ENGINEERING TRIPOS PART IIB

Wednesday 5 May 2010 9 to 10.30

Module 4D11

BUILDING PHYSICS

*Answer not more than **three** questions.*

All questions carry the same number of marks.

*The **approximate** percentage of marks allocated to each part of a question is indicated in the right margin.*

Attachments: Building Physics Environmental Data Sheets (10 pages)

STATIONERY REQUIREMENTS	SPECIAL REQUIREMENTS
Single-sided script paper	Engineering Data Book
	CUED approved calculator allowed

You may not start to read the questions printed on the subsequent pages of this question paper until instructed that you may do so by the Invigilator

1 (a) A cavity wall consists of four layers. The outer layer is 102 mm thick brickwork. Proceeding inwards, there is a 75 mm cavity filled with mineral wool, 100 mm of concrete blockwork then finally, on the internal face, 15 mm of dense plaster. The thermal conductivities of brick, wool, concrete block and plaster are 0.84 W/mK, 0.04 W/mK, 0.20 W/mK and 0.56 W/mK respectively. Assuming external and internal surface resistances of 0.06 m^2 K/W and 0.12 m^2 K/W respectively, determine the U-value of the wall. [30%]

(b) The fabric conductance $\sum UA$ of a single-roomed building has contributions of 600 W/K from all vertical surfaces (walls, windows and doors), 300 W/K from the roof and 100 W/K from the floor. The total ventilation conductance is 150 W/K. It is desired to maintain a steady internal air temperature of 20 °C when the external temperature is –2 °C.

(i) Estimate the heat input rate required, using the simplest possible model. [20%]

(ii) Assuming the heat input has a radiative fraction of zero, and assuming a total conductance of 1800 W/K between the air and environmental nodes, refine the estimate for the heat input calculated in (i) using the CIBSE Simple Model. Estimate the temperature at the environmental node. [30%]

(c) Explain briefly how an evaporative cooler may, in some circumstances, be used to improve the thermal comfort in a room. [20%]

2 A building has a load demand of 1000 Wh/day in direct current (12 V) and 6500 Wh/day in alternate current (240 V). A south facing photovoltaic (PV) array consisting of 200 W panels is required to meet this demand.

(a) With the aid of simple sketches describe how PV cells convert solar energy to electricity. [30%]

(b) By using the solar irradiation chart provided in the data sheets, determine the optimum inclination and the number of 200 W panels required for a grid connected PV system and for an alternative off-grid PV system. [30%]

(c) Given that the capital cost of a 200 W photovoltaic panel is £1000, the service life is 25 years and the discount rate is 5%, use the discounted cash flow method to calculate the cost of electricity generated with a grid-connected PV system. In your calculation you may assume that the cost of electricity supplied by the grid is £0.15 / kWh and the tariff for electricity supplied to the grid is £0.10 / kWh. [40%]

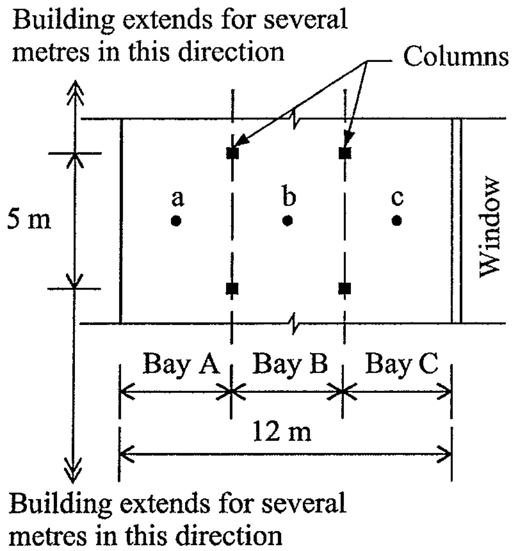
3 An open plan office located on a very long and narrow site has a lighting requirement of 500 lux. The office is 12 m wide and is split into three equal bays A, B and C as shown in Fig. 1. The office is daylit from a window along its longer side that overlooks open country views. The window extends the full length of the building and has a sill height of 0.8 m, a lintol height of 2.5 m and a transparency of 70%. The average internal reflectance of the office is 30%.

(a) Artificial lighting is provided by 2.6 m long fluorescent tubes rated at 80 W with an efficiency of 98 lm/W. The tubes are mounted in luminaries giving a utilisation factor of 75%. By ignoring the contribution from natural light, determine the number of luminaries required and suggest a suitable layout. [20%]

(b) Discuss briefly the advantages and disadvantages in maximising daylight levels in office buildings. [20%]

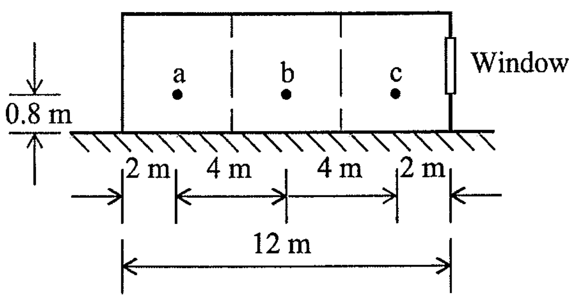
(c) By determining the total daylight factor at three points a, b and c shown in Fig. 2, use the data on daylight availability provided in the data sheets to calculate the fraction of standard annual office hours for which artificial lighting will be required in each of the three bays. [40%]

(d) Use your answers to part (c) to briefly describe a suitable switching option for exploiting daylight. Calculate the resulting electrical energy savings per unit area of office floor that would result from implementing this switching option. [20%]



Part Plan

Fig. 1



Typical Section

Fig. 2

4 A school gymnasium is to be converted into a basic theatre by the introduction of removable tiered seating, a stage, lighting and other technical facilities. Figure 3 shows a simplified section through the building. The seating when extended is represented by the triangle and the stage by the rectangle; both occupy the full 16 m width of the gymnasium. The measured reverberation time of the unoccupied gymnasium has been found to be 3.4 s at mid-frequencies and the average absorption coefficient of the room surfaces is negligible. The seating has a mid-frequency absorption coefficient of 0.7, but offers no significant absorption when retracted. The proposed absorbing material has a mid-frequency absorption coefficient of 0.8. No significant absorbent areas are revealed when the seating is retracted and the stage is acoustically reflecting.

(a) What is the recommended mid-frequency reverberation time for a theatre? [15%]

(b) Calculate the area of absorbing material required to achieve this reverberation with the tiered seating deployed and a further 70 m^2 of seating added to the flat floor? [30%]

(c) The seating retracts into a volume of 120 m^3. What would the reverberation time of the room be in this configuration if the additional absorption had been installed and the additional seating removed? [20%]

(d) Describe briefly where in the room the additional absorption should be placed to optimise the acoustic performance of the theatre? [25%]

(e) If no additional absorption is revealed when the seating is retracted, what acoustic fault would be created in this configuration? [10%]

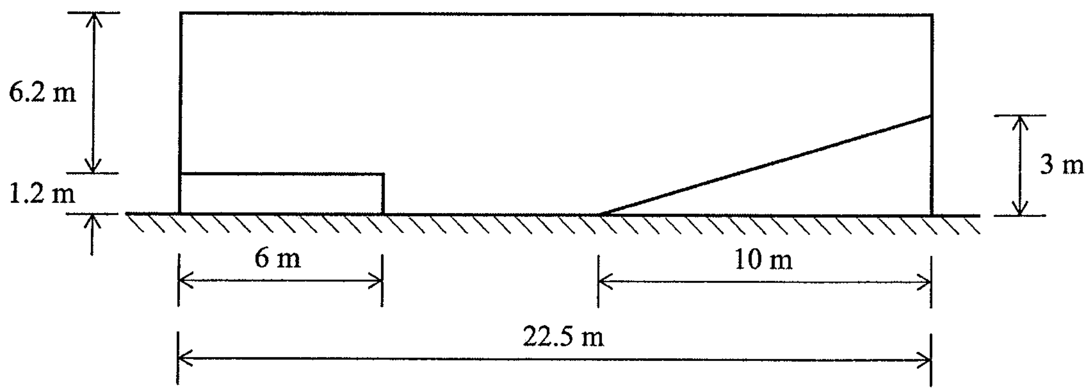


Fig. 3

mo03

END OF PAPER

1. Lighting & PV Systems

(a) <u>Definitions</u> — units

Luminous flux – rate of flow of light energy – lumens (1m)

Illuminance – density of light flux reaching a surface – lumens/m^2 or lux (1x)

Luminous intensity – light flux per unit solid angle from a point source, i.e. power to emit in a particular direction – candela (cd) (1 cd $\equiv 4\pi$ 1m)

(b) <u>Artificial light</u>

Recommended illuminances, on horizontal working plane, vary from 150 lux for storage areas, through 500 lux for general offices, to 1500 lux for precision bench work.

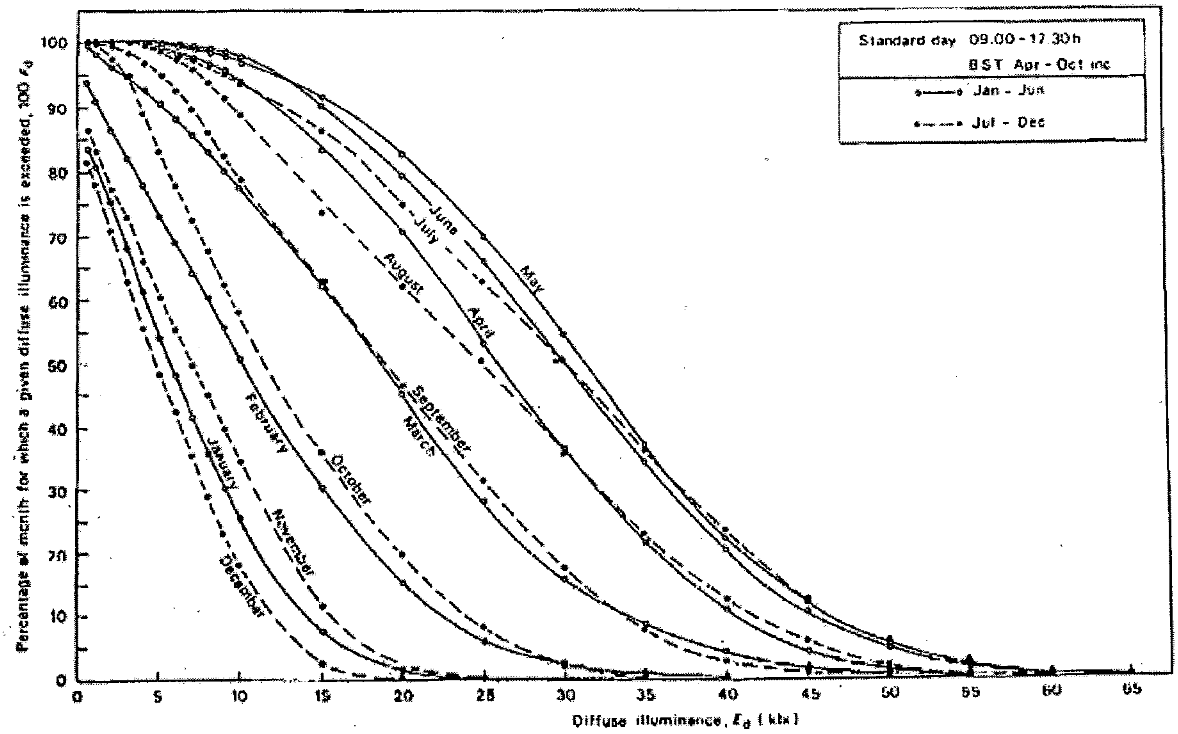
"Utilisation factor" is the proportion of light emitted by the luminaires which actually reaches the working plane.

Typical luminous efficacies (lumens/Watt): tungsten filament (GLS) 12, tubular florescent 60, low pressure sodium 180, daylight 115.

(c) <u>Daylight</u>

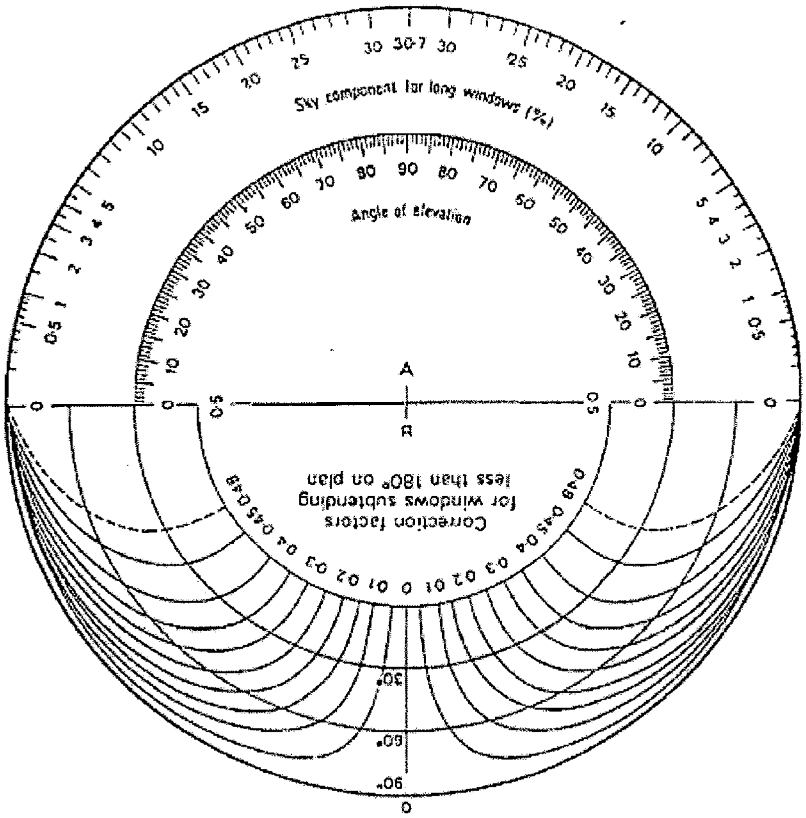
Sky as a diffuse source: sky luminance B_θ from elevation θ above horizon, where B_Z is luminance at the zenith: $B_\theta \approx B_Z(1+2\sin\theta)/3$ "CIE sky"

Example of monthly daylight availability curves for Bracknell; Percentage of working time exceeded vs diffuse illuminance level in *klx*.

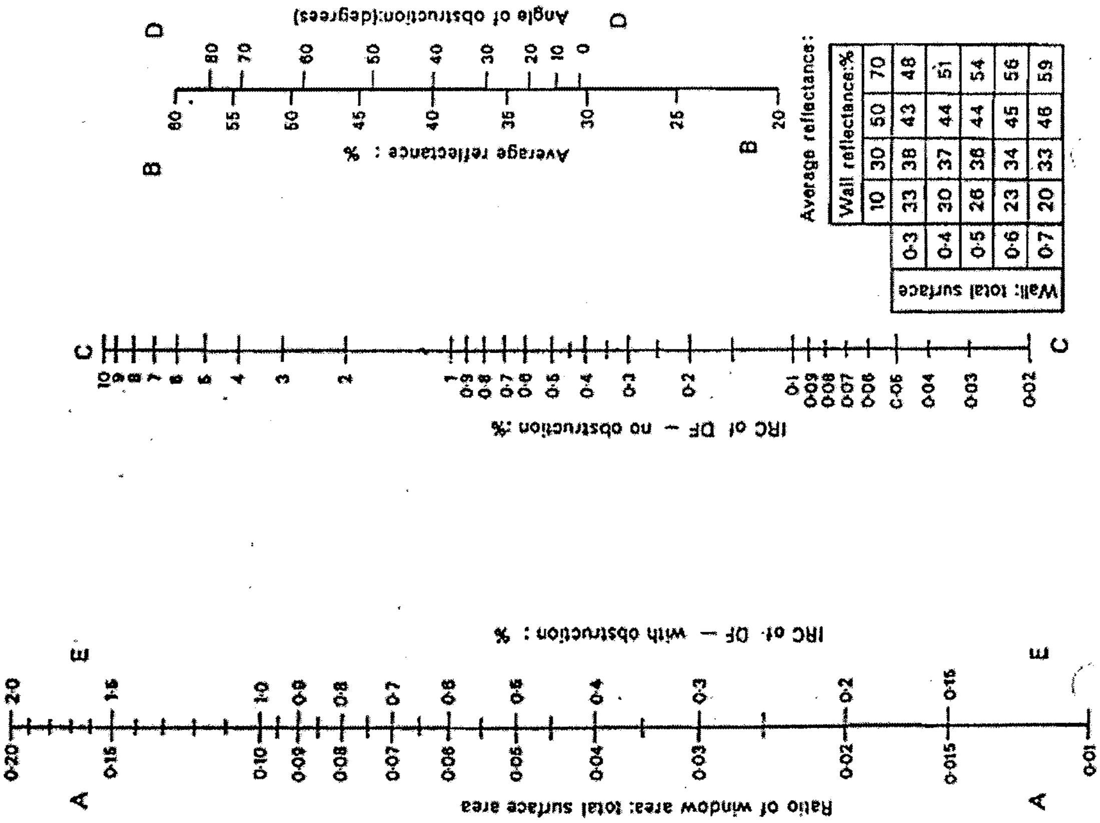


Average daylight factor (%) in a room is approx. TWMd/A(1 – ρ^2) where T is glazing transmittance; W is net window area; M is 'maintenance factor' (i.e. cleaning); d (degrees) is the angle at the window centre in the vertical plane between the vertical and the highest external building obstruction; A is the total area of all internal surfaces; and ρ is the weighted mean reflectance of the internal surfaces.

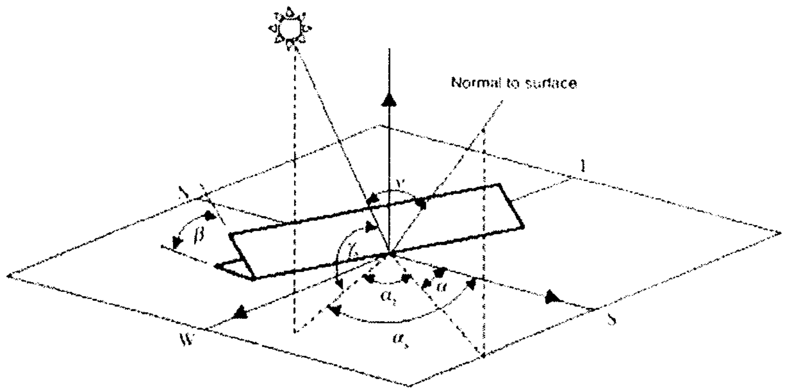
BRE Sky Component protractor: single clear vertical glazing, CIE overcast sky, illuminance on a horizontal surface indoors. Externally reflected component is SC with a further correction factor of 0.2.



Typical nomogram for internally-reflected component:
(ground reflectance 0.1 in this case)



(d) Solar irradiation



Typical mean daily irradiation on South facing panel in Southern Europe (kWh/m^2)

Angle	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Annual
0	2.25	2.92	3.88	4.98	5.66	6.38	6.67	5.75	4.20	3.18	2.23	1.70	4.15
5	2.56	3.19	4.08	5.09	5.69	6.36	6.68	5.84	4.46	3.42	2.49	1.93	4.32
10	2.86	3.44	4.26	5.18	5.70	6.32	6.66	5.90	4.61	3.64	2.74	2.16	4.46
15	3.14	3.67	4.42	5.25	5.67	6.25	6.61	5.94	4.73	3.84	2.98	2.37	4.57
20	3.40	3.87	4.55	5.28	5.62	6.15	6.52	5.94	4.82	4.03	3.19	2.57	4.66
25	3.65	4.05	4.66	5.29	5.54	6.03	6.41	5.91	4.89	4.18	3.39	2.75	4.73
30	3.86	4.21	4.73	5.26	5.44	5.88	6.26	5.85	4.93	4.31	3.57	2.92	4.77
35	4.05	4.34	4.78	5.21	5.31	5.70	6.08	5.75	4.94	4.42	3.72	3.07	4.78
40	4.22	4.45	4.81	5.13	5.15	5.49	5.88	5.63	4.93	4.50	3.85	3.19	4.77
45	4.36	4.53	4.80	5.03	4.97	5.26	5.65	5.48	4.88	4.55	3.96	3.30	4.73
50	4.47	4.58	4.77	4.89	4.77	5.01	5.39	5.29	4.81	4.57	4.04	3.39	4.66
55	4.55	4.60	4.71	4.73	4.55	4.74	5.11	5.09	4.71	4.57	4.09	3.45	4.57
60	4.60	4.59	4.62	4.55	4.30	4.45	4.80	4.85	4.58	4.53	4.12	3.49	4.46
65	4.62	4.55	4.50	4.34	4.04	4.14	4.48	4.59	4.42	4.47	4.12	3.51	4.32
70	4.61	4.49	4.36	4.11	3.77	3.83	4.15	4.31	4.25	4.38	4.10	3.51	4.15
75	4.57	4.39	4.19	3.86	3.48	3.50	3.80	4.02	4.05	4.27	4.04	3.48	3.97
80	4.50	4.27	4.00	3.59	3.18	3.17	3.44	3.70	3.82	4.13	3.97	3.43	3.77
85	4.40	4.13	3.79	3.31	2.88	2.84	3.08	3.37	3.58	3.96	3.87	3.36	3.55
90	4.27	3.95	3.55	3.02	2.57	2.51	1.86	3.04	3.32	3.78	3.74	3.27	3.24

(d) Photovoltaic Systems

Nominal power of PV panels (W): $P_0 = \dfrac{L}{PSH}$

Where L = load (Wh); PSH = peak solar hours (h)

Battery Capacity (Ah): $Q = \dfrac{L\ C_s}{V\ DOD_{\max}}$

Where L = load (Wh); C_s = days of autonomy; V = voltage; DOD_{max} = maximum draw down (%).

2. Thermal matters

(a) Temperatures

Air temperature in shade T_a – the 'dry bulb' temperature, usually in degrees C db.

'Wet bulb' temperature, C wb, in a small damp sponge in air current (taken with T_a indicates humidity).

'Mean radiant temperature' T_r – the uniform surface temperature of a surrounding black enclosure delivering the same radiant heat to the point in question as arrives in the actual non-uniform space (in practical rooms, approx. the mean surface temperature T_m of all the enclosing surfaces).

Radiant heat flow is roughly $Q_r = A\,\varepsilon\,\sigma\,(T_b^4 - T_r^4)$ where A and T_b (K) are the surface area and temperature of the radiating body, ε is emissivity (usually 0.95, but 1.0 for a 'black' body), and σ is 5.67×10^{-8} W/m^2K^4.

'Globe temperature' (measured inside a small black sphere) is roughly the 'operational temperature', the mean of T_a and T_r.

'Environmental temperature' T_e in a room is $(T_a + 2T_m)/3$.

'Corrected effective temperature' CET depends on globe and wet-bulb temperatures, and air velocity – the wind-chill effect – and is obtained from charts.

'Neutral temperature' (CET in C at which most people feel comfortable) is
$T_n = 11.9 + 0.564\ T_o \pm 2.5$ (Humphreys) for a sedentary occupation, where T_o is the mean outdoor temperature for the month in question.

(b) Thermal Comfort

Comfort will depend on many factors, not just the temperature but also such things as the humidity, the freshness of the air, and the amount of clothing being worn (0.1 clo for shorts only, 1.0 clo for a business suit, 2.5 clo for a heavy overcoat).

Also important will be the metabolic rate M (ranging from 70 W when sleeping, through 150 W when typing, 300 W for fast walking, to 650 W for hard sustained work); the rate W watts at which work is being done; and the rate H watts of loss of heat, which will depend on radiation, convection, and evaporation from the skin, as well as heat and water-vapour losses in breathing.

Fanger introduced the Predicted Mean Vote (PMV) for people's sensation of comfort on a scale of – 3 to + 3 (very cold to very hot). His equation has over 15 terms, based on metabolic rate, work being done, temperature etc, with empirical constants and factors based on surveys of large numbers of people – and with some subsequent dispute whether the equation is correct in all circumstances.

Fanger also investigated the 'Predicted Percentage Dissatisfied' at a given PMV, suggesting the relation $PPD = 100 - 95\ \exp\{-(0.04\ PMV^4 + 0.22\ PMV^2)\}$.

(c) Heat loss and gain calculations for buildings

Total 'specific heat loss' $Q = Q_c + Q_v$ in watts per degree difference between environmental temperature inside and air temperature outside.

By conduction $Q_c = \Sigma U A$ W/K , where A is area of wall, roof, windows etc, each with their individual 'U-value'. For layered construction, the U-value in W/m^{2}K is given by

$$1/U = R_i + \Sigma r t + R_c + R_e$$

where R_i and R_e are thermal resistances at internal and external surfaces (depending on radiative and convective heat transfer), R_c is for any cavity, and r and t are respectively reciprocal of conductivity, and thickness, of the various layer materials (typical conductivities being given in tables of data).

By ventilation Q_v depends on room volume, number of air changes per unit time, and the appropriate coefficient for air (ρc_p, approx 1200 J/m^{3}K at 20 C).

'Degree days', presented in statistics for each month for a locality, are the integral over time of the shortfall of the outside air temperature below a chosen internal design temperature, often 18 C.

'Mean internal environmental temperature' , over a long period say 24 hours, can be calculated, as an increment above the mean outside air temperature, from the mean internal casual heat gains (people, lights, computers etc) plus the mean solar gains (window area, gain in W/m^2 depending on aspect and time of year, and a Solar Gain Factor) – giving total mean heat gains (in W) – and the specific heat loss Q in W/K .

Typical solar gains in June for a South-facing window are 700 W/m^2 .

Swings in internal environmental temperature (mean to peak) can be estimated, for the time of day when the peak is likely to occur, from;

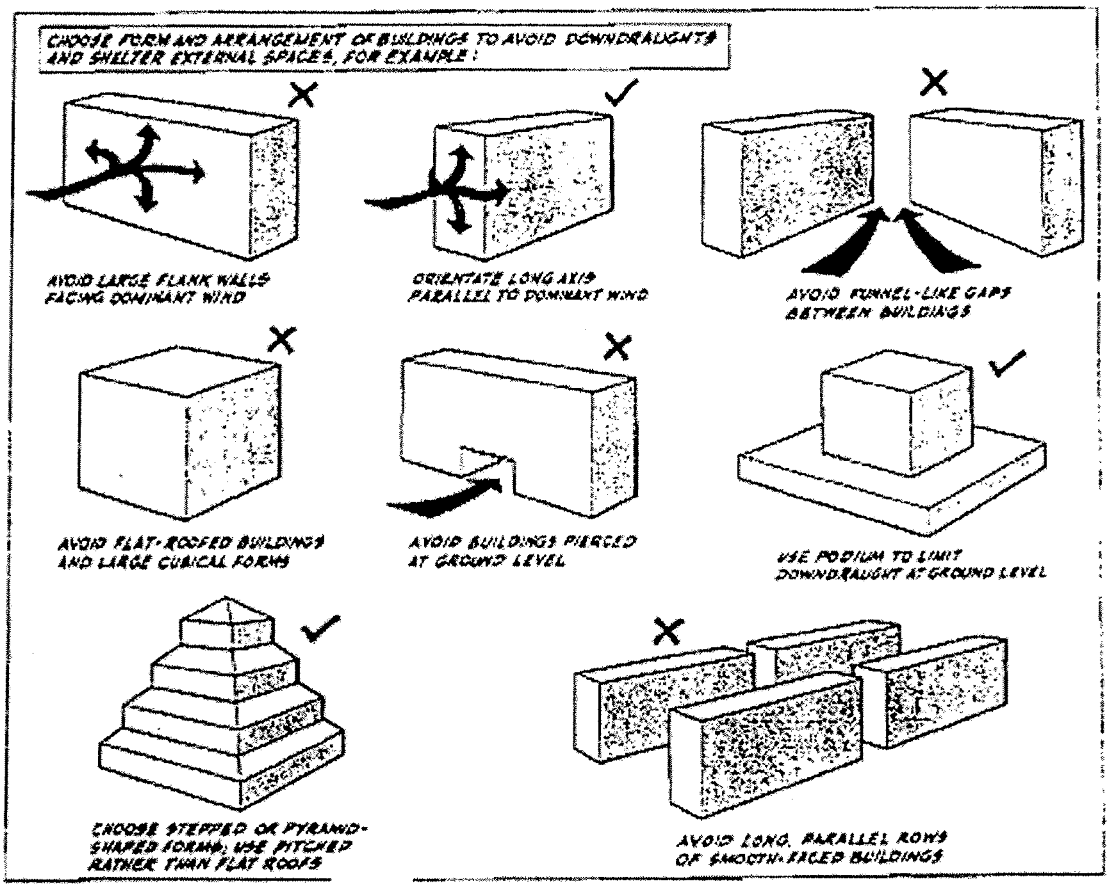
(i) the swing in solar gain (W) using the window area, the difference between peak and mean gains (W/m^2) and an Alternating Solar Gain Factor;
(ii) any simultaneous swing in casual gain (W) from its mean; and
(iii) any departure of the outdoor air temperature at the peak time from its mean, multiplied by a new 'specific heat loss factor' Q (now from window area and U-value for glazing, plus ventilation)

to give a total apparent swing of heat gain in W.

This total is divided by another specific heat loss factor – now ventilation Q_v plus the sum of wall areas times Y-values ('admittances' in W/m^{2}K) – to give the swing in environmental temperature from the mean. Y-values for the various materials are available from tables of data.

3. Ventilation

(a) Reducing the wind sensitivity of buildings (from Building Research Establishment, Digest 350)



Wind pressure coefficients (tabulated) ; $C_p = \dfrac{p - p_{ref}}{\frac{1}{2}\rho U_{ref}^2}$;

where $U_{ref} = U_{ambient}$ is typically taken as wind velocity 10 m above ground level.

(b) Orifice flow

$$\Delta p = K \frac{1}{2} \rho U^2 \text{ , with } K \approx 1$$

whence $$U = \sqrt{\frac{2\Delta p}{\rho}}$$

Q (volume flow rate) = $C_D \times U \times Area$
C_D = discharge coefficient due to streamline contraction.

(c) Momentum jets

$$R \quad \alpha \quad x$$

$$U \alpha \left(\frac{M_o}{\rho}\right)^{\frac{1}{2}} x^{-1} \text{ ; } M_o = source\ momentum\ flux = \rho\, Q_o U_o$$

(d) Buoyancy effects

Stack effect; $U \approx \sqrt{g \frac{\Delta\rho}{\rho} H}$ for two equal area vents, depending on discharge C_D etc.

Empirical equations used in practice:

nominal pressure difference $\Delta p = 0.043\ h\ (T_i - T_o)$ Pa , where h (m) is the height between inlet and outlet of the stack and T_i and T_o are average internal and air outside temperatures.

then volume flow is $Q = 0.827\ A\ (\Delta p)^{1/2}\ m^3/s$, where A (m^2) is given for inlet area A_1 and outlet area A_2 by $A = A_1 A_2/(A_1^2 + A_2^2)^{1/2}$.

Exchange flows Q (one fluid) $= \begin{pmatrix} 0.25\ horizontal \\ 0.05\ vertical \end{pmatrix} \sqrt{g \frac{\Delta\rho}{\rho} d}$ (Area)

Gravity currents $U = (\sim 1.0) \sqrt{g \frac{\Delta\rho}{\rho} h}$

Buoyant plumes

$$R = 0.12\, z$$

$$U = 2.55\ F_o^{\frac{1}{3}}\ z^{-\frac{1}{3}}$$

$$g\left(\frac{\Delta\rho}{\rho}\right) = 8.66\ F_o^{\frac{2}{3}}\ z^{-\frac{5}{3}}$$

$$F_o = Q_o g \frac{\Delta\rho_o}{\rho} = \frac{\dot{Q}_g}{\rho T C_p}$$

Consistent with assumptions in derivation we have $\rho \approx \rho_a$ and use ρ_a and T_a as reference conditions whenever necessary.

4. Acoustics

(a) Fundamentals and definitions

Velocity of sound in air at 20 °C : $c \approx 344 m/s$ when air density $\rho \approx 1.2 kg/m^3$

Consider root mean square pressure fluctuation $\bar{p}$ Pa and standard reference level $p_0 = 2.0 \times 10^{-5}$ Pa at 1000Hz. Sound pressure level (SPL) defined as $20 \log_{10} (\bar{p}/p_0)$ decibels.
Sound intensity (rate of energy transmission across given surface):

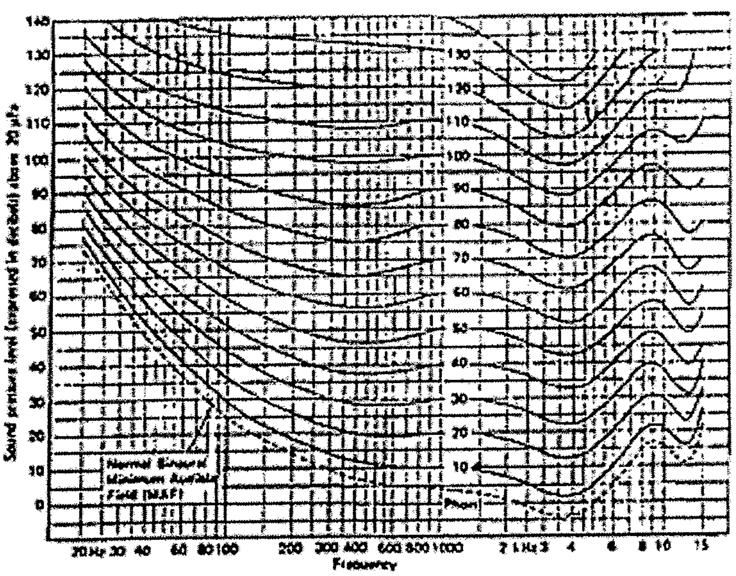
$$I = \frac{\bar{p}^2}{\rho c}: \quad IL = SPL = 10 \log (I/I_0): \quad I_0 = \frac{p_0^2}{\rho c} \approx 10^{-12}$$

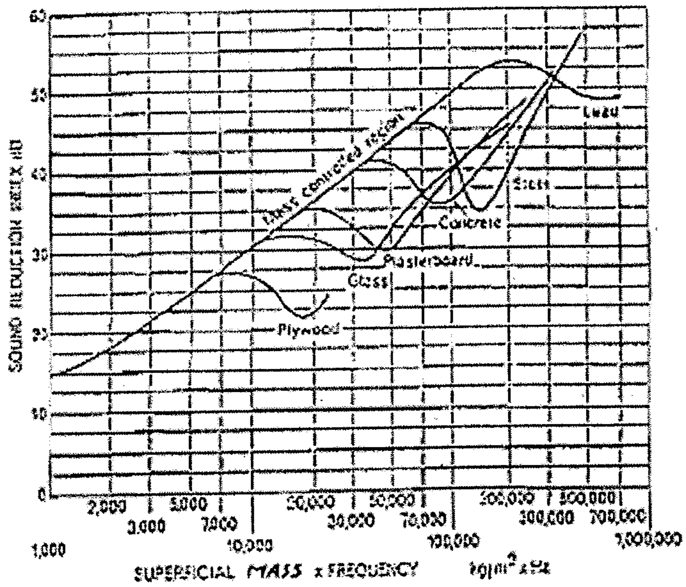
Sound energy density (energy per unit volume):

$$D = \frac{\bar{p}^2}{\rho c^2}: \quad SPL = 10 \log (D/D_0): \quad D_0 = \frac{p_0^2}{\rho c^2}$$

Source power W watts : SWL = 10 log (W/10^{-12})

Human ear: curves of equal perceived loudness (men) on left below: sound pressure level in dB versus frequency in Hz.





So adjusted curves (e.g. dBA from 40 phon line) to allow for ear characteristics varying with frequency.

Addition and comparison of incoherent sound: add mean square pressures to find overall mean square and hence SPL (or L_{eq} using average over time for varying sound levels).

(b) Acoustics of room-like enclosures (volume V, total surface S)

From uniform omnidirectional source W, at radius r (m) :
direct SPL = SWL – 10 log 4π - 20 log r

(from practical sources, intensity varies with direction). Reverberant sound due to reflections from walls: energy density D tends to 4W/Ac where A is the total absorption of the enclosure surface i.e. total of areas times absorption coefficients α , or $A = S\overline{\alpha}$ where $\overline{\alpha}$ is the mean absorption coefficient. Intensity in enclosure is Dc/4.

So reverberant SPL = SWL + 10 log 4 – 10 log A.

"Room radius" is the distance from the source at which direct and reverberant sound levels are equal.

On switching off source, energy density D decays exponentially, with time constant 4V/Ac related to the "reverberation time" T (Sabine's Law). Eyring's modification: use $A = -S\ln(1-\overline{a})$. Preferred values for T: speech 1.0 secs, orchestral music 1.8 to 2.2 secs.

(c) Noise control

Barriers and screens: if uninterrupted wave travels distance d and diffracted wave $a+b$, wavelength λ, "insertion loss" on introducing an infinitely long barrier across the path is 10 log (3 + 20N) dB where $N = 2(a+b-d)/\lambda$.

Partitions: 'transmission coefficient' τ = transmitted/ incident intensity, and 'sound reduction index' $R = 10\log(1/\tau)$. For source and receiver rooms separated by area S of partition,

difference in SPL's = R – 10 log (S / A) dB

where A is absorption in receiving room. For compound partitions, use transmission coefficient τ weighted by areas.

Mass law: plane wave incident at θ to normal of single leaf wall

$$R = 10\log\left[1+\left(\frac{\pi Mf\cos\theta}{\rho c}\right)^2\right]$$

where f is sound frequency and M is wall mass per unit area. For high frequencies and diffuse sound field $R = 20\log(Mf) - 47$ dB.

Problems with mass Law: resonance effect at frequencies proportional to $\sqrt{B/M}$ and dependent on panel span, where B is bending stiffness per unit width; "coincidence effect", when speed of bending waves in panel equals speed of sound in air, at frequency proportional to $\sqrt{M/B}$ and independent of panel span. See curves of R for different materials, on right above, storing R in dB versus superficial mass x frequency in kgHz/m^2.

Double wall mass-air-mass resonance: frequency f_d

$$f_d = \frac{1}{2\pi}\sqrt{\frac{\gamma P_o}{d}\cdot\frac{(M_1+M_2)}{M_1 M_2}}$$

where γ is 1.4 for air, P_o is atmospheric pressure, d is cavity width, and M is wall mass per unit area.

5. Whole-life costing

(a) Discounted cash flow table

Capital repayment period/years[1]	Real discount rate /%						
	0	2	5	8	10	12	15
5	200	212	231	250	264	277	298
10	100	111	130	149	163	177	199
15	67	78	96	117	131	147	171
20	50	61	80	102	117	134	160
25	40	51	71	94	110	127	155
30	33	45	65	89	106	124	152
40	25	37	58	84	102	121	151
50	20	32	55	82	101	120	150
60	17	29	53	81	100	120	150

1 This is not necessarily equal to the total physical lifetime of the project.