The CLTD Method

The CLTD method accounts for the thermal response in the heat transfer through the wall or roof, as well as the response due to radiation of part of the energy from the interior surface of the wall to objects and surfaces within the space. The CLTD method makes use of (a) the temperature difference in the case of walls and roofs and (b) the cooling load factors (CLF) in the case of solar heat gain through windows and internal heat sources, that is,

$$Q = U \times A \times CLTD_c \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (6.5)$$

Where

Q: is the net room conduction heat gain through roof, wall or glass (W)

A: is the area of roof, wall or glass (m^2)

U: is the overall heat transfer coefficient ($W/m^2.K$)

$CLTD_c$: is the cooling load temperature difference (^oC)

CLTD/CLF calculation

- **Walls and roofs**

To account for the temperature and the solar variations, the concept of cooling load temperature difference (CLTD) is introduced. The CLTD is a steady-state representation of the complex heat transfer involving actual temperature difference between indoors and outdoors, mass and solar radiation by the building materials, and of time of day. Table 6.1 lists the types of the walls according to installation structural, while Table 6.2 gives the values of CLTDs for different groups of sunlit walls. Table 6.3 lists the CLTDs values for thirteen type of roofs for the typical cooling design day. The following relation makes corrections in the CLTDs listed in the Tables 6.2 and 6.3 for walls and roofs respectively for deviations in design and solar conditions as follows:

$$CLTD_c = [(CLTD + LM)k + (25.5 - T_r) + (T_{o.m} - 29.4)]f \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (6.6)$$

Where

$CLTD_c$: is the corrected value of CLTD

LM: is latitude-month correction from Table (6.4)

K: is a color adjustment for light-coloured roof (1.0 for dark coloured roof; 0.5 if permanently light coloured)

T_r: is the design room temperature

$T_{o.m}$: is the average outdoor temperature, computed as the design temperature less half the daily range.

f: is attic fan factor (1.0 for no attic fans; 0.75 for positive attic ventilation).

Example 6.1:

Calculate the $CLTD_c$ for a wall of group D facing to the south and located at the latitude of 32°N, at 2:00 P.M. in the month of October. Assume that the wall is dark color, and the indoor design temperature is 25.5°C. The maximum outdoor temperature is 35°C with the daily range of 11.2°C.

Solution:

From Table 6.2, and for wall of group D, at solar time 14 and south direction;

The value of CLTD = 9°C

The value of corrected CLTD can be calculate from equation 6.6

$$CLTD_c = (CLTD + LM)k + (25.5 - T_r) + (T_{o.m} - 29.4)$$

$$= (9 + 6.1) \times 1.0 + (25.5 - 25.5) + (29.4 - 29.4) = 15.1^oC$$

Example 6.2:

Roof of one of the buildings consists of 102 mm high weight concrete with 50.8 mm insulation, and a suspended ceiling. The overall heat transfer coefficient of the roof is 0.511 W/m^2.°C. The building is located at latitude of 40°N. The outdoor design conditions are 36°C DBT and 26°C WBT, with daily range of 12°C, and the indoor design conditions are 24°C DBT and 50% relative humidity. Compute the cooling load per square meter of the roof at 4:00 P.M. in the month of August.

Solution:

Roof No. 9, U= 0.511 W/m^2.°C, 40°N

CLTD = 19°C, LM = 1.6, $T_{o.m} = 36 - (12/2) = 30^oC$

$$CLTD_c = [(19 + 1.6) \times 1 + (25.5 - 24) + (30 - 29.4)] \times 1 = 22.7^oC$$

$Q/A = U.CLTD_c = 0.511 \times 22.7 = 11.6\ W/m^2$

- **Windows and glass**

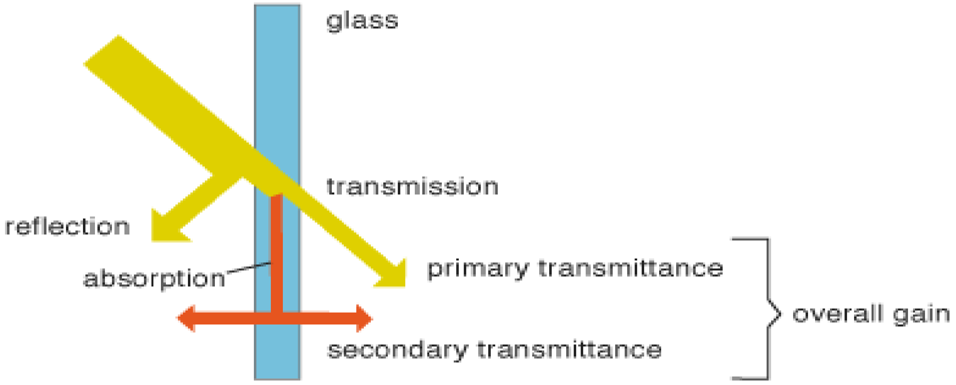

Figure 6.3 Absorption, reflection and transmission of solar radiation in glass.

When solar radiation strikes an unshaded window (Fig.6.3), about 8% of the radiant energy is typically reflected back outdoors, from 5-50% is absorbed within the glass, and the remainder is transmitted directly indoors, to become part of the cooling load.

The solar gain is the sum of the transmitted radiation and the portion of the absorbed radiation the flows inward. Because heat is also conducted through the glass wherever there is an outdoor- indoor temperature difference, the total of heat admission is

Total heat admission through glass = Radiation transmitted through glass + Inward flow of absorbed solar radiation + Conduction heat gain

We can rewrite the above equation

Total heat gain = Solar heat gain + Conduction heat gain ………………. (6.7)

Conduction heat gain = $U_g.A_g.CLTD_c$ …………………………………….. (6.8)

where

U_g: overall heat transfer coefficient of the glass

A_g: Area of the glass

$CLTD_c$: Correct cooling load temperature difference for glass. Table 6.5 gives CLTD.

The correction of the CLTD is as follow;

$CLTD_c = CLTD + (25.5 – T_r) + (T_{o.m} – 29.4)$ ………………………….. (6.9)

Solar heat gain = A_g. [SHG_{max} x SC x CLF]……………………… (6.10)

Where

SHG_{max}: Maximum solar heat gain in W/m^2 from Table 6.6

SC: Shading coefficient from Table 6.7, 6.8, 6.9, and 6.10

CLF: Cooling load factor for glass from Table 6.11 and 6.12

Shading coefficient (SC); is the ratio between the solar heat gain through any given type of fenestration system and the solar heat gain through unshaded clear glass (reference glass)

SC = (Solar heat gain of fenestration)/ (Solar heat gain of double-strength glass)… (6.11)

Example 6.3:

The wall of Example 6.1 has a 1.2 m x 1.5 m single glass window (U = 4.6 W/m^2.^{o}C). The window has light-colored venetian blinds. Compute the cooling load due to the window at 5:00 P.M. solar time for August, using the design condition given in Example 6.1.

Solution:

The total cooling load for the window can be determined as following;

$Q_g = A_g[\ U \times CLTD_c + SC \times SHG_{max} \times CLF\ ]$

$CLTD_c = CLTD + (25.5 – T_r) + (T_{o.m} – 29.4) = 7 + (25.5 - 25.5) + (29.4 – 29.4) = 7^oC$

SC = 0.55 from Table 6.8

SHG_{max} = 350 W/m^2 from Table 6.6

CLF = 0.27 from Table 6.12

$Q_g = 1.8[4.6 \times 7 + 0.55 \times 350 \times 0.27]$

= 151.5 W

- **Space cooling load through partitions**

Whenever a conditioned space in adjacent to a space with a difference temperature, transfer of heat through the separating physical section must be considered

$q_p = U \times A \times (T_a - T_r)$ ……………………………….. (6.11)

Where

U: overall heat transfer coefficient of the partition ($W/m^2.^oC$)

A: area of partition (m^2)

T_a: temperature of the adjacent space (^oC)

T_r: temperature of the conditioned space (^oC)

- **Infiltration heat gain**

The infiltration load should be considered in the space heat gain calculations. Sensible and latent heat from infiltration gains can be calculated in the same way that already explained in Chapter Five with some simple difference.

Sensible infiltration heat gain, $q_{s.i} = 1.22 \times V^{\cdot}_i \times (T_o - T_r)$ …………………… (6.12)

Latent infiltration heat gain, $q_{l.i} = 3010 \times V^{\cdot}_i \times (W_o - W_i)$…………………… (6.13)

Where

$V^{\cdot}_i$: is the infiltration volume flow rate (m^3/s) or (l/s)

Heat Gain from Internal Sources

Internal heat comprises sensible and latent heat gains from occupants, lights, appliances and equipment and piping, etc.

- **Occupancy**

The people who occupy the building give off thermal energy continuously, the rate of which depends on the level of activity (Table 6.13). For the sensible portion of the heat released, a cooling load factor (Table 6.14) has been developed to account for the lag in time between occupancy and the observed cooling load. The sensible cooling load due to people is therefore,

$$q_s = N \times G_s \times CLF_s \quad \ldots\ldots (6.14)$$

Where

q_s = sensible cooling load due to occupants (W)

N = number of occupants

G_s = sensible heat gain depending on activity and time from entry (W)

CLF_s = cooling load factor (dimensionless) for people.

The latent heat gain from occupants is found by

$$q_L = N \times G_L \quad \ldots\ldots (6.15)$$

Where

q_L: latent heat gain from occupants (W)

N: number of occupants

G_L: latent heat gain from occupants depending on a activity and time from entry (W)

Example 6.4:

An office suite is designed with 15 people. Estimate the cooling load from the occupants after 7 hours of their entering the office. Also, assumed that the occupants stay in the space are 9 hours.

Solution:

We will assume moderately active, office work, and use data from Table 6.13 and 6.14

q_L = 55 W/person, q_s = 75 W/person, CLF_s = 0.825

$q_L = 15 \times 55 = 825$ W

$q_s = 15 \times 75 \times 0.825 = 928$ W

$Q_P = q_L + q_s = 825 + 928 =$ 1753 W = 1.753 kW

- **Lighting**

Lighting is often the major space cooling load component. The rate of heat gain at any instant, however, is not the same as the heat equivalent of power supplied instantaneously to these lights. Only part of the energy from lights is transferred to the room air by convection, and thus becomes the cooling load. The remaining portion is the radiant heat that affects the conditioned space only after having been absorbed by walls, floors furniture, etc. and released after a time lag. The instantaneous heat gain for lights may be expressed as

$$q_L = W \times F_u \times F_s \quad \ldots\ldots\ldots\ldots \quad (6.\ 16)$$

Where

q_L: instantaneous heat gain for lights

W: summation of all installed light wattage

F_u: use factor- ratio of wattage in use to that installed

F_s: special allowance factor for lights, for fluorescent lamp F_s= 1.2

The cooling load is then given by

$$Q_L = q_L \times (CLF)_L \quad \ldots\ldots\ldots\ldots \quad (6.17)$$

The cooling load factor is a function of the building mass, air-circulation rate, type of fixture and time. Table 6.15 give the cooling load as a function of time for lights that are on for 8, 10, 12 and 14 hour. The "a" classification depends on the nature of light fixture, the return-air system, and the type of furnishings, where the "b" classification depends on the construction of the building and the type of supply and return air system. Design values of coefficients "a" and "b" are given in Tables 6.16 and 6.17 respectively.

Example 6.5:

The office suite of example 6.4 has total installed light wattage of 8400 W. The fluorescent light fixtures are recessed with 40 W lamps. Supply air is through the ceiling with air returning through the ceiling plenum. The lights are turned on at 8:00 A.M. and turned off at 6:00 P.M. Estimate the cooling load at 4:00 P.M. The floor is 75 mm concrete.

Solution:

Assuming that about 15% of the lights are off, the use factor; F_u = 85%, F_s = 1.2

The instantaneous heat gain for lights is

$q_L = W \times F_u \times F_s = 8400 \times 0.85 \times 1.2 = 8568$ W

The cooling load is then

$Q_L = q_L \times (CLF)_L$

The $(CLF)_L$ from Table 6.15 for 10 hours on and a = 0.55 (Table 6.16), b = B (Table 6.17)

$(CLF)_L = 0.82$

$Q_L = 8568 \times 0.82 =$ 7026 W

- **Miscellaneous Equipment**

Most appliances contribute both sensible and latent heats. The latent heat produced depends on the function the appliances perform, such as drying, cooking, etc. Gas appliances produce additional moisture as product of combustion.

The heat gain from Equipment Q_M is

$Q_{M.S} = q_{M.S} \times CLF_M$ ………………………………………… (6.18)

Where

$q_{M.S}$: sensible heat gain from appliances Table 6.18

CLF_M: cooling load factor for appliances Table 6.19

$Q_{M.L} = q_{M.L} \times$ No. of appliances …………………………… (6.19)

Where

$q_{M.L}$: latent heat gain from appliances Table 6.18

- **Heat Gain from Ventilation Air**

Provision of ventilation air is mandated by local codes and ordinances. The ASHRAE standard 62 recommends minimum ventilation rates for most common applications. For general applications, such as offices, 10 l/s per person is recommended. Ventilation air is generally introduced at the air handling unit (AHU) rather than directly into the conditioned space. It thus becomes a cooling coil load component instead of a space load component.

Heat gain corresponding to a flow rate of $V^{\cdot}$ through an enthalpy difference of Δh (for an air density of 1.20 kg/m^3) is shown below:

Sensible heat gain corresponding to the change in dry bulb temperature ΔT for a given air flow $V^{\cdot}$ is

$q_s = 1.22 \times V^{\cdot} \times \Delta T$ ……………………………….…… (6.20)

where

q_s: is the sensible heat gain (kW)

$V^{\cdot}$: is the ventilation flow rate (m^3/s)

$\Delta T = (T_o - Tr)$

Latent heat gain corresponding to the change in moisture content (ΔW) for a given air flow $V^{\cdot}$ is

$q_L = 3010 \times V^{\cdot} \times \Delta W$ ………………………………………… (6.21)

where

q_L: is the latent heat gain

$\Delta W = (W_o - W_r)$

- **<u>Required Air Quantity</u>**

The air quantity required to offset simultaneously the room sensible and latent loads may be calculated using the following equation

Air flow required $V^{\cdot}_R$ = room (zone) sensible heat/ 1.22 $(T_r - T_s)$ …………… (6.22)

where

T_r: is the room temperature (^oC)

T_s: is the supply air temperature (^oC)

The problem is how we can calculate or determine the supply air temperature T_s. One of the two unknown ($V^{\cdot}_R$, T_s) is chosen according to "good practice" (such as costs and job conditions), and the remaining unknown is then calculated from the equation.

Example 6.6:

A hair salon shop has a sensible cooling load of 16 kW and latent cooling load of 6.5 kW. The room conditions are to be maintained at 25ºC DBT and 50% RH. If 56 m^3/min of supply air is furnished, determine the required supply air DBT and WBT.

Solution:

$q_s = 1.22 \times V^{\cdot} \times (T_r - T_s)$

$16 = 1.22 \times (56/60) \times (25 - T_s)$

$T_s = 11^oC$ (DBT of the supply air)

$q_L = 3010 \times V^{\cdot} \times (W_r - W_s)$

$6.5 = 3010 \times (56/60) \times (0.01 - W_s)$

$W_s = 0.00232$ kg/kg dry air (moisture content of the supply air)

Now from the psychrometric chart at DBT=11 ºC and W= 0.00232 kg/kg, we find WBT=9ºC.

- General Design Guidelines

The general procedure required to calculate the space cooling load is as follows:

(a) *Building configuration and characteristics*: Determine the building location, orientation and external shading, building materials, external surface color and shape. These details are usually obtained from building plans and specifications.

(b) *Outdoor design conditions*: Obtain the outdoor weather data for the building location and select the outdoor design conditions.

(c) *Indoor design conditions*: Specify temperature, humidity, air velocity, etc.

(d) *Operating schedules*: Obtain a schedule of lighting, occupancy, internal equipment, appliances and processes generating heat load.

(e) *Date and time*: Select the time of the day and month to estimate the cooling load.

Several different times of the day and several different months need to be analysed to determine the peak load time. The particular day and month are often dictated by peak solar conditions. A calculation form [Table 6.20] is given blow, wherein the calculated values of the components of cooling load may be entered.

Exercises

1. A window in a south wall in a building which has a light construction. The dimensions of the window are (1.5 m x 1.5 m) and the glass is of heat absorbing type (U= 5.91 W/m^2.^{o}C) and thickness of 6 mm. The building is located at latitude of 32^{o}N. Outdoor design condition is 42^{o}C, with daily range of 14^{o}C, and the indoor design temperature is 25^{o}C. Calculate the cooling load from the window at 3:00 P.M. in the month of July.

2. Roof of one of the building consists of 152.4 mm high weight concert with 50.8 mm insulation, and without suspended ceiling. The overall heat transfer coefficient of the roof is 0.664 W/m^2.^{o}C, and the building is located at latitude of 48^{o}N. The outdoor design condition is 38^{o}C DBT with daily range of 11^{o}C, and the indoor design condition is 25^{o}C DBT and 50% relative humidity. Calculate the cooling load per square meter of the roof at 2:00 P.M. in the month of July.

3. Hall meetings can accommodate for 100 person located in Mosul city

(a) Choose the appropriate outdoor and indoor design conditions for the summer.

(b) The sensible heat and the latent heat gains from the occupants of the hall.

(c) The amount of air ventilation required to this hall, and the cooling load of this amount.

4. The west wall in a building in Baghdad (32^{o}N) has a window of the dimensions (1.0 m x 2.0 m). The glass is of heat absorbing type (U= 4.6 W/m^2.^{o}C) and thickness of 6 mm. Curtains type (III_D) was used. Outdoor design condition is 34^{o}C DBT, with daily range of 11^{o}C, and indoor design temperature is 24^{o}C. Calculate the cooling load from the window at 4:00 P.M. in the month of August.

5. A window in a southern west wall in a building which has a medium construction. The dimensions of the window are (2.0 m x 1.5 m), and the glass is of the clear type (U= 5.91 W/m^2.^{o}C) and thickness of 6 mm. The glass was shaded from the outside. The building is located at latitude of 36^{o}N. Outdoor design conditions are 35^{o}C DBT and 50% RH, with the daily range of 11.5^{o}C, and the indoor design conditions are 26^{o}C DBT and 18^{o}C WBT. Compute the cooling load from the window at 5:00 P.M. in the month of September.

6. An air-conditioning system is to be design for a restaurant with the following data:

Outside design conditions: 40°C DBT, 28°C WBT

Inside design conditions: 25°C DBT, 50% RH

Solar heat gain through walls, roofs and floor: 5.87 kW

Solar heat gain through glass: 5.52 kW

Occupant: 25

Sensible heat gain per person: 58 W, Latent heat gain per person: 58 W

Internal lighting load: 15 lamps of 100 W, 10 fluorescent of 80 W

Sensible heat gain from other sources: 11.63 kW

Rate of Infiltration air: 15 m^3/min

If 25% fresh air and 75% recirculate air is mixed and passed through the conditioner coil, find:

(a) The amount of total air required in m^3/min.

(b) The apparatus dew point temperature of the coil.

(c) The condition of the supply air to the room.

(d) The capacity of the conditioning plant.

Assume the by –pass factor equal to 0.2. Draw the schematic diagram of the system and show the system on psychrometric chart and insert the temperature and enthalpy values at salient points.