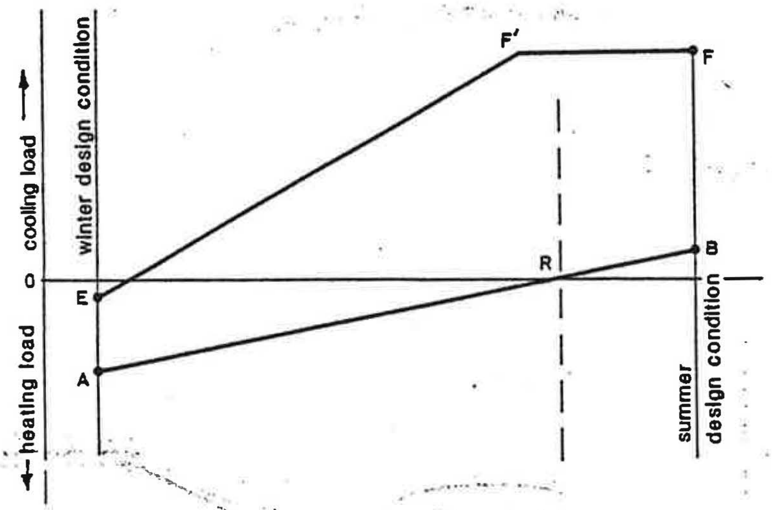

Fig. 12 Load diagram for zone in which peak solar heat gain does not coincide with outdoor design temperature

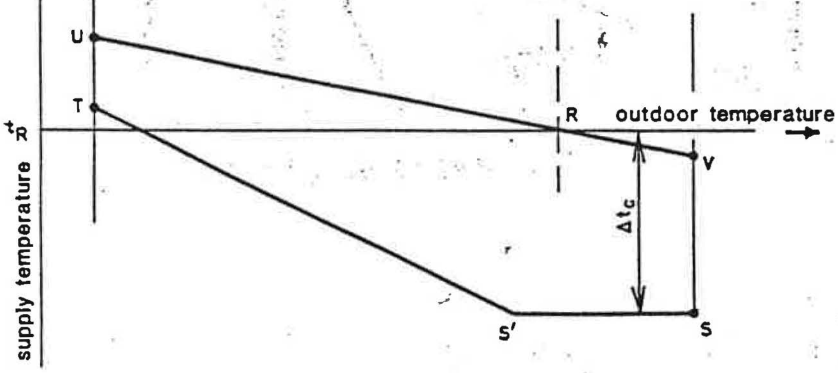

Fig. 13 Supply air temperature diagram

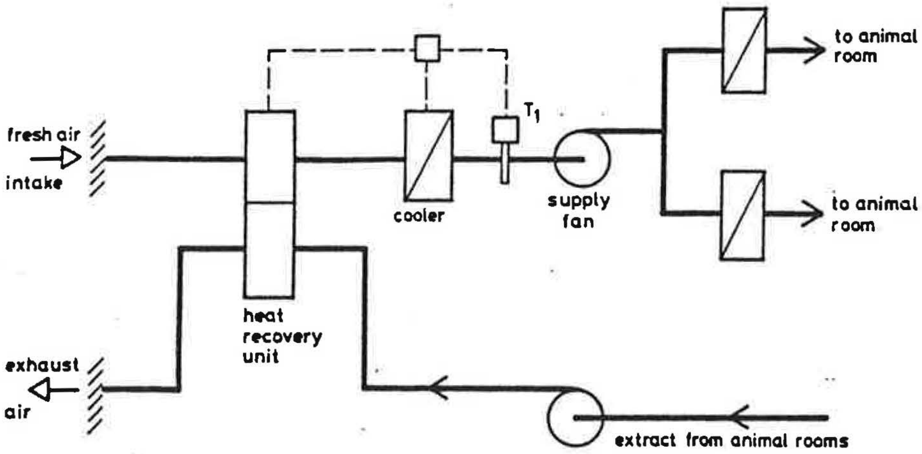

Fig. 14 System 3

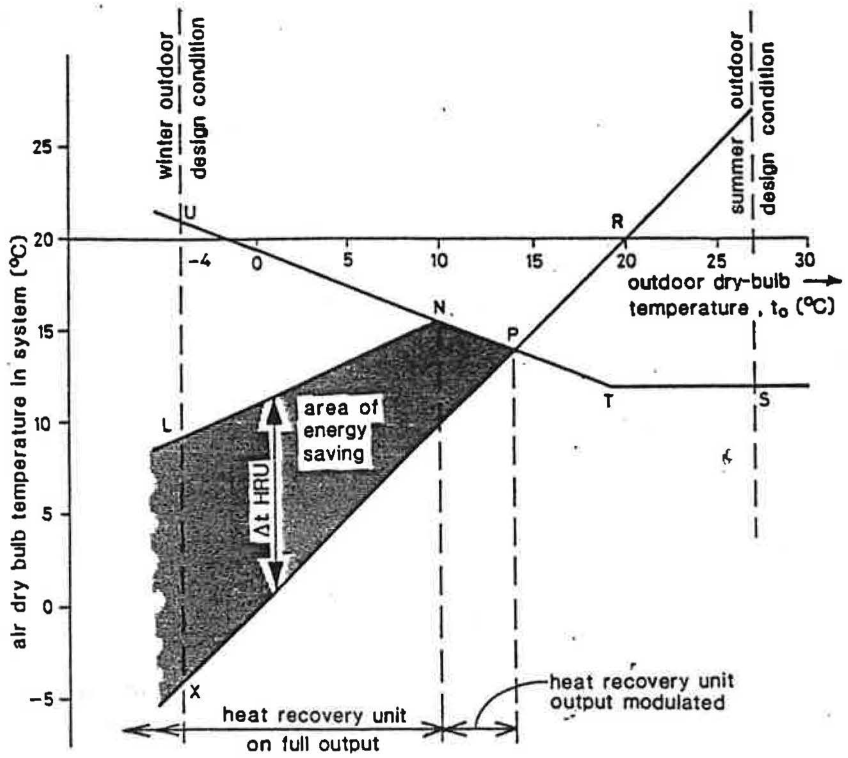

Fig. 15 Temperature analysis diagram for system 3