Outdoor and indoor air design conditions

The amount of heating and cooling loads that have to be accomplished to keep buildings comfortable in cold winter and hot summer depend on the desired condition indoors and on the outdoor conditions on a given day. These conditions are, respectively, termed the “indoor design condition” and the “outdoor design condition”.
In principle, the heating and cooling loads are calculated to maintain the indoor design conditions when the outdoor weather data do not exceed the design values.

Outdoor (outside) Design Conditions

It is not economical to choose either the annual maximum or annual minimum values of the outdoor weather data in determining the outdoor conditions. The outdoor design data is usually determined as follows,

1- According to the statistical analysis of the weather data (meteorological data) so that 1, 2.5 or 5% of the total possible operating hours is equaled or exceeded the outdoor design values.

2- According to ASHRAE (The American Society of Heating, Refrigeration and Air Conditioning Engineers) for some capitals in the world.

Summer Design Condition

The recommended summer design and coincident wet bulb temperature, when chosen as being equaled to or exceeded by 2.5% of the total number of hours (i.e. 2928 hours) in June, July, August and September, are

(i) 23 ^{o}C dry bulb temperature, and

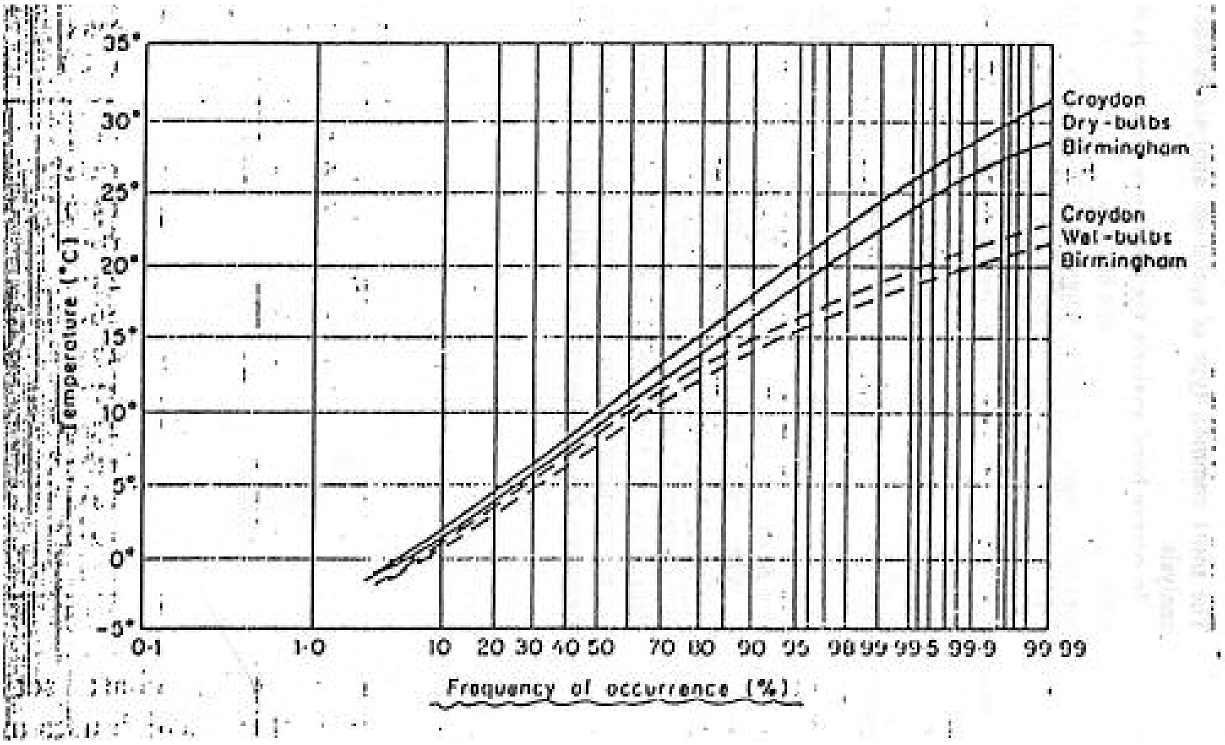
(ii) 28 ^{o}C wet bulb temperature

Figure below shows the outdoor dry bulb temperature and wet bulb temperature curves for a typically hot summer day in Hong Kong. Usually the maximum temperature of 33 ^{o}C occurs at 2 p.m. and the minimum temperature of 28 ^{o}C occurs just before sunrise. The daily range of dry bulb temperature is about 5 to 6 ^{o}C, and the daily mean dry bulb temperature is 30.5 ^{o}C.

Note:

Outdoor (outside) Design Conditions depend on the following:

- Geographical location.
- Latitude.
- Elevation.
- Season (summer or winter).



Winter design condition

The recommended winter design and coincident relative humidity, when chosen as being equaled to or exceeded by 1% or 2.5% of the total number of hours (i.e. 2160 hours) in December, January and February, are

(i) 9 ^{o}C dry bulb temperature, and

(ii) 50% relative humidity

Minimum temperature occurs at 6 a.m. or 7 a.m. before sunrise and the daily range is about 6 to 8 ^{o}C during very cold winter days.

Indoor design conditions:

For most of the comfort air-conditioning systems used in the commercial and public buildings, the recommended indoor temperature and relative humidity according to ASHRAE (The American Society of Heating, Refrigeration and Air Conditioning Engineers) are as follows:

(i) Summer: 23.5 - 25.5 oC dry bulb temperature, 50 - 60 % relative humidity

(ii) Winter: 21 - 23.5 oC dry bulb temperature, 30 - 40 % relative humidity.

Note:

1- The indoor design conditions are determined according to ASHRAE (The American Society of Heating, Refrigeration and Air Conditioning Engineers).

2- Outdoor and indoor design conditions are used to calculate design space load.

Cooling load calculation

The purpose of the load calculation (estimation):

The purpose of a load estimation is determine the size of the air conditioning and refrigeration equipment that is required to maintain inside design conditions during periods of maximum outside temperatures. The design load is based on inside and outside design conditions and it is air conditioning and refrigeration equipment capacity to produce and maintain satisfactory inside conditions.

In cooling load calculation, there are four related heat flow terms;

1- Space cooling load
2- Space heat gain
3- Space heat extraction rate.
4- Cooling coil load.

Space cooling load:

Definition of the cooling load:

The total heat required to be removed from the space in order to bring it at the desired temperature by the air conditioning and refrigeration equipment is known as cooling load.

Classification of the cooling loads:

Cooling loads are classified as follows,

1- Classification of the cooling load according to the kinds of heat for a hot weather:

The two main components of a cooling load imposed on an air conditioning plant operating during the hot weather are follows,

i- Sensible heat gain:

The sensible heat gain is defined as the direct addition of heat to the enclosed space, without any change in its specific humidity.
This sensible heat most be removed during the process of summer air conditioning. The sensible heat gain may occur due to any one or all of the following sources of the heat transfer,

a- The heat flowing into the building by conduction through exterior walls, floors, ceiling, doors and windows due to the temperature difference on their two sides.
b- The heat received from the solar radiation. It consists of the heat transmitted directly through glass of windows, ventilators or doors, and the heat absorbed by walls and roofs exposed to solar radiation and later on transferred to the room by conduction.
c- The heat conducted through interior partition from rooms in the same building which are not conditioned.
d- The heat given of by lights, motors, machinery, cooking operations, industrial processes …etc.
e- The heat liberated by the occupants.
f- Sensible heat gain of ventilation due to temperature difference between the fresh air and the air in space (outside and inside air)
g- The heat carried by the outside air which leaks in (infiltration) through the cracks in doors, windows and through their frequent openings.
h- The heat gain through the walls of ducts carrying conditioned air through unconditioned space in the building.
i- The heat gain from the fan work.

ii- Latent heat gain:

The latent heat gain is defined as the heat gain of space through addition of moisture, without change in its dry bulb temperature.
This latent heat is to be removed during the process of summer air-conditioning. The latent heat gain may occur due to any one of all the following sources,

a- Latent heat gain of ventilation due to difference of humidity between the fresh air and the air in space (outside and inside air).
b- The heat gain due to moisture in the outside air entering by infiltration.
c- The heat gain due to condensation of moisture from occupants.
d- The heat gain due to condensation of moisture from any process such as cooking foods which takes place within the conditioned space.
e- The heat gain due to moisture passing directly into the conditioned space through the permeable walls or partitions from the outside or from adjoining regions where the water vapor pressure is higher.

Note 1:

a- The total heat load to be removed by the air conditioning and refrigeration equipment is the sum of the sensible and latent heat loads as discussed above.
b- The sensible het load refers to the dry bulb temperature of the building and the latent heat load refers to the wet bulb temperature of the building.

2- Classification of the cooling load according to inside-outside environment:

There are two types,

i- External Loads:
External cooling loads consist of the following:

a- Sensible loads (conductive heat gain) through opaque envelope assemblies (surfaces) such as roofs, walls, floors, doors, windows, partitions,…etc
b- Sensible loads (solar heat gain) through transparent or translucent envelope assemblies (surfaces) such as skylights, windows, glazed openings,…etc
c- Sensible loads through ventilation and infiltration (air leakage)
d- Latent loads through ventilation and infiltration.

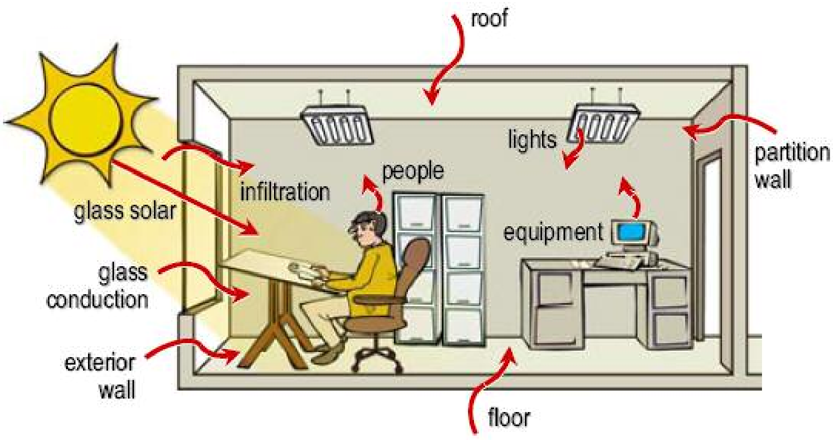
i- Internal Loads:
Internal cooling loads consist of the following:

a- Sensible & latent heat due to people
b- Sensible heat due to lighting
c- Sensible heat due to power loads and motors (elevators, pumps, fans & other machinery)
d- Sensible & latent heat due to appliances

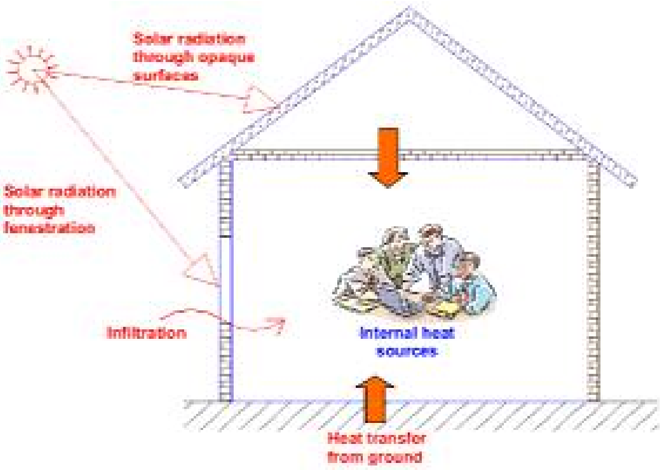
Space heat gain:

Space heat gain is defined as the rate at which heat enters a space, or heat generated within a space during a time interval.

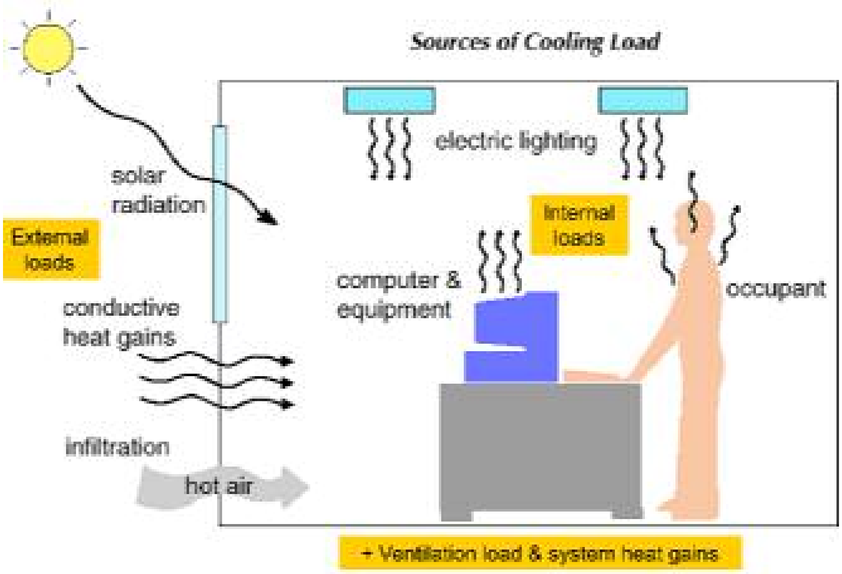
Note:
The total of external and internal heat gain which discussed above is known as room heat giants. Figure below describes the Components of external and internal loads.



a- Components of building cooling load

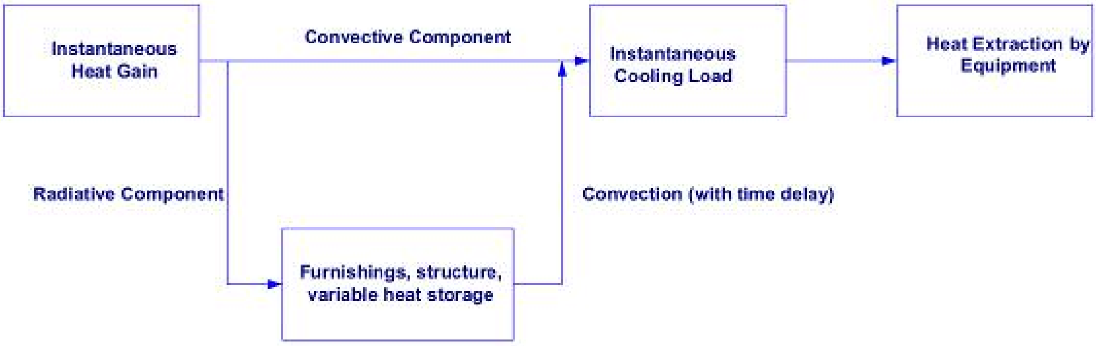


b- Components of building cooling load



c- Components of building cooling load
Components of external and internal loads

Figure below shows the schematic relation (difference) of heat gain to cooling load.



Schematic relation (difference) of heat gain to cooling load

Space heat extraction rate:

The rate at which heat is removed from the conditioned space equals the space cooling load only to the degree that room air temperature is held constant.

Note:

Theoretically, it may seem logical to address that the space heat gain is equivalent to space cooling load but in practice "heat gain ≠ cooling load."

Cooling coil load:

The cooling coil load is the summation of all the cooling loads of the various spaces served by the equipment plus any loads external to the spaces such as duct heat gain, duct leakage, fan heat, and outdoor makeup air.

Initial design considerations:

Generally the steps should be followed before the cooling load calculation.

1) Geographical site conditions (latitude, longitude, wind velocity, precipitation etc.).

2) Outdoor design conditions (temperature, humidity etc).

3) Indoor design conditions.

4) Building characteristics (materials, size, shape, surface colors are usually determine from building plan and specifications).

5) Configuration (location, orientation and external shading from building plan and specifications, also the shading from adjacent building can be determined by a site plan or by visiting the proposed site such as from adjacent water, sand, parking,…etc).

6) Operating schedules (lighting, occupancy, appliances, and equipment).

7) Additional considerations (type of air-conditioning system, fan energy, fan location, duct heat loss and gain, duct leakage, type and position of air return system…).

Cooling load calculation methods:

For a thorough calculation of the zones and whole- building loads, o ne of the following three method s should be employed:

a. Transfer Function Method (TFM): This is the most complex of the methods proposed by ASHRAE and requires the use of a computer program or advanced spread sheet.

b. Cooling Load Temperature Differential/Cooling Load Factors (CLTD/CLF): This method is derived from the TFM method and uses tabulated data to simplify the calculation process. The method can be fairly easily transferred into simple spread sheet programs but has some limitations due to the use of tabulated data.

c. Total Equivalent Temperature Differential/Time-Ave raging (TETD/TA): This was the preferred method for hand or simple spread sheet calculation before the introduction of the CLTD /CL F method.

d. Heat Balance (HB) & Radiant Time Series (RTS)

These three methods are well documented in ASHRAE Hand book Fundamentals, 200 1.

Note:

i- Transfer Function Method (TFM is widely used in HVAC & R.
ii- Heat Balance (HB) & Radiant Time Series (RTS) is generally used for research and analytical purposes and can be solved by differential equations using numerical solutions (Finite different method).

Cooling load temperature difference (CLTD) and cooling load factor (CLF):

Cooling load temperature difference and cooling load factor are used to convert the space sensible heat gain to space sensible cooling load.

Cooling load temperature difference:

Cooling load temperature difference is defined as an equivalent temperature difference used for calculating the instantaneous external cooling load across a wall or roof.

The space sensible cooling load Q_{rs} is calculated as:

$$Q_{rs} = A \cdot U \cdot CLTD$$

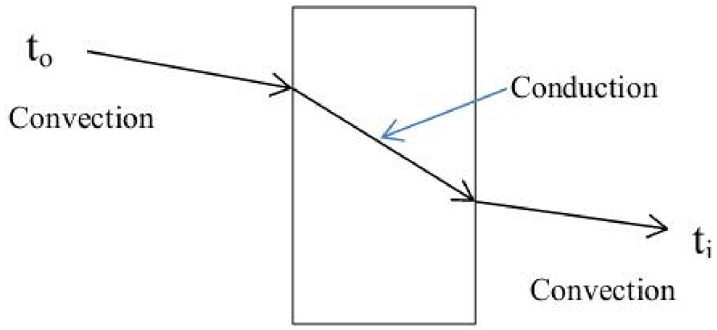
Where

A = area of external wall or roof (m^2)
U = overall heat transfer coefficient of the external wall or roof (W/m^2k).

Heat and moisture transfer through building envelope

Sensible heat gain through opaque surface:

The heat gain through a building structure such as walls, floors, ceiling, doors and windows constitute the major portion of sensible heat load. The temperature difference across opaque surfaces causes heat transfer through these surfaces.
As shown in Figure below, heat from outside air is transferred mainly by convection to outer surface. The heat is then transferred by conduction through the structure to inside surface. The heat from inside surface is transferred by convection to the room air.



Heat transfer through opaque surface

The rate of heat transfer (heat flows) from through a building is calculated by the formula.

$$Q = U \times A \times \Delta T = UA(T_o - T_{in}) \quad \text{(W)}$$

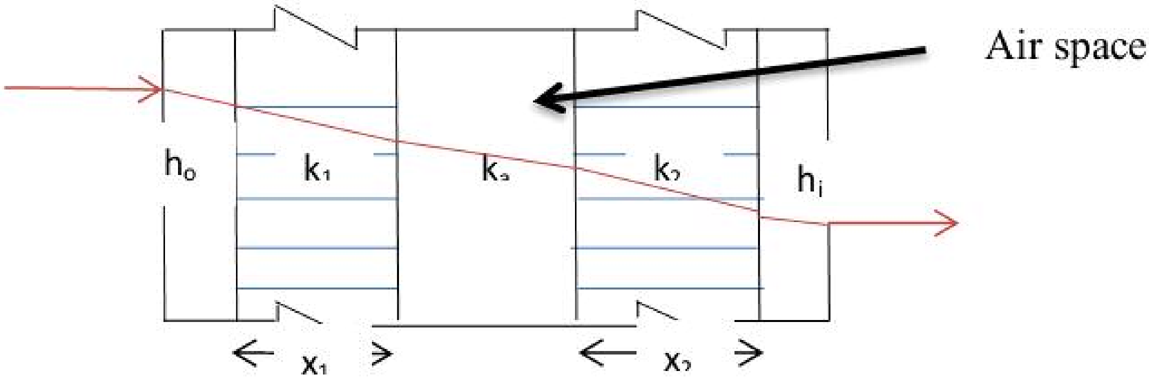
Where,

A: Surface area (m^2)

U : Overall heat transfer coefficient (W/ m^2 °C).
ΔT **:** Temperature difference between outside and inside air (°C or K).
T_o : Outside design temperature of conditioned space (°C or K).
T_{in} : Inside design temperature of conditioned space (°C or K).

Overall heat transfer coefficient (U):

When the wall, floor, or ceiling is made up of layer of different materials as shown in Figure below, then the overall heat transfer coefficient 'U' can be calculated by the formula,



Heat transfer through a composite wall with air space

$$U = \frac{1}{\frac{1}{h_o} + \Sigma\frac{\delta x}{k} + \frac{1}{h_i}} = \frac{1}{R}$$

Where,

h_i: Inside film or surface conductance (W/ m^2 °C).
h_o: Outside film or surface conductance (W/ m^2 °C).
δ_x: Thickness of the wall (m).
k: Thermal conductivity of the wall (W/ m °C).
R: Thermal resistance of a material (°C / W)

Table below explains the procedure of calculation of heat gain through a building.

Direction of surface	A (m^2)	ΔT (°C)	U (W/ m^2 °C)	Q_B (W)
Wall: E W N S				
Window N S				
Door N				
Ceiling				
Floor				
Sum				

Example 1:

Ceiling has dimension of 4 x 3 m consists of the following:

1. 2 cm Tiles (k_T = 1.1 W/ m k).
2. 2 cm Cement mortar (k_M = 0.68 W/ m k).
3. 3 cm Sand (k_S = 0.3 W/ m k).
4. 15 cm Concrete (k_C = 0.72 W/ m k).
5. 2 cm Cement plaster ($k_{C.P}$ = 0.72 W/ m k).
6. Heat-transfer coefficients of outside and inside air are 22.7 W/m^2°C, 8.29 W/m^2°C respectively.
7. Dry bulb temperatures of outside and inside air are 40°C and 25°C respectively. Calculate the following:
   a) Overall heat-transfer for ceiling.
   b) Heat transfer rate through the ceiling.

Solution:

a-

$$U = \frac{1}{\frac{1}{h_o} + \Sigma\frac{\delta x}{k} + \frac{1}{h_i}}$$

b-

$$Q = U \times A \times \Delta T = UA(T_o - T_{in})$$

Heat gain through interior partitions:

Partition in HVAC parlance is defined as an area which is separated by an adjacent non-conditioned space. The space-cooling load due to the conduction heat gain through interior partitions is calculated as:

$$Q = U \times A \times \Delta T = UA(T_a - T_{in})$$

A = Area of partition (m^2).

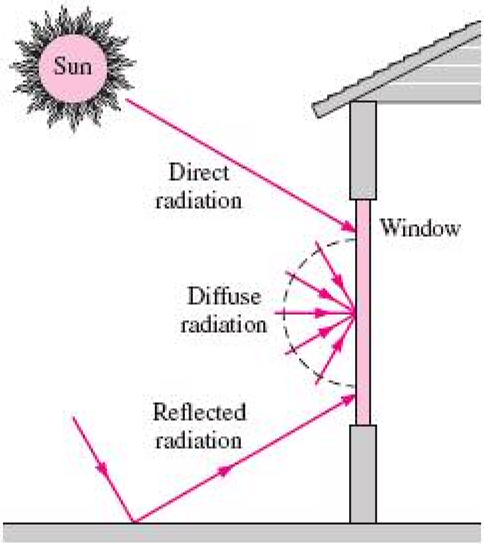
Ta = Temperature of adjacent space (°C or K).

Note:

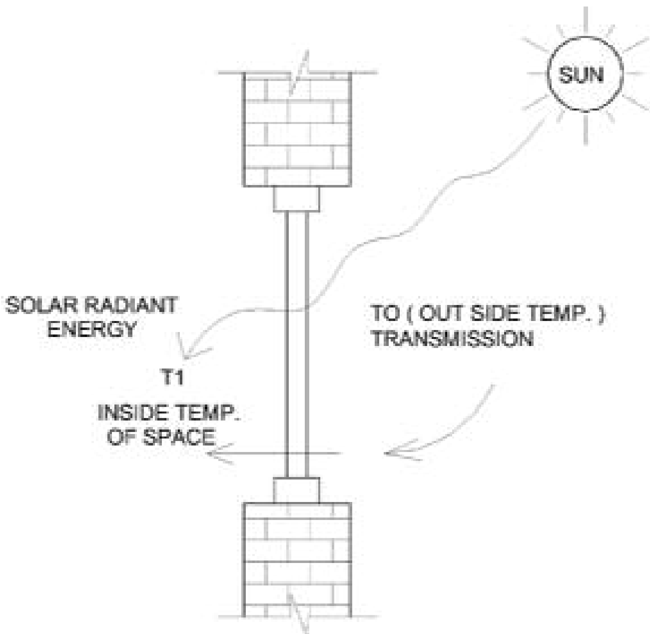
If adjacent space is not conditioned and temperature is not available use outdoor air temperature less 5°F.

Heat gain through glass:

The heat transfer of glass takes place in the two ways, transmission heat gain and solar heat gain. The following equations are used to calculate heat gain from glass areas.



Direct, diffuse, and reflected components of solar radiation incident on a window



Heat gain through glass

Transmission heat gain through glass:

$$Q = U \times A \times \Delta T = UA(T_a - T_{in})$$

By solar radiation:

$$Q = A \times SHGF_{max} \times SC \times CLF$$

$SHGF_{max}$ = maximum solar heat gain factor (W/m^2)
SC = shading coefficient depends on type of shading
CLF = cooling load factor

Table below gives the value of sensible heat gain factor for 22^o North latitude at different orientations and months.

$SHGF_{max}$ for 22^o North latitude

Months	Maximum solar heat gain factor W/m^2					
	N	NE/NW	E/W	SE/SW	S	HOR
January	82	129	568	726	640	647
February	88	243	647	697	533	744
March	98	372	685	609	366	811
April	110	473	662	476	192	826
May	129	527	631	372	129	820
June	167	543	612	328	123	811
July	136	520	618	360	129	808
August	114	461	640	457	205	785
September	104	356	650	587	360	729
October	91	240	621	675	517	644
November	82	129	558	722	631	353
December	79	91	533	729	672	647

The maximum solar heat gain factor $SHGF_{max}$ and CLF depends on latitude, orientation and month.

The cooling load factor (CLF) for different orientation and months are given on in Table below.

Cooling load factor (CLF) for windows glass with indoor shading device

Wall facing	Solar time hr.											
	1	2	3	4	5	6	7	8	9	10	11	12
N	0.08	0.07	0.06	0.06	0.07	0.73	0.66	0.65	0.73	0.8	0.86	0.89
NE	0.03	0.02	0.02	0.02	0.02	0.56	0.76	0.74	0.58	0.37	0.29	0.27
E	0.03	0.02	0.02	0.02	0.02	0.47	0.72	0.8	0.76	0.62	0.41	0.27
SE	0.03	0.03	0.02	0.02	0.02	0.3	0.57	0.74	0.81	0.79	0.68	0.49
S	0.04	0.04	0.03	0.03	0.03	0.09	0.16	0.23	0.38	0.58	0.75	0.83
SW	0.05	0.05	0.04	0.04	0.03	0.07	0.11	0.14	0.16	0.19	0.22	0.38
W	0.05	0.05	0.04	0.04	0.03	0.06	0.09	0.11	0.13	0.15	0.16	0.17
NW	0.05	0.04	0.04	0.03	0.03	0.07	0.11	0.14	0.17	0.19	0.2	0.21
HOR	0.06	0.05	0.04	0.04	0.03	0.12	0.27	0.44	0.59	0.72	0.81	0.85

Wall facing	Solar time hr.											
	13	14	15	16	17	18	19	20	21	22	23	24
N	0.89	0.86	0.82	0.75	0.78	0.91	0.24	0.18	0.15	0.13	0.11	0.10
NE	0.26	0.24	0.22	0.20	0.16	0.12	0.06	0.05	0.04	0.04	0.03	0.03
E	0.24	0.22	0.20	0.17	0.14	0.11	0.06	0.05	0.05	0.04	0.03	0.03
SE	0.33	0.28	0.25	0.22	0.18	0.13	0.08	0.07	0.06	0.05	0.04	0.04
S	0.80	0.68	0.50	0.35	0.27	0.19	0.11	0.09	0.08	0.07	0.06	0.05
SW	0.59	0.75	0.81	0.81	0.69	0.45	0.16	0.12	0.10	0.09	0.07	0.06
W	0.31	0.53	0.72	0.82	0.81	0.61	0.16	0.12	0.10	0.08	0.07	0.06
NW	0.22	0.30	0.52	0.73	0.82	0.69	0.16	0.12	0.10	0.08	0.07	0.06
HOR	0.85	0.81	0.71	0.58	0.42	0.25	0.14	0.12	0.10	0.08	0.07	0.06

Heat gain from lighting:

The sensible heat gains from the electric light is given as,

$$Q_l = N \times P \times F_d \times F_{sa} \times CLF \quad = \quad N \times P \times A_{fl}$$

Where,

N : Number of lamps.
F_d : Power per lamp.
F_d : Diversity factor (dimensionless).
F_{sa} : Ballast special allowance factor (dimensionless).
F_d : Cooling load factor (dimensionless).
A_{fl} : Floor area (m^2).

Note:

- The allowance factor is generally use in case of fluorescent tubes allow for the power used by the ballast. Its value is usually taken as 1.25.
- Electric lights depend on the lighting CLF values come from tables and are found in a fashion similar to that for the occupants. The lighting CLF will be 1.0 for building in which the lights are on 24 hours per day or where the cooling system is shut off at night or on the weekends.
- For bulb lamps (normal lamps), ballast special allowance factor (F_{sd}) = 1.0

Example 2:

Calculate the lighting load for 50 bulb lamps (normal lamps), power per lamp is 100W and 20 fluorescent tubes, power per lamp is 100W. Assume that the building in which the lights are on 24 hours per day

Solution:

$$Q_l = N \times P \times F_d \times F_{sa} \times CLF \quad = [50 \times 100 \times 1 \times 1 \times 1]_{Normal\ lamps} + [20 \times 100 \times 1 \times 1.25 \times 1]_{Fluorescet\ tube\ lamps} = 7500\,W$$

Heat gain from occupants:

The human body in a cooled space constitutes cooling load of sensible heat and latent heat. The heat gain from occupants is based on the average number of people that are expected to be present in the conditioned space. The heating load at any given hour due to the occupants is given as,

$$Q_{s,p} = N \times F_d \times q_s \times CLF$$
$$Q_{l,p} = N \times F_d \times q_l$$
$$Q_p = Q_{s,p} + Q_{l,p} = N \times F_d \times (q_s \times CLF + q_l)$$

Where,

$Q_{s,p}$: Person sensible heat.

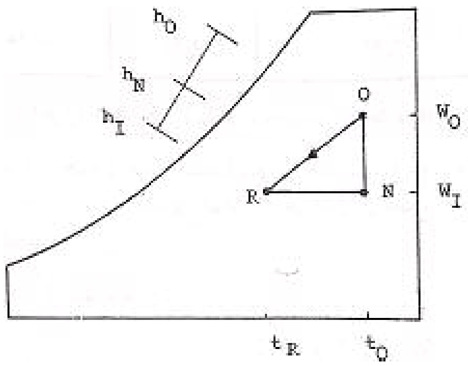
$Q_{l,p}$: Person latent heat.

N : Total number of people present in conditioned space.

F_d : Diversity factor (Dimensionless).
q_s : Sensible heat gains per person (W).
CLF : Cooling load factor (Dimensionless).
q_l : Latent heat gains per person (W).

Heat gain from ventilation:

To calculate the heat gain from ventilation, it is necessary to determine the outside and inside conditions of the conditioned space as shown in Figure below,



Outside and inside conditions of the conditioned space

The equation for determining the heat from ventilation (sensible and latent) is,

$$q_{s,ven} = \dot{m}_{ven}(h_N - h_R) = \dot{m}_{ven} C_{p,air}(T_o - T_R) \text{ (kW)}$$

$$q_{l,ven} = \dot{m}_{ven}(h_o - h_N) = \dot{m}_{ven}(W_o - W_N)h_{fg} = \dot{m}_{ven}(W_o - W_R)h_{fg} \text{ (kW)}$$

$$q_{ven} = q_{s,ven} + q_{l,ven} = \dot{m}_{ven}(h_o - h_R) = \dot{m}_{ven} C_{p,air}(T_o - T_R) \text{ (kW)}$$

Where,

$q_{s,ven}$: Sensible heat of ventilation air (kW)
$q_{l,ven}$: Latent of ventilation air (kW)
q_{ven} : Heat gain from ventilation air (kW)
h_{fg} : Latent heat of evaporation (kJ/kg K)
$\dot{m}_v$: Mass flow rate of ventilation air (kg/s)
h_o : Enthalpy of outside air (kJ/ kg).
h_R : Enthalpy of inside air (kJ/ kg).

Methods of ventilation flow rate of air calculation:

There are two methods of to calculate the ventilation flow rate of air,

1- Basic on air changes per hour (ACH):

$$\dot{q}_{ven} = \frac{V \times NACH}{3600} \qquad (\text{m}^3/\text{s})$$

Or

$$\dot{m}_{ven} = \frac{\dot{q}_{ven}}{v_o} = \frac{V \times NACH}{3600 \times v_o} \qquad (\text{kg/s})$$

Where,

V : Volume of conditioned space (room) m^3.
$\dot{q}_{ven}$: Ventilation flow rate of air (m^3/s).
$NACH$: Number of air changes per hour for conditioned space.

2- Basic on number of persons:

$$\dot{m}_{ven} = n(q_p / person) \ (\text{m}^3/\text{s})$$

or

$$\dot{m}_{ven} = \frac{n(q_p / person)}{v_o} \ (\text{kg/s})$$

Where,
n : Number of persons.
v_o : Specific volume of outside air (m^3/kg).
q_p : Ventilation air rate per person (l/s).

Heat gain from infiltration:

Infiltration is the flow of outdoor air into a building through cracks and other unintentional openings and through the normal use of exterior doors for entrance and egress. Infiltration is also known as air leakage into a building.
There are two methods used to estimate volume flow rate of infiltration air:

(1) Crack method:

The crack method assumes that a reasonably accurate estimate of the rate of air infiltration per square meter of crack opening can be measured or established. Energy codes list maximum permissible infiltration rates for new construction or renovation upgrading.

(2) Air change method:

This procedure for finding the infiltration rate is based on the number of air changes per hour (ACH) in a room caused by infiltration. Equation below can be used to find the air infiltration rate.

$$q_{s,\inf} = \dot{m}_{\inf}\left(h_N - h_R\right) = \dot{m}_{\inf} C_{p,air}\left(T_o - T_R\right) \text{ (kW)}$$

$$q_{l,\inf} = \dot{m}_{\inf}\left(h_o - h_N\right) = \dot{m}_{\inf}\left(W_o - W_N\right)h_{fg} = \dot{m}_{\inf}\left(W_o - W_R\right)h_{fg} \text{ (kW)}$$

$$q_{\inf} = q_{s,\inf} + q_{l,\inf} = \dot{m}_{\inf}\left(h_o - h_R\right) = \dot{m}_{\inf} C_{p,air}\left(T_o - T_R\right) \text{ (kW)}$$

Where,

$q_{s,\inf}$: Sensible heat of infiltration air (kW)

$q_{l,\inf}$: Latent of infiltration air (kW)

$q_{\inf}$: Heat gain from ventilation air (kW)

$\dot{m}_{\inf}$: Mass flow rate of infiltration air (kg/s)

$$\dot{m}_{\inf} = \frac{\dot{q}_{ven}}{v_o} = \frac{V \times NACH}{3600 \times v_o} \quad \text{(kg/s)}$$

Note:

For standard values, the number of air changes per hour (*ACH*) ranges between 0.5 to 1.5.

Example 3:

A room has dimensions of 4 x 3 x 2.75 m. It has the following conditions:
1- outside conditions:

- DBT = 40°C.
- WBT = 26°C.

2- Inside conditions:

- DBT = 25°C.
- RH = 50%.

3- Ventilation rate per person (11.8L/s/p).
4- Seating capacity 3 persons.
5- Number of air changes per hour (1hour).
Calculate the heat gain from ventilation and infiltration.

Solution:

$$\dot{m}_{ven} = \frac{n\left(q_p / person\right)}{v_o}$$

$$q_{s,ven} = \dot{m}_{ven}\left(h_N - h_R\right) = \dot{m}_{ven} C_{p,air}\left(T_o - T_R\right) \text{ (kW)}$$

$$q_{l,ven} = \dot{m}_{ven}\left(h_o - h_N\right) = \dot{m}_{ven}\left(W_o - W_N\right)h_{fg} = \dot{m}_{ven}\left(W_o - W_R\right)h_{fg} \text{ (kW)}$$

$$q_{ven} = q_{s,ven} + q_{l,ven} = \dot{m}_{ven}\left(h_o - h_R\right) = \dot{m}_{ven} C_{p,air}\left(T_o - T_R\right) \text{ (kW)}$$

$$\dot{m}_{\inf} = \frac{\dot{q}_{ven}}{v_o} = \frac{V \times NACH}{3600 \times v_o}$$

$$q_{s,\inf} = \dot{m}_{\inf}\left(h_N - h_R\right) = \dot{m}_{\inf} C_{p,air}\left(T_o - T_R\right) \text{ (kW)}$$

$$q_{l,\inf} = \dot{m}_{\inf}\left(h_o - h_N\right) = \dot{m}_{\inf}\left(W_o - W_N\right)h_{fg} = \dot{m}_{\inf}\left(W_o - W_R\right)h_{fg} \text{ (kW)}$$

$$q_{\inf} = q_{s,\inf} + q_{l,\inf} = \dot{m}_{\inf}\left(h_o - h_R\right) = \dot{m}_{\inf} C_{p,air}\left(T_o - T_R\right) \text{ (kW)}$$

Heat gain from equipment:

Sometimes, equipment may find inside the conditioned spaces, some give sensible heat such as televisions, computers and photographic machines. Heat gain from equipment is calculated as follows,

$$q_E = N\sum(1-\eta)P$$

or

$$q_E = NP$$

Note:

This equation is used when the efficiency of the equipment is unknown

Where,

η: Efficiency of equipment.
N : Number of equipment.
P : Power of equipment.

Calculate the heat gain from equipment two electric motors each have 0.5 kW with efficiency of 70%.

Solution:

$q_E = N\sum(1-\eta)P = 2\times(1-0.7)\times 0.5 = 0.3\ kW$