



PDHonline Course M318 (5 PDH)

HVAC Cooling Load Calculations and Principles

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1.0 OBJECTIVE

Cooling load calculations may be used to accomplish one or more of the following objectives:

- a) Provide information for equipment selection, system sizing and system design.
- b) Provide data for evaluating the optimum possibilities for load reduction.
- c) Permit analysis of partial loads as required for system design, operation and control.

This course provides a procedure for preparing a manual calculation for cooling load. A number of published methods, tables and charts from industry handbooks, manufacturer's engineering data and manufacturer's catalog data usually provide a good source of design information and criteria in the preparation of the HVAC load calculation. It is not the intent of this course to duplicate this information but rather to extract appropriate data from these documents as well as provide a direction regarding the proper use or application of such data so that engineers and designers involved in preparing the calculations can make the appropriate decision and/or apply proper engineering judgment.

The course includes two example calculations for better understanding of the subject.

2.0 TERMINOLOGY

Commonly used terms relative to heat transmission and load calculations are defined below in accordance with ASHRAE Standard 12-75, Refrigeration Terms and Definitions.

Space – is either a volume or a site without a partition or a partitioned room or group of rooms.

Room – is an enclosed or partitioned space that is usually treated as single load.

Zone – is a space or group of spaces within a building with heating and/or cooling requirements sufficiently similar so that comfort conditions can be maintained throughout by a single controlling device.

British thermal unit (Btu) - is the approximate heat required to raise 1 lb. of water 1 deg Fahrenheit, from 59⁰F to 60⁰F. Air conditioners are rated by the number of British Thermal Units (Btu) of heat they can remove per hour. Another common rating term for air conditioning size is the "ton," which is 12,000 Btu per hour and Watts. Some countries utilize one unit, more than the others and therefore it is good if you can remember the relationship between *BTU/hr*, *Ton*, and *Watts*.

- 1 ton is equivalent to 12,000 BTU/hr. and
- 12,000 BTU/hr is equivalent to 3,516 Watts - or 3.516 kW (kilo-Watts).

Cooling Load Temperature Difference (CLTD) – an equivalent temperature difference used for calculating the instantaneous external cooling load across a wall or roof.

Sensible Heat Gain – is the energy added to the space by conduction, convection and/or radiation.

Latent Heat Gain – is the energy added to the space when moisture is added to the space by means of vapor emitted by the occupants, generated by a process or through air infiltration from outside or adjacent areas.

Radiant Heat Gain – the rate at which heat absorbed is by the surfaces enclosing the space and the objects within the space.

Space Heat Gain – is the rate at which heat enters into and/or is generated within the conditioned space during a given time interval.

Space Cooling Load – is the rate at which energy must be removed from a space to maintain a constant space air temperature.

Space Heat Extraction Rate - the rate at which heat is removed from the conditioned space and is equal to the space cooling load if the room temperature remains constant.

Temperature, Dry Bulb – is the temperature of air indicated by a regular thermometer.

Temperature, Wet Bulb – is the temperature measured by a thermometer that has a bulb wrapped in wet cloth. The evaporation of water from the thermometer has a cooling effect, so the temperature indicated by the wet bulb thermometer is less than the temperature indicated by a dry-bulb (normal, unmodified) thermometer. The rate of evaporation from the wet-bulb thermometer depends on the humidity of the air. Evaporation is slower when the air is already full of water vapor. For this reason, the difference in the temperatures indicated by ordinary dry bulb and wet bulb thermometers gives a measure of atmospheric humidity.

Temperature, Dewpoint – is the temperature to which air must be cooled in order to reach saturation or at which the condensation of water vapor in a space begins for a given state of humidity and pressure.

Relative humidity - describes how far the air is from saturation. It is a useful term for expressing the amount of water vapor when discussing the amount and rate of evaporation. One way to approach saturation, a relative humidity of 100%, is to cool the air. It is therefore useful to know how much the air needs to be cooled to reach saturation.

Thermal Transmittance or Heat Transfer Coefficient (U-factor) – is the rate of heat flow through a unit area of building envelope material or assembly, including its boundary films, per unit of temperature difference between the inside and outside air. The U-factor is expressed in $\text{Btu}/(\text{hr } ^\circ\text{F ft}^2)$.

Thermal Resistance (R) – is the reciprocal of a heat transfer coefficient and is expressed in $(\text{hr } ^\circ\text{F ft}^2)/\text{Btu}$. For example, a wall with a U-value of 0.25 would have a resistance value of $R = 1/U = 1/0.25=4.0$. The value of R is also used to represent Thermal Resistivity, the reciprocal of the thermal conductivity.

3.0 SIZING YOUR AIR-CONDITIONING SYSTEM

Concepts and fundamentals of air conditioner sizing is based on heat gain, and/or losses in a building. It is obvious that you will need to remove the amount of heat gain - if it is hot outside. Similarly, you'll need to add in the heat loss from your space - if outside temperature is cold. In short, heat gain and loss, must be *equally* balanced by heat removal, and addition, to get the desired room comfort that we want.

The heat gain or heat loss through a building depends on:

- a. The temperature difference between outside temperature and our desired temperature.
- b. The type of construction and the amount of insulation is in your ceiling and walls. Let's say, that you have two identical buildings, one is build out of glass, and the other out of brick. Of course the one built with glass would require much more heat addition, or removal, compared to the other - given a same day. This is because the glass has a high thermal conductivity (U-value) as compared to the brick and also because it is transparent, it allows direct transmission of solar heat.
- c. How much shade is on your building's windows, walls, and roof? Two identical buildings with different orientation with respect to the direction of sun rise and fall will also influence the air conditioner sizing.
- d. How large is your room? The surface area of the walls. The larger the surface area - the more heat can loose, or gain through it.
- e. How much air leaks into indoor space from the outside? Infiltration plays a part in determining our air conditioner sizing. Door gaps, cracked windows, chimneys - are the "doorways" for air to enter from outside, into your living space.
- f. The occupants. It takes a lot to cool a town hall full of people.
- g. Activities and other equipment within a building. Cooking? Hot bath? Gymnasium?
- h. Amount of lighting in the room. High efficiency lighting fixtures generate less heat.
- i. How much heat the appliances generate. Number of power equipments such as oven, washing machine, computers, TV inside the space; all contribute to heat.

The air conditioner's efficiency, performance, durability, and cost depend on matching its size to the above factors. Many designers use a simple square foot method for sizing the air-conditioners. The most common rule of thumb is to use "1 ton for every 500 square feet of floor area". Such a method is useful in preliminary estimation of the equipment size. The main drawback of rules-of-thumb methods is the presumption that the building design will not make any difference. Thus the rules for a badly designed building are typically the same as for a good design.

It is important to use the correct procedure for estimating heat gain or heat loss. Two groups—the Air Conditioning Contractors of America (ACCA) and the American Society of Heating, Refrigerating, and Air Conditioning Engineers (ASHRAE)—publish calculation procedures for sizing central air conditioners.

Reputable air conditioning contractors will use one of these procedures, often performed with the aid of a computer, to size your new central air conditioner.

3.1 Heating Load V/s Cooling Load Calculations

As the name implies, heating load calculations are carried out to estimate the heat loss from the building in winter so as to arrive at required heating capacities. Normally during winter months the peak heating load occurs before sunrise and the outdoor conditions do not vary significantly throughout the winter season. In addition, internal heat sources such as occupants or appliances are beneficial as they compensate some of the heat losses. As a result, normally, the heat load calculations are carried out assuming steady state conditions (no solar radiation and steady outdoor conditions) and neglecting internal heat sources. This is a simple but conservative approach that leads to slight overestimation of the heating capacity. For more accurate estimation of heating loads, one has to take into account the thermal capacity of the walls and internal heat sources, which makes the problem more complicated.

For estimating cooling loads, one has to consider the unsteady state processes, as the peak cooling load occurs during the day time and the outside conditions also vary significantly throughout the day due to solar radiation. In addition, all internal sources add on to the cooling loads and neglecting them would lead to underestimation of the required cooling capacity and the possibility of not being able to maintain the required indoor conditions. Thus cooling load calculations are inherently more complicated.

In determining the heating load, credit for solar heat gain or internal heat gains is usually NOT included and the thermal storage effects of building structure are generally ignored. Whereas in cooling load calculations, the thermal storage characteristics of the building play a vital role because the time at which the space may realize the heat gain as a cooling load will be considerably offset from the time the heat started to flow.

We will discuss this further in succeeding sections.

4.0 HEAT FLOW RATES

In air-conditioning design, four related heat flow rates, each of which varies with time, must be differentiated:

- a. Space heat gain -----How much heat (energy) is entering the space?
- b. Space cooling load -----How much energy must be removed from the space to keep temperature and relative humidity constant?
- c. Space heat extraction-----How much energy is the HVAC removing from the space?
- d. Cooling load (coil) -----How much energy is removed by the cooling coil serving various spaces plus any loads external to the spaces such as duct heat gain, duct leakage, fan heat and outdoor makeup air?

4.1 Space Heat Gain

This instantaneous rate of heat gain is the rate at which heat enters into and/or is generated within a space at a given instant. Heat gain is classified by:

The manner in which it enters the space –

- a. Solar radiation through transparent surfaces such as windows
- b. Heat conduction through exterior walls and roofs
- c. Heat conduction through interior partitions, ceilings and floors
- d. Heat generated within the space by occupants, lights, appliances, equipment and processes
- e. Loads as a result of ventilation and infiltration of outdoor air
- f. Other miscellaneous heat gains

Whether it is a sensible or latent gain –

Sensible heat - Heat which a substance absorbs, and while its temperature goes up, the substance does not change state. Sensible heat gain is directly added to the conditioned space by conduction, convection, and/or radiation. Note that the sensible heat gain entering the conditioned space does not equal the sensible cooling load during the same time interval because of the stored heat in the building envelope. Only the convective heat becomes cooling load instantaneously. Sensible heat load is total of

- a. Heat transmitted thru floors, ceilings, walls
- b. Occupant's body heat
- c. Appliance & Light heat
- d. Solar Heat gain thru glass
- e. Infiltration of outside air
- f. Air introduced by Ventilation

Latent Heat Loads - Latent heat gain occurs when moisture is added to the space either from internal sources (e.g. vapor emitted by occupants and equipment) or from outdoor air as a result of infiltration or ventilation to maintain proper indoor air quality. Latent heat load is total of

- a. Moisture-laden outside air from Infiltration & Ventilation
- b. Occupant Respiration & Activities

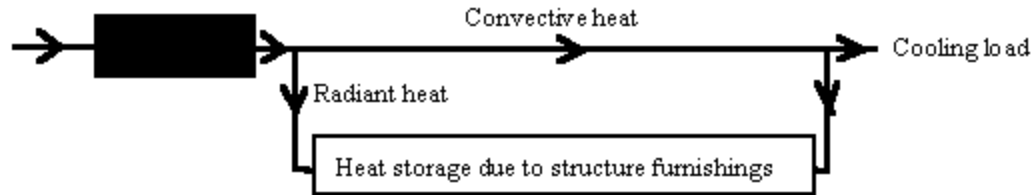
c. Moisture from Equipment & Appliances

To maintain a constant humidity ratio, water vapor must condense on cooling apparatus at a rate equal to its rate of addition into the space. This process is called dehumidification and is very energy intensive, for instance, removing 1 kg of humidity requires approximately 0.7 kWh of energy.

4.2 Space Heat Gain V/s Cooling Load (Heat Storage Effect)

Space Heat Gain is \neq to Space Cooling Load

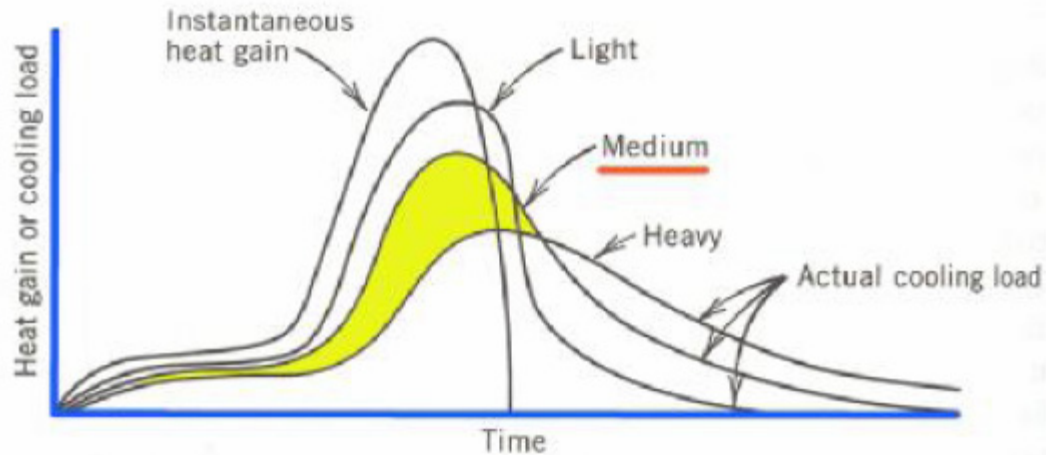
The heat received from the heat sources (conduction, convection, solar radiation, lightning, people, equipment, etc...) does not go immediately to heating the room air. Only some portion of it is absorbed by the air in the conditioned space instantaneously leading to a minute change in its temperature. Most of the radiation heat especially from sun, lighting, people is first absorbed by the internal surfaces, which include ceiling, floor, internal walls, furniture etc. Due to the large but finite thermal capacity of the roof, floor, walls etc., their temperature increases slowly due to absorption of radiant heat. The radiant portion introduces a time lag and also a decrement factor depending upon the dynamic characteristics of the surfaces. Due to the time lag, the effect of radiation will be felt even when the source of radiation, in this case the sun is removed.



Differences between Space Heat Gain and Space Cooling Load

Differences between instantaneous heat gain and cooling load is due to heat storage affect.

The relation between heat gain and cooling load and the effect of the mass of the structure (light, medium & heavy) is shown below. From the figure it is evident that, there is a delay in the peak heat, especially for heavy construction.



Actual cooling load and solar heat gain for light, medium and heavy construction

4.3 Space Cooling V/s Cooling Load (Coil)

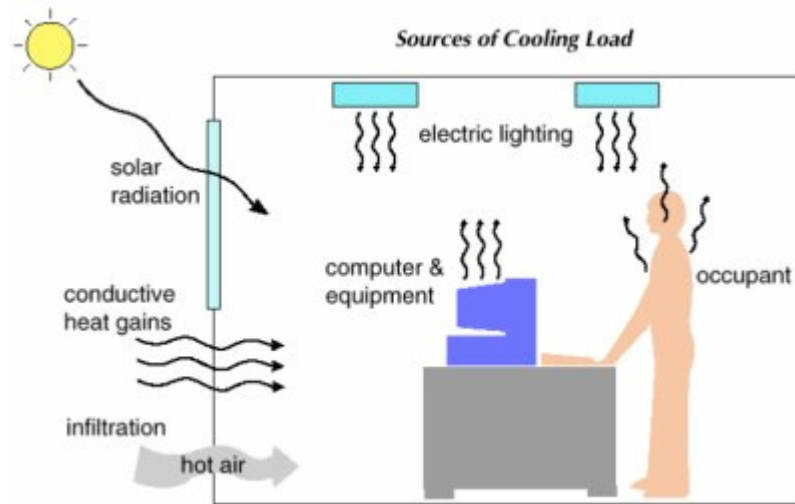
Space cooling is the rate at which heat must be removed from the spaces to maintain air temperature at a constant value. Cooling load, on the other hand, is the rate at which energy is removed at the cooling coil that serves one or more conditioned spaces in any central air conditioning system. It is equal to the instantaneous sum of the space cooling loads for all spaces served by the system plus any additional load imposed on the system external to the conditioned spaces items such as fan energy, fan location, duct heat gain, duct leakage, heat extraction lighting systems and type of return air systems all affect component sizing.

5.0 COMPONENTS OF COOLING LOAD

The total building cooling load consists of heat transferred through the building envelope (walls, roof, floor, windows, doors etc.) and heat generated by occupants, equipment, and lights. The load due to heat transfer through the envelope is called as **external load**, while all other loads are called as **internal loads**. The percentage of external versus internal load varies with building type, site climate, and building design. The total cooling load on any building consists of both **sensible** as well as **latent** load components. The sensible load affects the dry bulb temperature, while the latent load affects the moisture content of the conditioned space.

Buildings may be classified as externally loaded and internally loaded. In externally loaded buildings the cooling load on the building is mainly due to heat transfer between the surroundings and the internal conditioned space. Since the surrounding conditions are highly variable in any given day, the cooling load of an externally loaded building varies widely. In internally loaded buildings the cooling load is mainly due to internal heat generating sources such as occupants, lights or appliances. In general the heat generation due to internal heat sources may remain fairly constant, and since the heat transfer from the variable surroundings is much less compared to the internal heat sources, the cooling load of an internally loaded building remains fairly constant. Obviously from energy efficiency and economics points of view, the system design strategy for an externally loaded building

should be different from an internally loaded building. Hence, prior knowledge of whether the building is externally loaded or internally loaded is essential for effective system design.



6.0 COOLING LOAD CALCULATION METHOD

For a thorough calculation of the zones and whole-building loads, one of the following three methods 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 spreadsheet.
- 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 spreadsheet programs but has some limitations due to the use of tabulated data.
- c. Total Equivalent Temperature Differential/Time-Averaging (TETD/TA): This was the preferred method for hand or simple spreadsheet calculation before the introduction of the CLTD/CLF method.

These three methods are well documented in ASHRAE Handbook Fundamentals, 2001.

6.1 Accuracy and Reliability of Various Calculation Methods

For each cooling load calculation method, there are several benefits/limitations which feature each method. Simplicity and accuracy are two contradicting objectives to be fulfilled. If a method could be considered to be simple, its accuracy would be a matter of question, and vice versa.

While modern methods emphasize on improving the procedure of calculating solar and conduction heat gains, there are also other main sources coming from internal heat gains (people, lighting and equipment).

Handbooks include tables for the heat gain estimations from the internal sources. However, such tables are incomplete. For example, for equipment not mentioned in the tables, only limited information is indicated about them. Sometimes recommendations are mentioned about using 25% to 50% of the nameplate power consumption, where the final value is left to the interpretation of the designer. In other times it is the accurate predictability of the occurrence is also important, e.g. the frequency of using of equipment is very important to determine the heat gain. This example for internal heat gain shows that, when thinking about accuracy, it is not only the method (simple vs. complex) which is effective, but uncertainties in the input data are also important.

There are high degrees of uncertainty in input data required to determine cooling loads. Much of this is due to the unpredictability of occupancy, human behavior, outdoors weather variations, lack of and variation in heat gain data for modern equipments, and introduction of new building products and HVAC equipments with unknown characteristics. These generate uncertainties that far exceed the errors generated by simple methods compared to more complex methods. Therefore, the added time/effort required for the more complex calculation methods would not be productive in terms of better accuracy of the results if uncertainties in the input data are high. Otherwise, simplified methods would, likely, have a similar level of satisfactory accuracy.

For strictly manual cooling load calculation method, the most practical to use is the CLTD/SCL/CLF method as described in the 1997 ASHRAE Fundamentals. This method, although not optimum, will yield the most conservative results based on peak load values to be used in sizing equipment. It should be noted that the results obtained from using the CLTD/CLF method depend largely on the characteristics of the space being considered and how they vary from the model used to generate the CLTD/CLF data shown on the various tables. Engineering judgment is required in the interpretation of the custom tables and applying appropriate correction factors.

7.0 DESIGN INFORMATION

To calculate the space cooling load, detailed building information, location, site and weather data, internal design information and operating schedules are required. Information regarding the outdoor design conditions and desired indoor conditions are the starting point for the load calculation and is discussed below.

7.1 Outdoor Design Weather Conditions

ASHRAE Handbook 1993 Fundamentals (Chapter 26) list tables of climate conditions for the US, Canada and other International locations: In these tables:

The information provided in table 1a, 2a and 3a are for heating design conditions that include:

- a. Dry bulb temperatures corresponding to 99.6% and 99% annual cumulative frequency of occurrence,

- b. Wind speeds corresponding to 1%, 2.5% and 5% annual cumulative frequency of occurrence,
- c. Wind direction most frequently occurring with 99.6% and 0.4% dry-bulb temperatures and
- d. Average of annual extreme maximum and minimum dry-bulb temperatures and standard deviations.

The information provided in table 1b, 2b and 3b are for cooling and humidity control conditions that include:

- a. Dry bulb temperature corresponding to 0.4%, 1.0% and 2.0% annual cumulative frequency of occurrence and the mean coincident wet-bulb temperature (warm). These conditions appear in sets of dry bulb (DB) temperature and the mean coincident wet bulb (MWB) temperature since both values are needed to determine the sensible and latent (dehumidification) loads in the cooling mode.
- b. Wet-bulb temperature corresponding to 0.4%, 1.0% and 2.0% annual cumulative frequency of occurrence and the mean coincident dry-bulb temperature
- c. Dew-point temperature corresponding to 0.4%, 1.0% and 2.0% annual cumulative frequency of occurrence and the mean coincident dry-bulb temperature and humidity ratio (calculated for the dew-point temperature at the standard atmospheric pressure at the elevation of the station).
- d. Mean daily range (DR) of the dry bulb temperature, which is the mean of the temperature difference between daily maximum and minimum temperatures for the warmest month (highest average dry-bulb temperature). These are used to correct CLTD values.

In choosing the HVAC outdoor design conditions, it is neither economical nor practical to design equipment either for the annual hottest temperature or annual minimum temperature, since the peak or the lowest temperatures may occur only for a few hours over the span of several years. Economically speaking short duration peaks above the system capacity might be tolerated at significant reductions in first cost; this is a simple risk - benefit decision for each building design.

Therefore, as a practice, the 'design temperature and humidity' conditions are based on frequency of occurrence. The summer design conditions have been presented for annual percentile values of 0.4, 1 and 2% and winter month conditions are based on annual percentiles of 99.6 and 99%. The term "design condition" refers to the %age of time in a year (8760 hours), the values of dry-bulb, dew-point and wet-bulb temperature exceed by the indicated percentage. The 0.4%, 1.0%, 2.0% and 5.0% values are exceeded on average by 35, 88, 175 and 438 hours.

The 99% and 99.6% cold values are defined in the same way but are viewed as the values for which the corresponding weather element are less than the design condition 88 and 35 hours, respectively. 99.6% value suggests that the outdoor temperature is equal to or lower than design data 0.4% of the time.

Design condition is used to calculate maximum heat gain and maximum heat loss of the building. For comfort cooling, use of the 2.5% occurrence and for heating use of 99% values is recommended. The 2.5% design condition means that the outside summer temperature and coincident air moisture content will be exceeded only

2.5% of hours from June to September or 73 out of 2928 hours (of these summer months) i.e. 2.5% of the time in a year, the outdoor air temperature will be above the design condition.

Note, in energy use calculations, hour-by-hour outdoor climate data of a design day should be adopted instead of summer and winter design values.

7.2 Indoor Design Conditions and Thermal Comfort

The indoor design conditions are directly related to human comfort. Current comfort standards, ASHRAE Standard 55-1992 [4] and ISO Standard 7730 [5], specify a “comfort zone,” representing the optimal range and combinations of thermal factors (air temperature, radiant temperature, air velocity, humidity) and personal factors (clothing and activity level) with which at least 80% of the building occupants are expected to express satisfaction. The environmental factors that affect the thermal comfort of the occupants in an air-conditioned space are mainly:

- Metabolic rate, expressed in met ($1 \text{ met} = 18.46 \text{ Btu/hr.ft}^2$) determines the amount of heat that must be released from the human body and it depends mainly on the intensity of the physical activity.
- Indoor air temperature* (T_r) and *mean radiant temperature* (T_{rad}), both in °F. T_r affects both the sensible heat exchange and evaporative losses, and T_{rad} affects only sensible heat exchange.
- Relative humidity of the indoor air in %, which is the primary factor that influences evaporative heat loss.
- Air velocity of the indoor air in fpm, which affects the heat transfer coefficients and therefore the sensible heat exchange and evaporative loss.
- Clothing insulation* in clo ($1 \text{ clo} = 0.88 \text{ h.ft}^2 \cdot \text{°F/Btu}$), affects the sensible heat loss. Clothing insulation for occupants is typically 0.6 clo in summer and 0.8 to 1.2 clo in winter.

For comfort air-conditioning systems, according to ANSI/ASHRAE Standard 55-1992 and ASHRAE/IES Standard 90.1-1989, the following indoor design temperatures and air velocities apply for conditioned spaces where the occupant’s activity level is 1.2 met, indoor space relative humidity is 50% (in summer only), and $T_r = T_{rad}$:

	Clothing Insulation (clo)	Indoor Temperature (°F)	Air Velocity (fpm)
Winter	0.8 – 0.9	69 - 74	< 30
Summer	0.5 – 0.6	73 -79	< 50

If a suit jacket is the clothing during summer for occupants, the summer indoor design temperature should be dropped to 74 to 75°F.

The recommended indoor relative humidity, in %, is:

	Relative Humidity	
	Tolerable Range (%)	Preferred Range (%)
Winter		25 -30
Summer	30 - 65	40 - 55

The Psychrometric chapter of the Fundamentals Handbook (Chapter 6, 2001) provides more details on this aspect. The load calculations are usually based at 75°F dry bulb temperatures & 50% relative humidity.

7.3 Indoor Air Quality and Outdoor Air Requirements

According to the National Institute for Occupational Safety and Health (NIOSH), 1989, the causes of indoor air quality complaints in buildings are inadequate outdoor ventilation air. There are three basic means of improving indoor air quality: (1) eliminate or reduce the source of air pollution, (2) enhance the efficiency of air filtration, and (3) increase the ventilation (outdoor) air intake.

Abridged outdoor air requirements listed in ANSI/ASHRAE Standard 62-1989 are as follows:

Applications	Cfm/person
Offices, conference rooms, offices	20
Retail Stores	0.2 – 0.3 cfm/ft ²
Classrooms, theaters, auditoriums	15
Hospitals patient rooms	25

These ventilation requirements are based on the analysis of dilution of CO₂ as the representative human bio-effluent. As per ASHRAE standard 62-1999, comfort criteria with respect to human bio-effluents is likely to be satisfied, if the indoor carbon dioxide concentrations remain within 700 ppm above the outdoor air carbon dioxide concentration.

Refer to ANSI/ASHRAE Standard 62-1999 for details.

7.4 Building Pressurization

The outdoor air requirements are sometimes governed by the building pressurization needs. Most air-conditioning systems are designed to maintain a slightly higher pressure than the surroundings, a positive pressure, to prevent or reduce infiltration and untreated air entering the space directly. For laboratories, restrooms, or workshops where toxic, hazardous, or objectionable gases or contaminants are produced, a slightly lower pressure than the surroundings, a negative pressure, should be maintained to prevent or reduce the diffusion of these contaminants to the surrounding area.

For comfort air-conditioning systems, the recommended pressure differential between the indoor and outdoor air is 0.02 to 0.05 inch-WG. WG indicates the pressure at the bottom of a top-opened water column of specific inches of height; 1 in -WG = 0.03612 psig.

7.5 Building Characteristics

To calculate space heat gain, the following information on building envelope is required:

- a. Architectural plans, sections and elevations – for estimating building dimensions/area/volume
- b. Building orientation (N, S, E, W, NE, SE, SW, NW, etc), location etc
- c. External/Internal shading, ground reflectance etc.
- d. Materials of construction for external walls, roofs, windows, doors, internal walls, partitions, ceiling, insulating materials and thicknesses, external wall and roof colors - select and/or compute U-values for walls, roof, windows, doors, partitions, etc. Check if the structure is insulated and/or exposed to high wind.
- e. Amount of glass, type and shading on windows

7.6 Operating Schedules

Obtain the schedule of occupants, lighting, equipment, appliances, and processes that contribute to the internal loads and determine whether air conditioning equipment will be operated continuously or intermittently (such as, shut down during off periods, night set-back, and weekend shutdown). Gather the following information:

- Lighting requirements, types of lighting fixtures
- Appliances requirements such as computers, printers, fax machines, water coolers, refrigerators, microwave, miscellaneous electrical panels, cables etc
- Heat released by the HVAC equipment.
- Number of occupants, time of building occupancy and type of building occupancy

8.0 COOLING LOAD METHODOLOGY – CONSIDERATIONS & ASSUMPTIONS

Design cooling load takes into account all the loads experienced by a building under a specific set of assumed conditions. The assumptions behind design cooling load are as follows:

- a. Weather conditions are selected from a long-term statistical database. The conditions will not necessarily represent any actual year, but are representative of the location of the building. ASHRAE has tabulated such data.

- b. The solar loads on the building are assumed to be those that would occur on a clear day in the month chosen for the calculations.
- c. The building occupancy is assumed to be at full design capacity.
- d. The ventilation rates are either assumed on air changes or based on maximum occupancy expected.
- e. All building equipment and appliances are considered to be operating at a reasonably representative capacity.
- f. Lights and appliances are assumed to be operating as expected for a typical day of design occupancy.
- g. Latent as well as sensible loads are considered.
- h. Heat flow is analyzed assuming dynamic conditions, which means that heat storage in building envelope and interior materials is considered.
- i. The latent heat gain is assumed to become cooling load instantly, whereas the sensible heat gain is partially delayed depending on the characteristics of the conditioned space. According to the ASHRAE regulations, the sensible heat gain from people is assumed 30% convection (instant cooling load) and 70% radiative (delayed portion).
- j. **Peak load** calculations evaluate the maximum load to size and select the refrigeration equipment. The energy analysis program compares the total energy use in a certain period with various alternatives in order to determine the optimum one.
- k. **Space (zone) cooling load** is used to calculate the supply volume flow rate and to determine the size of the air system, ducts, terminals, and diffusers. The coil load is used to determine the size of the cooling coil and the refrigeration system. Space cooling load is a component of the cooling coil load.
- l. The heat transfer due to ventilation is not a load on the building but a load on the system.

8.1 Thermal Zoning

Thermal zoning is a method of designing and controlling the HVAC system so that occupied areas can be maintained at a different temperature than unoccupied areas using independent setback thermostats. A zone is defined as a space or group of spaces in a building having similar heating and cooling requirements throughout its occupied area so that comfort conditions may be controlled by a single thermostat.

When doing the cooling load calculations, always divide the building into zones. Always estimate the building peak load and individual zones airflow rate. The building peak load is used for sizing the refrigeration capacity and the individual zone loads are helpful in estimating the airflow rates (air-handling unit capacity).

In practice the corner rooms and the perimetric spaces of the building have variations in load as compared to the interior core areas. The following facts may be noted:

- a) The buildings are usually divided into two major zones.
 - Exterior Zone: The area inward from the outside wall (usually 12 to 18 feet, if rooms do not line the outside wall). The exterior zone is directly affected by outdoor conditions during summer and winter.
 - Interior Zone: The area contained by the external zone. The interior zone is only slightly affected by outdoor conditions and usually has a uniform cooling.
- b) Single-zone models shall be limited to open floor plans with perimeter walls not exceeding 40 feet in length.
- c) For large building footprints, assume a minimum of five zones per floor: one zone for each exposure (north, south, east & west) and an interior zone.

9.0 CLTD/SCL/CLF METHOD OF LOAD CALCULATION (ASHRAE FUNDAMENTALS 1997)

As mentioned before, the heat gain to the building is not converted to cooling load instantaneously. CLTD (cooling load temperature difference), SCL (solar cooling load factor), and CLF (cooling load factor): all include the effect of (1) time-lag in conductive heat gain through opaque exterior surfaces and (2) time delay by thermal storage in converting radiant heat gain to cooling load.

This approach allows cooling load to be calculated manually by use of simple multiplication factors.

- a. **CLTD** is a theoretical temperature difference that accounts for the combined effects of inside and outside air temp difference, daily temp range, solar radiation and heat storage in the construction assembly/building mass. It is affected by orientation, tilt, month, day, hour, latitude, etc. *CLTD factors are used for adjustment to conductive heat gains from walls, roof, floor and glass.*
- b. **CLF** accounts for the fact that all the radiant energy that enters the conditioned space at a particular time does not become a part of the cooling load instantly. The CLF values for various surfaces have been calculated as functions of solar time and orientation and are available in the form of tables in ASHRAE Handbooks. *CLF factors are used for adjustment to heat gains from internal loads such as lights, occupancy, power appliances.*
- c. **SCL** factors are used for adjustment to transmission heat gains from glass.

9.1 External Cooling Load

As discussed before, the total cooling load on a building consists of external as well as internal loads. The external loads consist of heat transfer by conduction through the building walls, roof, floor, doors etc, heat transfer by radiation through fenestration such as windows and skylights. All these are sensible heat transfers.

9.1.1 ROOF

The basic conduction equation for heat gain is $q = U A \Delta T$.

Where

- q = Heat gain in Btu/hr
- U = Thermal Transmittance for roof in Btu/hr.ft².°F
- A = area of roof in ft²
- ΔT = Temperature difference in °F

The heat gain is converted to cooling load using the room transfer functions (sol-air temperature) for the rooms with light, medium and heavy thermal characteristics. The equation is modified as

$$Q = U * A * (CLTD)$$

Where

- Q = cooling load, Btu/hr
- U = Coefficient of heat transfer roof or wall or glass, Btu/hr.ft².°F
- A = area of roof, ft²
- CLTD = cooling load temperature difference °F. The values are determined from tables available in chapter 28 of ASHRAE fundamentals handbook.

Since the ASHRAE tables provide hourly CLTD values for one typical set of conditions i.e. outdoor maximum temperature of 95°F with mean temperature of 85°F and daily range of 21°F, the equation is further adjusted to apply correction factors for conditions other than the mentioned base case. Thus,

$$Q_{\text{Roof}} = U * A * CLTD_{\text{Roof Corrected}}$$

The typical steps to calculate the Roof load are as follows:

- Step #1 Determine roof construction and overall heat transfer coefficient (U) (Chapter 28 ASHRAE 1997, Table A24-4, A29-5)
- Step #2 Select roof no. from ASHRAE Table 31 or Text table 7-34 which is closest to matching actual roof construction.
- Step #3 Select $CLTD_{\text{Roof}}$ for time of interest, typically on an hourly basis (Chapter 28 ASHRAE Table A28-32, A28-34).
- Step #4 Corrections: Values on tables are for
- a. Four latitudes on July or August

- b. Indoor temperature of 78°F
- c. Outdoor maximum temperature of 95°F with mean daily temperature of 85°F and daily range of 21°F

$$CLTD_{ROOF\ Corrected} = [CLTD_{Roof} + (78 - TR) + (TM - 85)]$$

Where

- $(78 - TR)$ = indoor design temperature correction
- $(TM - 85)$ = outdoor design temperature correction
- TR = Indoor room temperature
- Tm = Mean outdoor temperature
- Tmax = Maximum outdoor temperature
- $Tm = Tmax - (Daily\ Range) / 2$

Step # 5 Calculate roof area (A) from architectural plans

Step #6 $Q_{Roof} = U * A * CLTD_{Roof\ Corrected}$

What is Sol-air temperature?

The *sol-air temperature* is a fictitious number that defines the value of the outside air temperature which would, in the absence of all radiation exchanges, give the same rate of heat flow (Q) into the outer surface of the wall as the actual combination of temperature difference and radiation exchanges.

The sol-air temperature represents the equivalent outdoor design air temperature that combines convection to the outdoor air, radiation to the ground and sky, and solar radiation heat transfer effects on the outer surface of a building. It takes into account the mass of the wall or roof, the daily temperature range (which affects the heat storage), the color of the outside surface (which affects solar heat absorption rate) and the latitude and month. When the sol-air data is combined with the inside design temperature, a cooling load temperature difference (CLTD) is obtained.

Hourly sol-air temperature values are presented in tables for various orientations of a surface for 21 July at 40° N latitude, with standard surface absorption factor for light and dark colored surfaces in ASHRAE handbook of fundamentals. The sol-air temperatures are listed for a given air temperature cycle.

Adjustments can be introduced for other dates, latitudes and air temperature cycles. When this method is used in conjunction with custom tables generated by appropriate computer software, and for buildings where external shading is not significant, it can be expected that it will produce results very close to that produced by the TFM.

For a complete discussion of this concept, refer to the ASHRAE Handbook of fundamentals.

9.1.2 WALLS

The cooling load from walls is treated in a similar way as roof:

$$Q_{\text{Wall}} = U * A * \text{CLTD}_{\text{Wall Corrected}}$$

Where

- Q_{Wall} = Load through the walls in Btu/hr
- U = Thermal Transmittance for walls in Btu/ (h ft² F)
- A = area of walls in ft²
- CLTD = Cooling Load Temperature Difference for walls in °F

The typical steps to calculate the Wall load are as follows:

Step #1 Determine wall construction and overall heat transfer coefficient (U) (Chapter 24 ASHRAE 1997, Table A24-4, A29-5)

Step #2 Select wall type from ASHRAE Table 33 which is closest to matching actual wall construction. Pay attention to effect of mass distribution (inside insulation, outside insulation or evenly distributed).

Step #3 Select $\text{CLTD}_{\text{Wall}}$ for time of interest, typically on an hourly basis (Chapter 28 ASHRAE Table A28-32, A28-34).

Step #4 Corrections: Values on tables are for

- a. Four latitudes on July or August
- b. Indoor temperature of 78°F
- c. Outdoor maximum temperature of 95°F with mean daily temperature of 85°F and daily range of 21°F

$$\text{CLTD}_{\text{Wall Corrected}} = [\text{CLTD}_{\text{Wall}} + (78 - \text{TR}) + (\text{TM} - 85)]$$

Where

- $(78 - \text{TR})$ = indoor design temperature correction
- $(\text{TM} - 85)$ = outdoor design temperature correction
- TR = Indoor room temperature

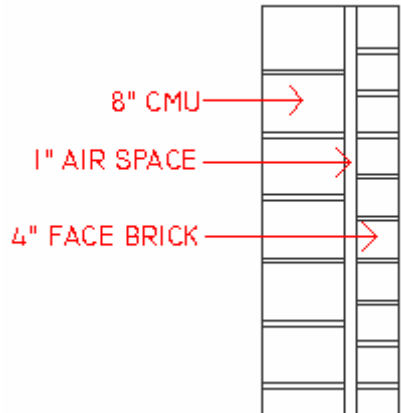
- T_m = Mean outdoor temperature
- T_{max} = Maximum outdoor temperature
- T_m = T_{max} – (Daily Range) / 2

Step #5 Calculate walls area (A). Use the architectural drawings to determine how much wall area you have. Since you will be finding window and door losses separately, exclude windows and door and count *only* wall area.

Step #6 $Q_{Wall} = U * A * CLTD_{Wall\ Corrected}$

Example:

The wall has a 4" brick exterior, a 1" air gap, and 8" concrete block wall. Calculate the overall U value of the walls (Btu/hr/ft²/°F) based on the materials used and the standards set forth in ASHRAE. The tabulated values are as follows:



Construction Material	R-Value (hr x ft ² x °F/ Btu)	U-Value (Btu/hr/ft ² /°F)
Outside Air Resistance (R ₁)	0.33	0.228
4" Face Brick (R ₂)	0.43	
1" Air Gap (R ₃)	0.91	
8" CMU (R ₄)	2.02	
Inside Air Resistance (R ₅)	0.69	
R _{Total}	4.38	

$$R_{Total} = R_1 + R_2 + \dots$$

$$U = \frac{1}{R_1 + R_2 + R_3 + \dots + R_n}$$

Two ideas are important here. The first is the idea of **thermal resistance**. Heat energy flows from hot spaces to cold spaces and it increases as the temperature difference increases. The material which separates the temperature extremes has a certain resistance to energy flow. When the resistance is high, the rate at which energy flows through the material is low; when the resistance is low, high energy flow rates are expected. We insulate walls to lower the rate at which energy flows through the walls. Insulation increases the thermal resistance or “R value” of the wall.

The second important idea is the idea of the **heat capacity** of the building materials. Heat capacity is a measure of a material’s ability to store heat energy. Metals tend to have low heat capacities. When heat energy flows through a metal, it changes temperature quickly. On the other hand, stone or cement has a much higher heat capacity. When heat energy flow into stone, it changes temperature very slowly and tends to “store” the heat energy. Passive solar homes usually include a large mass of stone, rock, or other material with a high heat capacity. This thermal mass will heat up during the day when the sun shines. At night when it is cooler, energy can be drawn from the thermal mass. It is interesting to note that water has a very high heat capacity. Open water inside a house might create humidity problems, but in our atmosphere, the heat capacity of water has a significant effect on moderating temperature and on the weather pattern in general.

9.1.3 SOLAR LOAD THROUGH GLASS

Solar load through glass has two components: 1) Conductive and 2) Solar Transmission

The absorbed and then conductive portion of the radiation through the windows is treated like the roof & walls where CLTD values for standard glazing are tabulated in ASHARE fundamentals handbook.

For solar transmission, the cooling load is calculated by the cooling load SCL factor and shading coefficient (SC).

The cooling load equations for glass are:

Conductive $Q_{\text{Glass Conductive}} = U * A * \text{CLTD}_{\text{Glass Corrected}}$

Solar Transmission $Q_{\text{Glass Solar}} = A * \text{SC} * \text{SCL}$

Where

- $Q_{\text{Conductive}}$ = Conductive load through the glass in Btu/hr
- Q_{Solar} = Solar transmission load through the glass in Btu/hr
- U = Thermal Transmittance for glass in Btu/ (h ft² F)
- A = area of glass in ft²
- CLTD = Cooling Load Temperature Difference for glass in °F

- SC = Shading coefficient
- SCL = Solar Cooling Load Factor

STEPS FOR CONDUCTIVE CALCULATIONS

Step # 1 For the glass types used, select from ASHRAE tables the overall heat transfer coefficient (U). The glass is the major contributor of heat gain in the commercial buildings. Pay attention to effect of shading, reflective films, curtains, drapes etc (refer to example below)

Step # 2 Select CLTD_{Glass} for time of interest, typically on an hourly basis (Chapter 28 ASHRAE Table 34).

Step # 3 Corrections:

$$\text{CLTD}_{\text{Glass Corrected}} = [\text{CLTD}_{\text{Glass}} + (78 - \text{TR}) + (\text{TM} - 85)]$$

Where

- $(78 - \text{TR})$ = indoor design temperature correction
- $(\text{TM} - 85)$ = outdoor design temperature correction
- TR = Indoor room temperature
- Tm = Mean outdoor temperature
- Tmax = Maximum outdoor temperature
- $\text{Tm} = \text{Tmax} - (\text{Daily Range}) / 2$

Step # 4 Calculate glass area (A) from architectural plans

Step # 5 $Q_{\text{Glass}} = U * A * \text{CLTD}_{\text{Glass Corrected}}$

STEPS FOR SOLAR TRANSMISSION CALCULATIONS

Step # 1 Determine shading coefficient (SC) from ASHRAE 1997 Chapter 27, Table 11

Step # 2 Determine zone type from ASHRAE 1997 Chapter 29, Table 35 B

Step # 3 Determine solar cooling load factor (SCL) from ASHRAE 1997 Chapter 28, Table A28-36

Step # 4 Calculate glass area (A) from architectural plans

Step # 5 $Q_{\text{Glass Solar}} = A * \text{SC} * \text{SCL}$

Example:

Estimating U value and compare cooling load of different glass types/with or without drapes

U values can also be estimated, if thermal resistance (R) or thermal conductivity of glass material is known. U value is inverse of R-value.

The R-value is calculated as $R = l/k$ where l is the thickness of the material and k is the thermal conductivity.

The thermal resistances for the window with and without drapes are:

$$\text{Without drape: } R_{Total} = R_{airfilm} + R_{window} + R_{airfilm}$$

$$\text{With drapes: } R_{Total} = R_{airfilm} + R_{drapes} + R_{gap} + R_{window} + R_{airfilm}$$

Assumed data

- Thickness of glass = 1/8 inch
- Curtain-window spacing = 3.5 inch
- Conductivity of glass = 0.81 BTU/(hr ft °F)
- Conductivity of drapes = 0.035 BTU/(hr ft °F)
- Thickness of drapes = 1/16 inch
- Thickness of insulated drapes = 1/2 inch

R - Values from ASHRAE Fundamentals (2001):

Air Space/Gap	R-value (hr ft ² °F/BTU)
Inside vertical air film, $R_{airfilm}$	0.68
Uninsulated drapes, R_{drapes}	0.15
Insulated drapes, R_{drapes}	1.2
3.5 inch vertical air gap, R_{gap}	1.1
Single glazed window (wood or vinyl frame), R_{window}	1.2
Double glazed window (wood or vinyl frame), R_{window}	2.0
Outside vertical air film, $R_{airfilm}$	0.25

For calculating U-values:

$$R_{series} = R1 + R2 + R3 + \text{etc}$$

$$U_{series} = 1/ R_{series}$$

Using the equation above the different type of window/drapes will yield following results:

Window/ Drapes	Governing Equation	Overall R-value (hr ft ² °F/BTU)
Single-paned window No drapes	$R_{total} = R_{airfilm} + R_{window} + R_{airfilm}$	$R_{total} = 0.68 + 1.2 + 0.25 = 2.13$
Double-paned window No drapes	$R_{total} = R_{airfilm} + R_{window} + R_{airfilm}$	$R_{total} = 0.68 + 2 + 0.25 = 2.93$
Single-paned window Regular drapes	$R_{total} = R_{airfilm} + R_{drapes} + R_{gap} + R_{window} + R_{airfilm}$	$R_{total} = 0.68 + 0.15 + 1.1 + 1.2 + 0.25 = 3.38$
Double-paned window Regular drapes	$R_{total} = R_{airfilm} + R_{drapes} + R_{gap} + R_{window} + R_{airfilm}$	$R_{total} = 0.68 + 0.15 + 1.1 + 2 + 0.25 = 4.18$
Single-paned window Insulated drapes	$R_{total} = R_{airfilm} + R_{drapes} + R_{gap} + R_{window} + R_{airfilm}$	$R_{total} = 0.68 + 1.2 + 1.1 + 1.2 + 0.25 = 4.43$
Double-paned window Insulated drapes	$R_{total} = R_{airfilm} + R_{drapes} + R_{gap} + R_{window} + R_{airfilm}$	$R_{total} = 0.68 + 1.2 + 1.1 + 2 + 0.25 = 5.23$

For single glazed windows, adding drapes reduces heat loss by 37%. Adding the same drapes to double glazed windows reduces heat loss by 30%. Adding insulated drapes to a single glazed window reduces heat loss by 56% (48% for double glazed windows).

So, for single and double glazed windows, both claims are approximately true. The reduction in heat loss is smaller for double glazed windows than it is for single glazed windows.

9.1.4 PARTITIONS, CEILINGS AND FLOORS

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

$$Q = U A (T_a - T_{rc})$$

Where

- U = coefficient of overall heat transfer between adjacent and conditioned space in Btu/ (h ft² F) See 1997 ASHRAE Fundamentals, Chapter 24 or 2001 ASHRAE Fundamentals, and Chapter 25.
- A = area of partition in ft², ceiling or floor calculated from building plans
- T_a = Temperature of adjacent space in °F (Note: If adjacent space is not conditioned and temperature is not available, use outdoor air temperature less 5°F)
- T_{rc} = Inside design temperature of conditioned space in °F (assumed constant)

Note that the temperature T_a may range widely from that in the conditioned space. The temperature in a kitchen or boiler room, for example, may be as much as 15 to 50°F above the outdoor air temperature. Actual temperatures in adjoining spaces should be measured when possible. Where nothing is known, assume outdoor air temperature less 5°F.

9.2 Internal Cooling Loads

The various internal loads consist of sensible and latent heat transfers due to occupants, products, processes appliances and lighting. The lighting load is only sensible. The conversion of sensible heat gain (from lighting, people, appliances, etc.) to space cooling load is affected by the thermal storage characteristics of that space and is thus subject to appropriate cooling load factors (CLF) to account for the time lag of the cooling load caused by the building mass. The weighting factors equation determines the CLF factors.

$$\text{CLF} = Q_{\text{cooling load}} / Q_{\text{internal gains}}$$

Note that the latent heat gains are considered instantaneous.

9.2.1 People

$$Q_{\text{sensible}} = N (Q_S) (\text{CLF})$$

$$Q_{\text{latent}} = N (Q_L)$$

Where

- N = number of people in space from ASHRAE, Table A28-3
- Q_S, Q_L = Sensible and Latent heat gain from occupancy is given in 1997 ASHRAE Fundamentals Chapter 28, Table 3)
- CLF = Cooling Load Factor, by hour of occupancy. See 1997 ASHRAE Fundamentals, Chapter 28, Table 37

Note: CLF = 1.0, if operation is 24 hours or of cooling is off at night or during weekends.

The table below gives representative rates at which heat and moisture are given off by human beings in different states of activity. Often these sensible and latent heat gains constitute a large fraction of the total load. Even for short-term occupancy, the extra heat and moisture brought in by people may be significant.

Snapshot of Occupancy Heat gain values in ASHRAE

Heat Gain from Occupants at Various Activities (At Indoor Air Temperature of 78°F)

Activity	Total heat, Btu/h		Sensible heat, Btu/h	Latent heat, Btu/h
	Adult, male	Adjusted		
Seated at rest	400	350	210	140
Seated, very light work, writing	480	420	230	190
Seated, eating	520	580	255	325
Seated, light work, typing,	640	510	255	255
Standing, light work or walking	800	640	315	325
slowly,	880	780	345	435
Light bench work	1040	1040	345	695
Light machine work, walking 3mi/hr	1360	1280	405	875
Moderate dancing	1600	1600	565	1035
Heavy work, lifting	2000	1800	635	1165
Athletics				

These values are for 78°F room dry bulb temperature. For 80°F dry bulb temperature, the total heat remains the same, but the sensible heat value should be decreased by approximately 8% and the latent heat values increased accordingly.

Occupants generate both sensible and latent heat components according to activity level. The sensible heat rate increases slightly with higher activity but latent heat increases dramatically because of greater perspiration rates needed to maintain body temperature.

9.2.2 Lights

The primary source of heat from lighting comes from light-emitting elements. Calculation of this load component is not straightforward; the rate of heat gain at any given moment can be quite different from the heat equivalent of power supplied instantaneously to those lights. Only part of the energy from lights is in the form of convective heat, which is picked up instantaneously by the air-conditioning apparatus. The remaining portion is in the form of radiation, which affects the conditioned space only after having been absorbed and re-released by walls, floors, furniture, etc. This absorbed energy contributes to space cooling load only after a time lag, with some part of such energy still present and reradiating after the lights have been switched off.

Generally, the instantaneous rate of heat gain from electric lighting may be calculated from:

$$Q = 3.41 \times W \times F_{UT} \times F_{SA}$$

Cooling load factors are used to convert instantaneous heat gain from lighting to the sensible cooling load; thus the equation is modified to

$$Q = 3.41 \times W \times F_{UT} \times F_{SA} \times (CLF)$$

Where

- W = Watts input from electrical lighting plan or lighting load data
- F_{UT} = Lighting use factor, as appropriate
- F_{SA} = special ballast allowance factor, as appropriate

- CLF = Cooling Load Factor, by hour of occupancy. See 1997 ASHRAE Fundamentals, Chapter 28, Table 38

Note: CLF = 1.0, if operation is 24 hours or if cooling is off at night or during weekends.

The total light wattage is obtained from the ratings of all lamps installed, both for general illumination and for display use.

The special allowance is the ratio of the wattage in use for the conditions under which the load estimate is being made, to the total installed wattage. For commercial applications such as stores, the use factor would generally be unity. The special allowance factor is for fluorescent fixtures and/or fixtures that are either ventilated or installed so that only part of their heat goes to the conditioned space. For fluorescent fixtures, the special allowance factor accounts for the ballast losses, and can be as high as 2.19 for 32 W single lamp high-output fixtures on 277 V circuits. Rapid start, 40 W lamp fixtures have special allowance factors that vary from a low of 1.18 for two lamps at 277 V to a high of 1.30 for one lamp at 118V, with a recommended value of 1.20 for general applications. Industrial fixtures other than fluorescent, such as sodium lamps, may have special allowance factors varying from 1.04 to 1.37, depending on the manufacturer, and should be dealt with individually.

For ventilated or recessed light fixtures, manufacturers' or other data must be sought to establish the fraction of the total wattage that may be expected to enter the conditioned space directly (and subject to time lag effect), versus that which must be picked up by return air or in some other appropriate manner.

9.2.3 Power Loads

The industrial and commercial applications use various equipments such as fans, pumps, machine tools, elevators, escalators and other machinery, which add significantly to the heat gain. There are 5 equations in use for different scenarios.

Case #1

If the motor and the machine are in the room the heat transferred can be calculated as

$$Q = 2545 * (P / \text{Eff}) * F_{UM} * F_{LM}$$

P = Horsepower rating from electrical power plans or manufacturer's data

Eff = Equipment motor efficiency, as decimal fraction

F_{UM} = Motor use factor (normally = 1.0)

F_{LM} = Motor load factor (normally = 1.0)

Note: F_{UM} = 1.0, if operation is 24 hours

In this situation the total power are transferred as heat to the room.

Note! If the machine is a pump or a fan most of the power are transferred as energy to the medium and may be transported out of the room. For such cases, case 4 shall be used.

Case #2

If the motor is outside and the machine is in the room the heat transferred can be calculated as

$$Q = 2545 * P * F_{UM} * F_{LM}$$

P = Horsepower rating from electrical power plans or manufacturer's data

F_{UM} = Motor use factor

F_{LM} = Motor load factor

Note: $F_{UM} = 1.0$, if operation is 24 hours

Case #3

If the motor is belt driven and the motor and belt is outside and the machine is in the room the heat transferred can be calculated as

$$Q = 2545 * P * \text{Belt Eff} * F_{UM} * F_{LM}$$

P = Horsepower rating from electrical power plans or manufacturer's data

Belt Eff = Belt transmission efficiency, as decimal fraction

F_{UM} = Motor use factor

F_{LM} = Motor load factor

Note: $F_{UM} = 1.0$, if operation is 24 hours

Case #4

If the motor is in the room and the machine is outside the heat transferred can be calculated as

$$Q = 2545 * [P/\text{Eff} - P] * F_{UM} * F_{LM}$$

P = Horsepower rating from electrical power plans or manufacturer's data

Eff = Motor efficiency, as decimal fraction

F_{UM} = Motor use factor

F_{LM} = Motor load factor

Note: $F_{UM} = 1.0$, if operation is 24 hours

Case #5

If the motor is belt driven and the motor and belt is in the room and the machine is outside the heat transferred can be calculated as

$$Q = 2545 * [P / (\text{motor Eff}) - P / (\text{belt Eff})] * F_{UM} * F_{LM}$$

P = Horsepower rating from electrical power plans or manufacturer's data

F_{UM} = Motor use factor

F_{LM} = Motor load factor

The highest quantum of heat gain shall be from the case #1, when both the motor and driven equipment are located inside the space. The physical location of equipment is not the only governing criteria on using a particular case. An exception is taken for pumps and fans to use case# 4 equation in lieu case#1, even if both the driver & driven equipment is located inside the space, provided the fluid is pumped outside to the conditioned space. This is due to the fact that the heat energy is carried away by the fluid. It is important to use the discretion judiciously, for example for the direct driven exhaust fans, the case #3 is applicable while for the supply or intake ventilation fans, the case #1 is applicable.

Note that the heat output of a motor is generally proportional to the motor load, within the overload limits. Because of typically high no-load motor current, fixed losses, and other reasons, FLM is generally assumed to be unity, and no adjustment should be made for under-loading or overloading unless the situation is fixed, can be accurately established, and the reduced load efficiency data can be obtained from the motor manufacturer.

9.2.4 Appliances

In a cooling load estimate, heat gain from all appliances-electrical, gas, or steam-should be taken into account. Because of the variety of appliances, applications, schedules, use, and installations, estimates can be very subjective. Often, the only information available about heat gain from equipment is that on its name-plate.

$$Q_{\text{Sensible}} = Q_{in} \times F_u \times F_r \times (\text{CLF})$$

$$Q_{\text{Latent}} = Q_{in} \times F_u$$

Where

- Q_{in} = rated energy input from appliances. See 1997 ASHRAE Fundamentals, Chapter 28, and Table 5 thru 9 or use manufacturer's data. For computers, monitors, printers and miscellaneous office equipment, see 2001 ASHRAE Fundamentals, Chapter 29, Tables 8, 9, & 10. [If the appliance nameplate provides the power ratings in Watts, Q_{in} can be estimated as $3.14 \times \text{Wattage of equipment}$]
- F_u = Usage factor. See 1997 ASHRAE Fundamentals, Chapter 28, Table 6 and 7
- F_r = Radiation factor. See 1997 ASHRAE Fundamentals, Chapter 28, Table 6 and 7
- CLF = Cooling Load Factor, by hour of occupancy. See 1997 ASHRAE Fundamentals, Chapter 28, Table 37 and 39

Note 1: CLF = 1.0, if operation is 24 hours or of cooling is off at night or during weekends.

Note 2: Set latent load = 0 if appliance under exhaust hood.

9.2.5 Infiltration Air

$$Q_{\text{sensible}} = 1.08 \times \text{CFM} \times (T_o - T_i)$$

$$Q_{\text{latent}} = 4840 \times \text{CFM} \times (W_o - W_i)$$

$$Q_{\text{total}} = 4.5 \times \text{CFM} \times (h_o - h_i)$$

Where

- CFM = Infiltration air flow rate. See 1997 ASHRAE Fundamentals, Chapter 25, for determining infiltration
- T_o, T_i = Outside/Inside dry bulb temperature, °F
- W_o, W_i = Outside/Inside humidity ratio, lb water/lb dry air
- h_o, h_i = Outside/Inside air enthalpy, Btu per lb (dry air)

9.3 Heat Gain from Miscellaneous Sources

9.3.1 Supply Fan Heat Load

Supply and/or return fans that circulate or supply air to the space add heat to the space or system depending on the location relative to the conditioned space. The heat added may take one or all of the following forms:

- Instantaneous temperature rise in the air stream due to fan drive inefficiency.
- Temperature rise in the air stream when the air is brought to static equilibrium and the static and kinetic energy is transformed into heat energy.

The location of the fan and motor relative to the cooling coil and space being conditioned determines how the heat is added to the system. If the fan is downstream of the cooling coil (draw-thru) then the fan heat load is added to the space cooling load. If the fan is upstream of the cooling coil, then the fan heat load is added to the system cooling coil load.

The heat energy is calculated as follows:

$$Q = 2545 \times [P / (\text{Eff}_1 \times \text{Eff}_2)]$$

Where

- P = Horsepower rating from electrical power plans or manufacturer's data
- 2545 = conversion factor for converting horsepower to Btu per hour
- Eff_1 = Full load motor and drive efficiency
- Eff_2 = Fan static efficiency

Note: See 1997 ASHRAE Fundamentals, Chapter 28, and Table 4 for motor heat gain.

9.3.2 Ventilation Air

Ventilation air is the amount of outdoor air required to makeup for air leaving the space due to equipment exhaust, exfiltration and/or as required to maintain Indoor Air Quality for the occupants. (See ASHRAE Standard 62 for minimum ventilation requirements). The heat is usually added to the air stream before the cooling coil and has no direct impact on the space conditions. The additional cooling coil load is calculated as follows:

$$Q_{\text{sensible}} = 1.08 \times \text{CFM} \times (T_o - T_c)$$

$$Q_{\text{latent}} = 4840 \times \text{CFM} \times (W_o - W_c)$$

$$Q_{\text{total}} = 4.5 \times \text{CFM} \times (h_o - h_c)$$

Where

- CFM = Ventilation airflow rate.
- T_o = Outside dry bulb temperature, °F
- T_c = Dry bulb temperature of air leaving the cooling coil, °F
- W_o = Outside humidity ratio, lb (water) per lb (dry air)
- W_c = Humidity ratio of air leaving the cooling coil, lb (water) per lb (dry air)
- h_o = Outside/Inside air enthalpy, Btu per lb (dry air)
- h_c = Enthalpy of air leaving the cooling coil Btu per lb (dry air)

9.3.3 Duct Heat Gain

Unless the return ductwork system is extensive and uninsulated or passes over a non-conditioned space, only the heat gained by the duct supply system is significant. This heat gain is normally estimated as a

percentage of the space sensible cooling load (usually 1% to 5%) and applied to the temperature of the air leaving the cooling coil in the form of temperature increase.

9.3.4 Duct Leakage

Air leakage out of or into ductwork can have much greater impact than the duct heat gain or loss. Outward leakage from a supply duct is a direct loss of cooling and/or dehumidifying capacity and must be offset by increased air flow or reduced supply air temperature. Leakage into a return duct system causes additional cooling coil capacity but it does not directly affect the space conditions.

Commercial type or existing older systems can have leakage from **10% to 20%** of the total system airflow. Per energy conservation guidelines, in a new installation, the duct system should not leak more than **1% to 3%** of total system airflow.

The engineer or designer is cautioned to make sure that the proper allowance for leakages is included in the calculations in order to ensure that the equipment selected is properly sized.

9.3.5 Diversity Factors

Diversity of cooling load results from not using part of the load on a design day. Therefore, diversity factors are factors of usage and are applied to the refrigeration capacity of large air conditioning systems. These factors vary with location, type and size of application. Generally, diversity factors can be applied to loads from people and lights as there is neither 100% occupancy nor total lighting at the time of such other peak loads as peak solar and transmission loads. The reduction in cooling loads from non-use is real and should be accounted for. Listed are some average typical diversity factors for large buildings during occupied periods.

Type of Application	Diversity Factor	
	Lights	People
Office	0.70 to 0.85	0.75 to 0.90
Apartment, Hotel	0.30 to 0.50	0.40 to 0.60
Department Store	0.90 to 1.00	0.80 to 0.90
Industrial	0.80 to 0.90	0.85 to 0.95

In the case of Industrial, diversity should also be applied to the machinery load.

10.0 SUPPLY AIR CALCULATIONS

Calculations for the design supply air quantities are dependent on the type of system used. Some generalizations and assumptions need to be made to assist in the use of the cooling load calculations for equipment selection and sizing. For constant air volumes with reheat type of system, the design supply air quantities required are based on the peak requirements for each space. However, if the system selected allows for air flow to each zone to vary based on load, the peak load on an air conditioning unit serving several zones or spaces is not equal to the sum of the peak loads of each zone, but will be that amount necessary to handle the maximum coincident load on the system at a given time. Determining the maximum coincident load may require additional calculation and analysis.

Supply air flow rate to a space is based only on the total space sensible heat load, thus

$$\text{CFM} = Q_{\text{sensible}} / [1.08 \times (T_R - T_s)]$$

Where

- CFM = air flow in cubic feet per minute
- 1.08 = conversion constant = 0.244 X (60/13.5)
 - 0.244 = specific heat of moist air, Btu/lb d.a.
 - 13.5 = specific volume of moist air, cu-ft. per lb. d.a. (@70°F, 50% RH)
- Q_{sensible} = total room sensible heat gain, BTU per hr.
- T_R = Room dry bulb temperature, °F
- T_s = Room supply air dry bulb temperature, °F (not necessarily the same as the temperature of the air leaving the cooling coil).

Refer to PDH engineer Course No: M-6009 “Application of Psychrometrics in Heating and Cooling” for further details on the subject. This course gives an example that will describe “How to estimate the airflow rate for a given cooling load”?

11.0 EXAMPLE

11.1 Cooling Load Example in US units

An office building in Montreal area requires air conditioning for a single story building. The following data is gathered:

Location: Montreal

Type of building: Office

Number of floors:	1
Floor area:	64 ft x 80 ft = 5120 ft ²
Floor-to-floor Height:	15 ft
Window area:	20 % of the wall area
Windows:	Double glazed
Wall:	Wall number 13 [U= 0.2 Btu/h.ft ² .F, as per table 33A*, pg 28.46
Roof:	Roof number 13 [U= 0.2 Btu/h.ft ² .F as per table 31, pg 28.42]
Windows:	U= 0.55 Btu/h.ft ² .F
Recommended ventilation:	½ air change per hour (min) or 20 cfm/person
Occupancy:	7 persons per 1000 sq-ft
Working:	8 hrs of working - 9.00 to 17.00 hrs

Estimate the cooling load using the Cooling Load Temperature Difference / Solar Cooling Load/ Cooling Load Factor (CLTD/SCL/CLF) method.

DESIGN CONDITIONS:

Outdoor design conditions for Montreal area [Reference: 1997 ASHARE Handbook - Fundamentals Table-2, pp.26.25]

Location:	(45° N 73° 45'W) ----- [for simplicity consider 40° N latitude]
Outdoor design dry-bulb:	90 °F
Outdoor design wet-bulb:	75 °F
Indoor design Dry-bulb:	78 °F
Daily Range:	20 °F
Relative humidity:	50%
Wind velocity:	7½ mph

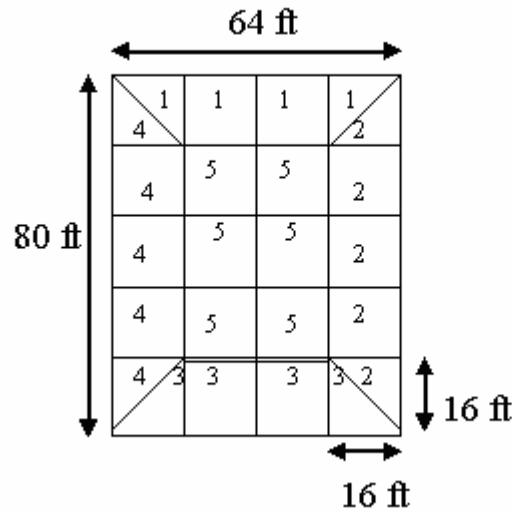
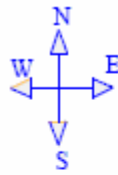
DESIGN BASIS

Zone Division:

Due to solar exposure and sun movement, the heat gain through exterior walls (in perimetric areas) will be much higher than the interior zones. As such the supply air requirements for exterior zones shall be much higher than

the interior zones. This aspect is very important in sizing and control of air-conditioning equipment especially when single equipment (AHU) is feeding the entire area.

For this job, the building envelope is divided into 5 zones having 20 rooms of equal areas (It's assumed the two corners in each zone as a one room with the same situation with the other rooms on that zone). The central zone heat gain conduction through the walls are zero due to $\Delta T=0$ (There isn't exterior walls in central zone)



- Zone -1 : North
- Zone -2 : East
- Zone -3 : South
- Zone -4 : West
- Zone -5 : Central

Zone Description:

Zone Number	Number of Walls per room	Wall Side	Number of Room	Area of wall (ft ²)	Area of Window (ft ²)	Area of Roof (ft ²)
1	1	N	3	192	48	256
2	1	E	4	192	48	256
3	1	S	3	192	48	256
4	1	W	4	192	48	256
5	-	Interior	6	-	48	256

Roof area each room is 256 ft² (16 * 16)

Window area for each wall is 48 ft² which is 20 % of gross wall area of 240 sq-ft (16 * 15 * 0.2=48 ft²)

Net area for each wall is 192 sq. ft, which is gross wall area minus window area ($240 \text{ ft}^2 - 48 \text{ ft}^2 = 192 \text{ ft}^2$)

ASSUMPTIONS

The following assumptions were made during the course of calculation:

- i. Occupancy hours for people from 0900 to 1700 hrs
- ii. 2 persons per room
- iii. Light remains on from 0900 to 1700 hrs.
- iv. 1536 watts of lights per room
- v. 200 watts for computer per room
- vi. Roof number is 13 as per Table 31t page-28.42 and U value for roof is $0.055 \text{ Btu/h.ft}^2 \cdot ^\circ\text{F}$.
- vii. Wall number is 13 as per Table 33A, page-28.46 and U value for wall is $0.2 \text{ Btu/h.ft}^2 \cdot ^\circ\text{F}$.
- viii. Window is un-coated double glazed and U value is 0.55 which is found from Table 11, page-29.8
- ix. Values of Cooling Load Temperature Difference (CLTD) for roof are taken from Table30, Page-28.42
- x. Values of Cooling Load Temperature Difference (CLTD) for wall are taken from Table32, Page-28.45
- xi. Values of Cooling Load Temperature Difference (CLTD) for glass are taken from Table34, Page-28.49
- xii. Values of Shading Coefficient (SC) are taken from Table 11, page-29.25
- xiii. Values of Solar Cooling Load (SCL) are taken from Table 36, page-28.50, zone Type C
- xiv. Values of Cooling Load Factor (CLF) for lighting are taken from Table38 based on lights on for 8 hours, Page-28.52, zone Type C
- xv. Values of Cooling Load Factor (CLF) for people are taken from Table37 based on 8 hours in space, Page-28.51, zone Type C
- xvi. Rates of Sensible Heat Gain (SHG) and Latent Heat Gain (LHG) from people are taken from Table 3, page-28.8
- xvii. It is assumed that there will be one Computer in each room also assumed that heat gain will be 680 Btu/hr

*Source: ASHRAE Handbook- 1997 Fundamentals

CALCULATIONS

Calculation of CLTD Correction

$$CLTD_c = [CLTD + (78 - TR) + (TM - 85)]$$

TR = inside design temperature = 78°F

TM = mean outdoor temperature = Outdoor design dry bulb temperature - (Daily range / 2)

$$TM = 90 - (20 / 2) = 90 - 10 = 82$$

$$\text{Therefore } CLTD_c = CLTD + (82 - 85) = CLTD - 5$$

ROOF

Calculation of Heat Load due to Roof (per room):

Time of day	CLTD	CLTD _c	*Q(Btu/hr)
9	16	11	155
10	17	12	169
11	20	15	211
12	24	19	268
13	28	23	324
14	33	28	394
15	38	33	465
16	42	37	521
17	46	41	577

Equations

$$Q = U * A * CLTD_c$$

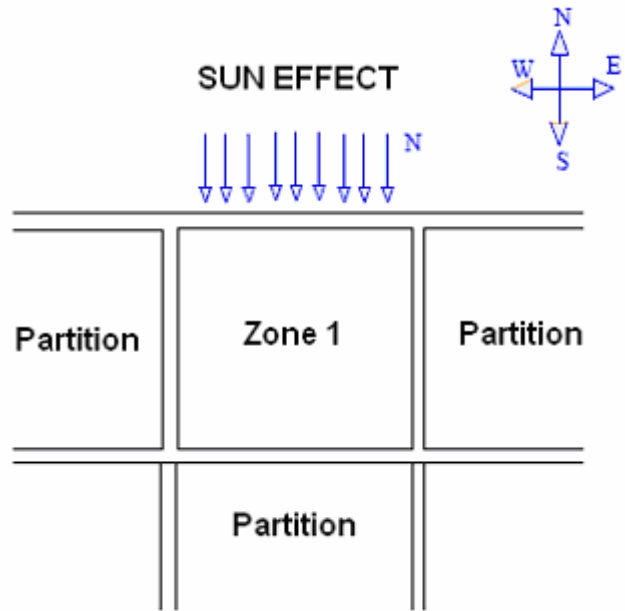
$$U = 0.2 \text{ Btu/h.ft}^2.\text{F}$$

$$A = 256 \text{ ft}^2 (16' * 16')$$

WALLS

Calculation of Heat Load due to Conduction through the walls for Zone-1:

Zone-1(per room)			
Time of day	Side Facing		Q(Btu/hr)
	North CLTD	CLTD _c	
9	9	4	154
10	9	4	154
11	9	4	154
12	10	5	192
13	10	5	192
14	11	6	230
15	12	7	269
16	14	9	346
17	15	10	384



Equations

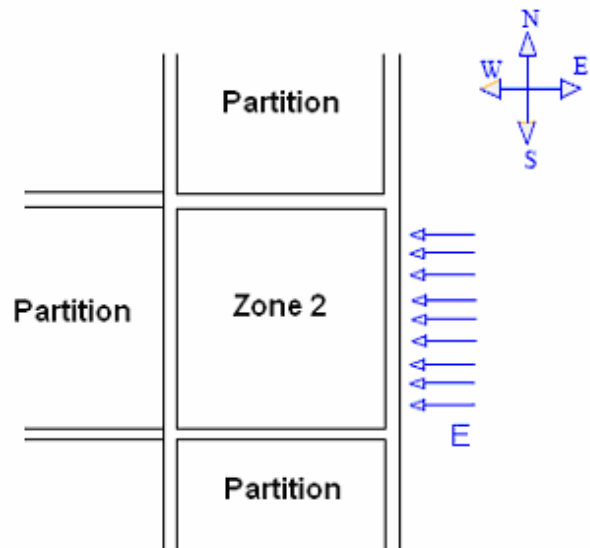
$Q = U * A * CLTD_c$

$U = 0.2 \text{ Btu/h.ft}^2.F$

$A = 192 \text{ ft}^2 (16' * 15' * 80\%) \text{ -----Note 80\% is wall area and 20\% is window glazing}$

Calculation of Heat Load due to Conduction through the walls for Zone-2:

Zone-2(per room)			
Time of day	Side Facing		Q(Btu/hr)
	East CLTD	CLTD _c	
9	16	11	422
10	19	14	538
11	22	17	653
12	25	20	768
13	27	22	845
14	28	23	883
15	29	24	922
16	29	24	922
17	29	24	922



Equations

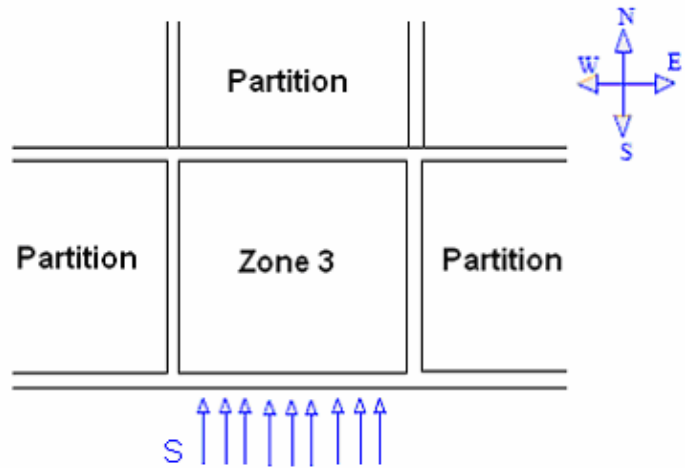
$$Q = U * A * CLTD_c$$

$$U = 0.2 \text{ Btu/h.ft}^2.F$$

$$A = 192 \text{ ft}^2 (16' * 15' * 80\%) \text{ -----Note 80\% is wall area and 20\% is window glazing}$$

Calculation of Heat Load due to Conduction through the walls for Zone-3:

Zone-3(per room)			
Time of day	Side Facing South		Q (Btu/hr)
	CLTD	CLTD _c	
9	10	5	192
10	10	5	192
11	11	6	230
12	12	7	269
13	14	9	346
14	16	11	422
15	19	14	538
16	21	16	614
17	23	18	691



Equations

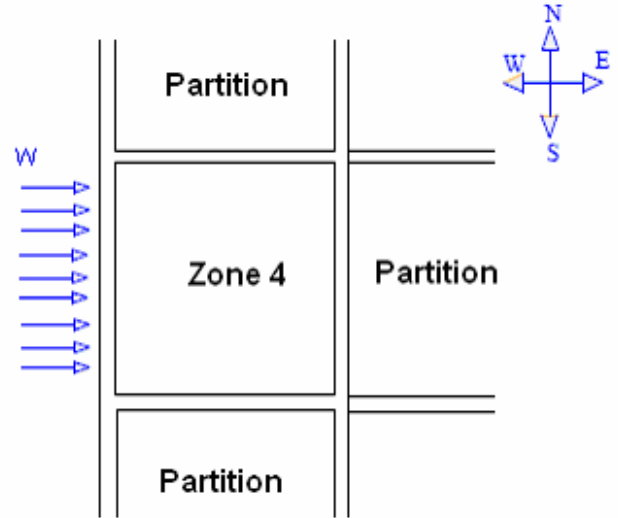
$$Q = U * A * CLTD_c$$

$$U = 0.2 \text{ Btu/h.ft}^2.F$$

$$A = 192 \text{ ft}^2 (16' * 15' * 80\%) \text{ -----Note 80\% is wall area and 20\% is window glazing}$$

Calculation of Heat Load due to Conduction through the walls for Zone-4:

Zone-4(per room)			
Time of day	Side Facing		Q (Btu/hr)
	West CLTD	CLTD _c	
9	15	10	384
10	15	10	384
11	14	9	346
12	14	9	346
13	15	10	384
14	15	10	384
15	17	12	461
16	20	15	576
17	23	18	691



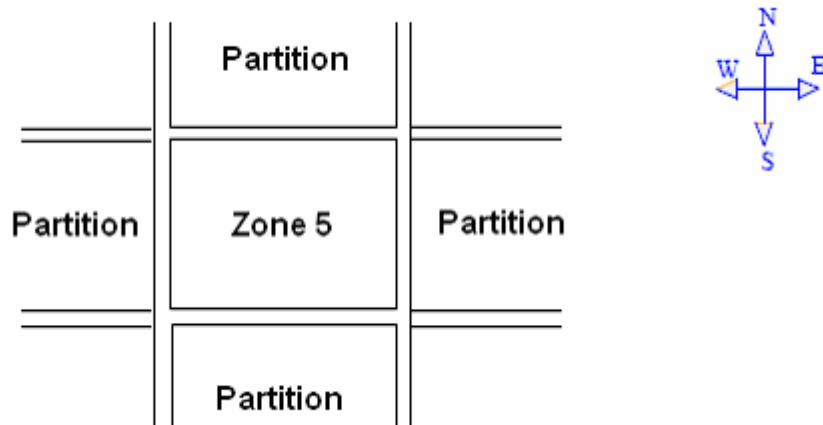
Equations

$$Q = U * A * CLTD_c$$

$$U = 0.2 \text{ Btu/h.ft}^2.F$$

$$A = 192 \text{ ft}^2 (16' * 15' * 80\%) \text{ -----Note 80\% is wall area and 20\% is window glazing}$$

Calculation of Heat Load due to Conduction through the walls for Zone- 5:



Equations

$$Q = U * A * CLTD_c$$

$$U = 0.2 \text{ Btu/h.ft}^2.F$$

$$A = 0 \text{ ft}^2$$

Zone 5 is an internal zone with NO solar exposure. Therefore heat load due to conduction through the walls for zone 5 is Zero.

WINDOWS

Windows heat load has two components 1) Conduction and 2) Radiation

CONDUCTION LOAD

CONDUCTION LOAD FOR WINDOW(per room)			
Solar Time	CLTD	CLTDc	Q(Btu/hr)
9	2	-3	-79
10	4	-1	-26
11	7	2	53
12	9	4	106
13	12	7	185
14	13	8	211
15	14	9	238
16	14	9	238
17	13	8	211

Equations

$$Q \text{ (Btu/hr)} = U \text{ (Btu/hr- ft}^2 \text{ }^\circ\text{F)} \times A \text{ (sq-ft)} \times \text{CLTDc (F)}$$

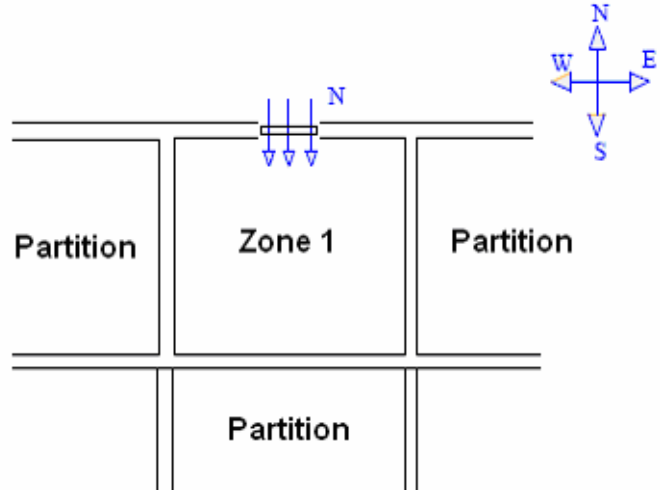
$$U = 0.55 \text{ Btu/hr -ft}^2 \text{ }^\circ\text{F}$$

$$A = 48 \text{ ft}^2 \text{ (16' * 15' * 20\%)} \text{ ----- 80\% is wall area and 20\% is glazing area}$$

SOLAR TRANSMISSION THROUGH WINDOWS (RADIATION LOAD)

Calculation of Radiation Load through the Windows for Zone 1:

Zone-1(per room)			
Time of day	Side Facing North		Q(Btu/hr)
	SCL	SC	
9	27	0.72	933
10	30	0.72	1037
11	33	0.72	1140
12	34	0.72	1175
13	35	0.72	1210
14	34	0.72	1175
15	32	0.72	1106
16	29	0.72	1002
17	29	0.72	1002



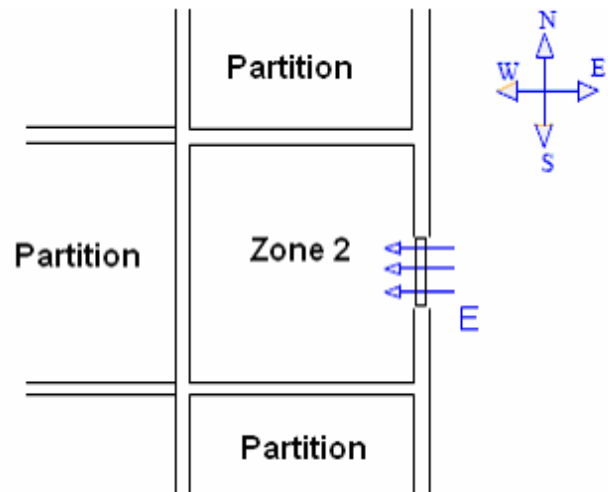
Equations

$Q \text{ (Btu/hr)} = A \text{ (sq-ft)} \times SC \times SCL$

$A = 48 \text{ ft}^2 \text{ (16' * 15' * 20\%)} \text{ ----- } 80\% \text{ is wall area and } 20\% \text{ is glazing area}$

Calculation of Radiation Load through the Windows for Zone 2:

Zone-2(per room)			
Time of day	Side Facing East		Q (Btu/hr)
	SCL	SC	
9	145	0.72	5011
10	124	0.72	4285
11	89	0.72	3076
12	62	0.72	2143
13	56	0.72	1935
14	52	0.72	1797
15	47	0.72	1624
16	43	0.72	1486
17	37	0.72	1279



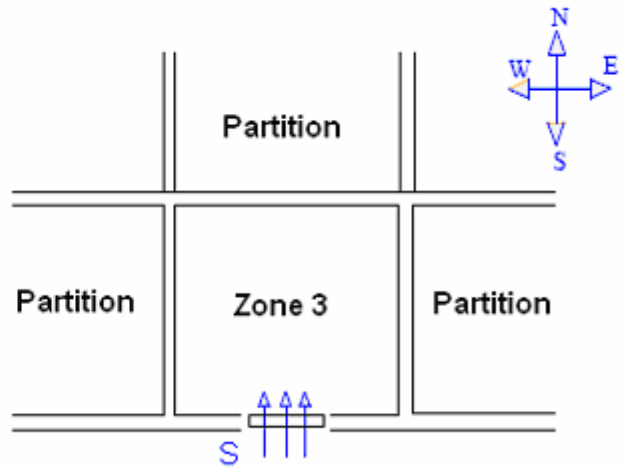
Equations

$Q \text{ (Btu/hr)} = A \text{ (sq-ft)} \times SC \times SCL$

$A = 48 \text{ ft}^2 \text{ (16' * 15' * 20\%)} \text{ ----- } 80\% \text{ is wall area and } 20\% \text{ is glazing area}$

Calculation of Radiation Load through the Windows for Zone 3:

Zone-3(per room)			
Time of d	Side Facing		Q (Btu/hr)
	South	SC	
	SCL	SC	
9	36	0.72	1244
10	54	0.72	1866
11	70	0.72	2419
12	79	0.72	2730
13	79	0.72	2730
14	70	0.72	2419
15	54	0.72	1866
16	40	0.72	1382
17	33	0.72	1140



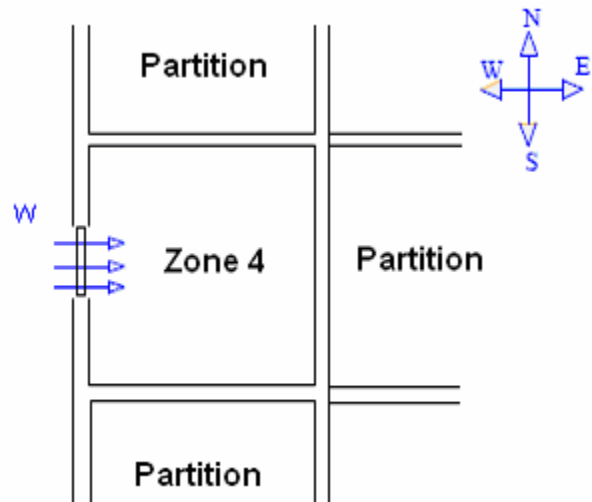
Equations

$Q \text{ (Btu/hr)} = A \text{ (sq-ft)} \times SC \times SCL$

$A = 48 \text{ ft}^2 \text{ (} 16' \times 15' \times 20\% \text{)} \text{ ----- } 80\% \text{ is wall area and } 20\% \text{ is glazing area}$

Calculation of Radiation Load through the Windows for Zone 4:

Zone-4(per room)			
Time of day	Side Facing		Q (Btu/hr)
	West	SC	
	SCL	SC	
9	31	0.72	1071
10	34	0.72	1175
11	36	0.72	1244
12	37	0.72	1279
13	59	0.72	2039
14	98	0.72	3387
15	132	0.72	4562
16	153	0.72	5288
17	156	0.72	5391



Equations

$Q \text{ (Btu/hr)} = A \text{ (sq-ft)} \times SC \times SCL$

$A = 48 \text{ ft}^2 \text{ (} 16' \times 15' \times 20\% \text{)} \text{ ----- } 80\% \text{ is wall area and } 20\% \text{ is glazing area}$

PEOPLE

Calculation of Internal Load (People):

People :

Time	Number of people	Hours after Entry	SHG (Btu/hr)	LHG (Btu/hr)	CLF	Q (sensible) (Btu/hr)	Q (latent) (Btu/hr)	Q (total) (Btu/hr)
9	2	0	250	200	0	0	400	400
10	2	1	250	200	0.61	305	400	705
11	2	2	250	200	0.69	345	400	745
12	2	3	250	200	0.75	375	400	775
13	2	4	250	200	0.79	395	400	795
14	2	5	250	200	0.83	415	400	815
15	2	6	250	200	0.86	430	400	830
16	2	7	250	200	0.89	445	400	845
17	2	8	250	200	0.91	455	400	855

Equations

$Q_{\text{sensible}} \text{ (Btu/hr)} = N \text{ (number of people)} \times \text{SHG (Btu/hr)} \times \text{CLF}$

$Q_{\text{latent}} \text{ (Btu/hr)} = N \text{ (number of people)} \times \text{LHG (Btu/hr)}$

SHG = Sensible heat gain

LHG = Latent heat gain

N = 2 people per room

LIGHTS

Calculation of Internal Load (Lights):

LIGHTS(per room):					
Time	Lights ON for Hours	Wattage	f(b)	CLF	Q (Btu/hr)
9	0	1536	1.3	0	0
10	1	1536	1.3	0.72	4888
11	2	1536	1.3	0.8	5431
12	3	1536	1.3	0.84	5703
13	4	1536	1.3	0.87	5907
14	5	1536	1.3	0.88	5974
15	6	1536	1.3	0.89	6042
16	7	1536	1.3	0.9	6110
17	8	1536	1.3	0.91	6178

Equations

$Q_{\text{lights}} \text{ (Btu/hr)} = 3.4[\text{Btu/hr /Watt}] \times \text{CLF} \times \text{Wattage} \times f(b)$

f_b = Fluorescent ballast factor = 1.3

VENTILATION

Ventilation load has two components 1) Sensible load and 2) Latent load

Sensible Heat

$$Q_{\text{Sensible}} = 60 * \text{CFM} * \rho * C_p * \Delta T$$

$$Q_{\text{Sensible}} = 60 \text{ hr/min} * \text{CFM (ft}^3\text{/min)} * (0.075) \text{ lbm/ft}^3 * (0.24) \text{ Btu/lbm-}^\circ\text{F} * \Delta T (^\circ\text{F})$$

$$Q_{\text{Sensible}} = 1.08 * \text{CFM} * \Delta T$$

ρ = Air density (0.075 lbm/ft³)

C_p = Specific heat of air (0.24 Btu/lb F)

Recommended ventilation is 20 cfm/person; 7 person/1000 sq-ft (block occupancy). For zone wise calculations, 2 persons are assumed per room (this will ensure conservative analysis), therefore,

CFM (Air flow rate) = 2 * 20 = 40 ft³/min (for each room)

Now

$$Q_{\text{Sensible}} = 1.08 * \text{CFM} * \Delta T$$

ΔT = Indoor minus Outdoor temperature difference ($T_o - T_i$)

$T_o = 90^\circ\text{F}$, $T_i = 78^\circ\text{F}$

Therefore

$$Q_{\text{Sensible}} = 1.08 * 40 * (90 - 78)$$

$$Q_{\text{Sensible}} = 44 * (90-78) = 528 \text{ Btu/hr}$$

Latent Heat

$$Q_{\text{Latent}} = 4840 * \text{CFM} * \Delta W$$

Where:

CFM (Air flow rate) = 2 * 20 = 40 ft³/min (for each room)

ΔW = Humidity ratio of indoor air minus humidity ratio of outdoor air, ($W_o - W_i$) in lbm water/ lbm dry air

From Psychometric chart:

Indoor - For 78° F dry bulb temperature (DBT) at 50% Relative Humidity:

$W_i = 0.0102$ lbm water/lbm dry air

Outdoor - For 90 ° F dry bulb temperature (DBT) at 50% Relative Humidity:

$$W_o = 0.0152 \text{ lbm water/lbm dry air}$$

$$\Delta W = (W_o - W_i) = 0.0152 - 0.0102 = 0.005 \text{ lbm water/lbm dry air}$$

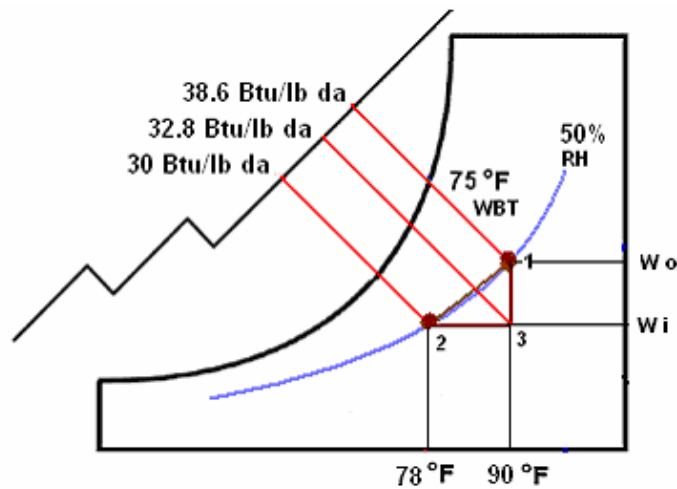
Therefore

$$Q_{\text{Latent}} = 4840 * 40 * 0.005$$

$$Q_{\text{Latent}} = 968 \text{ Btu/hr}$$

ALTERNATIVELY

From Psychrometric Chart (Sea Level):



h_1 = Enthalpy at outdoor conditions. The node 1 represents outdoor conditions at 90°F DBT and 75°F WBT. The corresponding enthalpy (h_1) at this point is 38.6 Btu /lb-da

h_2 = Enthalpy at indoor conditions. The node 2 represents indoor conditions at 78°F DBT and 50% RH. The corresponding enthalpy (h_2) at this point is 30 Btu /lb-da

h_3 = Enthalpy at 3 is 32.8 Btu /lb-da

$$Q_{\text{Sensible}} = \rho \text{ (lbm/ft}^3\text{)} * \text{CFM (ft}^3\text{/min)} * (h_3 - h_2) \text{ (Btu/lbm da)} * 60 \text{ (hr/min)}$$

$$Q_{\text{Sensible}} = 0.075 \text{ (lbm/ft}^3\text{)} * 40 \text{ (ft}^3\text{/min)} * (32.8 - 30) \text{ (Btu/lbm-da)} * 60 \text{ (hr/min)}$$

$$Q_{\text{Sensible}} = 504 \text{ Btu/hr}$$

$$Q_{\text{Latent}} = \rho \text{ (lbm/ft}^3\text{)} * \text{CFM (ft}^3\text{/min)} * (h_1 - h_3) \text{ (Btu/lbm-da)} * 60 \text{ (hr/min)}$$

$$Q_{\text{Latent}} = 0.075 \text{ (lbm/ft}^3\text{)} * 40 \text{ (ft}^3\text{/min)} * (38.6 - 32.8) \text{ (Btu/lbm-da)} * 60 \text{ (hr/min)}$$

$$Q_{\text{Latent}} = 1044 \text{ Btu/hr}$$

APPLIANCES**For each room:**

Number of Computer in each room = 1

Therefore, $Q = 1 * 3.4 \text{ Btu/hr / watt} * 200 \text{ watt}$

$Q = 680 \text{ Btu/hr}$

Kitchen (in central zone 5):

Q_{Sensible} [Water cooler (1no)] = $1 * 5970$ = 5970 Btu/hr (sensible heat)

Q_{Sensible} [Microwaves (2no)] = $2 * 1360$ = 2720 Btu/hr (sensible heat)

Q_{Sensible} [Coffee maker (2no)] = $2 * 3580$ = 7160 Btu/hr (sensible heat)

Q_{Latent} [Coffee maker (2no)] = $2 * 1540$ = 3080 Btu/hour (latent heat)

Total Sensible load due to Kitchen = $5970 + 2720 + 7160$ = 15850 Btu/hr

Total latent load due to Kitchen = 3080 Btu/hr

Office Services (in central zone 5):

Q_{Sensible} [Small Copier (2no)] = $2 * 1570$ = 3140 Btu/hr

Q_{Sensible} [Type Writer (2no)] = $2 * 230$ = 460 Btu/hr

Q_{Sensible} [Letter quality printer (2no)] = $2 * 1000$ = 2000 Btu/hr

Q_{Sensible} [computer (2no)] = $2 * 680$ = 1360 Btu/hr

Therefore, Total head load (Sensible) due to office services is: $= 3140 + 460 + 2000 + 1360 = 6960 \text{ Btu/hr}$

Table below summarizes the heat load for each zone.

APPLIANCES :			
Zone Number	q(sensible) (Btu/hr)	q(latent) (Btu/hr)	q(total) (Btu/hr)
1	680	-	680
1	680	-	680
1	680	-	680
2	680	-	680
2	680	-	680
2	680	-	680
2	680	-	680
3	680	-	680
3	680	-	680
3	680	-	680
4	680	-	680
4	680	-	680
4	680	-	680
4	680	-	680
5	680	-	680
5	680	-	680
5	680	-	680
5	680	-	680
5(Kitchen)	15850	3080	18930
5(Office services)	6960	-	6960

ZONE LOAD SUMMARY

Tables below summarize the sensible, latent and total load for each zone. Note that in tables below, the

- a. Space loads (sensible and latent) = [Q roof + Q walls + Q windows + Q people + Q lights + Q appliances]
(It does not include ventilation load). ----- [This is also called Room Total Load].
- b. Total loads = Space Loads + Ventilation Load----- [This is also called Gross Total Load].

The yellow highlighted row indicates the peak load at particular time of day.

Zone #1 Load

Zone 1:Per Room													
Time of day	Roof	Walls	Windows		People		Lights	Ventilation		Appliances	Space Loads		Q(total)
	(Btu/hr)	(Btu/hr)	Radiation	Conduction	Latent	Sensible	(Btu/hr)	Latent	Sensible	Sensible	Sensible	Latent	(Btu/hr)
			(Btu/hr)	(Btu/hr)	(Btu/hr)	(Btu/hr)		(Btu/hr)	(Btu/hr)	(Btu/hr)	(Btu/hr)	(Btu/hr)	
9	155	154	933	-79	400	0	0	1044	504	680	1843	400	3791
10	169	154	1037	-26	400	305	4888	1044	504	680	7207	400	9155
11	211	154	1140	53	400	345	5431	1044	504	680	8014	400	9962
12	268	192	1175	106	400	375	5703	1044	504	680	8499	400	10447
13	324	192	1210	185	400	395	5906	1044	504	680	8892	400	10840
14	394	230	1175	211	400	415	5974	1044	504	680	9079	400	11027
15	465	260	1106	238	400	430	6042	1044	504	680	9221	400	11169
16	521	346	1002	238	400	445	6110	1044	504	680	9342	400	11290
17	577	384	1002	211	400	455	6178	1044	504	680	9487	400	11435

Zone #2 Load

Zone 2:Per Room													
Time of day	Roof	Walls	Windows		People		Lights	Ventilation		Appliances	Space Loads		Q(total)
	(Btu/hr)	(Btu/hr)	Radiation	Conduction	Latent	Sensible	(Btu/hr)	Latent	Sensible	Sensible	Sensible	Latent	(Btu/hr)
			(Btu/hr)	(Btu/hr)	(Btu/hr)	(Btu/hr)		(Btu/hr)	(Btu/hr)	(Btu/hr)	(Btu/hr)		
9	155	422	5011	-79	400	0	0	1044	504	680	6189	400	8137
10	169	538	4285	-26	400	305	4888	1044	504	680	10839	400	12787
11	211	653	3076	53	400	345	5431	1044	504	680	10449	400	12397
12	268	765	2143	106	400	375	5703	1044	504	680	10040	400	11988
13	324	845	1935	185	400	395	5906	1044	504	680	10270	400	12218
14	394	883	1797	211	400	415	5974	1044	504	680	10354	400	12302
15	465	922	1624	238	400	430	6042	1044	504	680	10401	400	12349
16	521	922	1486	238	400	445	6110	1044	504	680	10402	400	12350
17	577	922	1279	211	400	455	6178	1044	504	680	10302	400	12250

Zone # 3 Load

Zone 3:Per Room													
Time of day	Roof	Walls	Windows		People		Lights	Ventilation		Appliances	Space Loads		Q(total)
	(Btu/hr)	(Btu/hr)	Radiation	Conduction	Latent	Sensible	(Btu/hr)	Latent	Sensible	Sensible	Sensible	Latent	(Btu/hr)
			(Btu/hr)	(Btu/hr)	(Btu/hr)	(Btu/hr)		(Btu/hr)	(Btu/hr)	(Btu/hr)	(Btu/hr)		
9	155	192	1244	-79	400	0	0	1044	504	680	2192	400	4140
10	169	192	1866	-26	400	305	4888	1044	504	680	8074	400	10022
11	211	230	2419	53	400	345	5431	1044	504	680	9369	400	11317
12	268	269	2730	106	400	375	5703	1044	504	680	10131	400	12079
13	324	346	2730	185	400	395	5906	1044	504	680	10566	400	12514
14	394	422	2419	211	400	415	5974	1044	504	680	10515	400	12463
15	465	538	1866	238	400	430	6042	1044	504	680	10259	400	12207
16	521	614	1382	238	400	445	6110	1044	504	680	9990	400	11938
17	577	691	1140	211	400	455	6178	1044	504	680	9932	400	11880

Zone # 4 Load

Zone 4:Per Room

Time of day	Roof	Walls	Windows		People		Lights	Ventilation		Appliances	Space Loads		Q(total)
	(Btu/hr)	(Btu/hr)	Radiation	Conduction	Latent	Sensible	(Btu/hr)	Latent	Sensible	(Btu/hr)	Sensible	Latent	(Btu/hr)
			(Btu/hr)	(Btu/hr)	(Btu/hr)	(Btu/hr)		(Btu/hr)	(Btu/hr)		(Btu/hr)	(Btu/hr)	
9	155	384	1071	-79	400	0	0	1044	504	680	2211	400	4159
10	169	384	1175	-26	400	305	4888	1044	504	680	7575	400	9523
11	211	346	1244	53	400	345	5431	1044	504	680	8310	400	10258
12	268	346	1279	106	400	375	5703	1044	504	680	8757	400	10705
13	324	384	2039	185	400	395	5906	1044	504	680	9913	400	11861
14	394	384	3387	211	400	415	5974	1044	504	680	11445	400	13393
15	465	461	4562	238	400	430	6042	1044	504	680	12878	400	14826
16	521	576	5288	238	400	445	6110	1044	504	680	13858	400	15806
17	577	691	5391	211	400	455	6178	1044	504	680	14183	400	16131

Zone # 5 Load

Zone 5:Per Room

Time of day	Roof	Walls	Windows		People		Lights	Ventilation		Appliances	Space Loads		Q(total)
	(Btu/hr)	(Btu/hr)	Radiation	Conduction	Latent	Sensible	(Btu/hr)	Latent	Sensible	(Btu/hr)	Sensible	Latent	(Btu/hr)
			(Btu/hr)	(Btu/hr)	(Btu/hr)	(Btu/hr)		(Btu/hr)	(Btu/hr)		(Btu/hr)	(Btu/hr)	
9	155	0	0	0	400	0	0	1044	504	680	835	400	2783
10	169	0	0	0	400	305	4888	1044	504	680	6042	400	7990
11	211	0	0	0	400	345	5431	1044	504	680	6667	400	8615
12	268	0	0	0	400	375	5703	1044	504	680	7026	400	8974
13	324	0	0	0	400	395	5906	1044	504	680	7305	400	9253
14	394	0	0	0	400	415	5974	1044	504	680	7463	400	9411
15	465	0	0	0	400	430	6042	1044	504	680	7617	400	9565
16	521	0	0	0	400	445	6110	1044	504	680	7756	400	9704
17	577	0	0	0	400	455	6178	1044	504	680	7890	400	9838

Zone # 5 Kitchen Room

Zone 5:Per Room

Time of day	Roof	Walls	Windows		People		Lights	Ventilation		Appliances	Space Loads		Q(total)
	(Btu/hr)	(Btu/hr)	Radiation	Conduction	Latent	Sensible	(Btu/hr)	Latent	Sensible	(Btu/hr)	Sensible	Latent	(Btu/hr)
			(Btu/hr)	(Btu/hr)	(Btu/hr)	(Btu/hr)		(Btu/hr)	(Btu/hr)		(Btu/hr)	(Btu/hr)	
9	155	0	0	0	400	0	0	1044	504	680	835	400	2783
10	169	0	0	0	400	305	4888	1044	504	680	6042	400	7990
11	211	0	0	0	400	345	5431	1044	504	680	6667	400	8615
12	268	0	0	0	400	375	5703	1044	504	680	7026	400	8974
13	324	0	0	0	400	395	5906	1044	504	680	7305	400	9253
14	394	0	0	0	400	415	5974	1044	504	680	7463	400	9411
15	465	0	0	0	400	430	6042	1044	504	680	7617	400	9565
16	521	0	0	0	400	445	6110	1044	504	680	7756	400	9704
17	577	0	0	0	400	455	6178	1044	504	680	7890	400	9838

Zone # 5 Office Services Room

Zone 5:(Office services room)														
Time of day	Roof	Walls	Windows		People		Lights	Ventilation		Appliances		Space Loads		q(total)
	(Btu/hr)	(Btu/hr)	Radiation (Btu/hr)	Conduction (Btu/hr)	Latent (Btu/hr)	Sensible (Btu/hr)	(Btu/hr)	Latent (Btu/hr)	Sensible (Btu/hr)	Latent (Btu/hr)	sensible (Btu/hr)	Sensible (Btu/hr)	Latent (Btu/hr)	(Btu/hr)
9	155	0	0	0	400	0	0	1044	504	0	6960	7115	400	9063
10	169	0	0	0	400	305	4888	1044	504	0	6960	12322	400	14270
11	211	0	0	0	400	345	5431	1044	504	0	6960	12947	400	14895
12	268	0	0	0	400	375	5703	1044	504	0	6960	13306	400	15254
13	324	0	0	0	400	395	5906	1044	504	0	6960	13585	400	15533
14	394	0	0	0	400	415	5974	1044	504	0	6960	13743	400	15691
15	465	0	0	0	400	430	6042	1044	504	0	6960	13897	400	15845
16	521	0	0	0	400	445	6110	1044	504	0	6960	14036	400	15984
17	577	0	0	0	400	455	6178	1044	504	0	6960	14170	400	16118

Results:

Sensible and Latent Peaks for Each Zone (highlighted yellow)

<u>Time</u>	9	10	11	12	13	14	15	16	17
Load of the Zones (Btu/hr)									
Zone-1 (Three Room)									
Sensible	5529	21621	24042	25497	26676	27237	27663	28026	28461
Latent	1200	1200	1200	1200	1200	1200	1200	1200	1200
Zone-2(Four Rooms)									
Sensible	24756	43356	41796	40160	41080	41416	41604	41608	41208
Latent	1600	1600	1600	1600	1600	1600	1600	1600	1600
Zone-3(Three Room)									
Sensible	6576	24222	28107	30393	31698	31545	30777	29970	29796
Latent	1200	1200	1200	1200	1200	1200	1200	1200	1200
Zone-4(Three Rooms)									
Sensible	8844	30300	33240	35028	39652	45780	51512	55432	56732
Latent	1600	1600	1600	1600	1600	1600	1600	1600	1600
Zone-5(Four Rooms)									
Sensible	3340	24168	26668	28104	29220	29852	30468	31024	31560
Latent	1600	1600	1600	1600	1600	1600	1600	1600	1600
Zone-5(Kitchen Room)									
Sensible	16005	21212	21837	22196	22475	22633	22787	22926	23060
Latent	3480	3480	3480	3480	3480	3480	3480	3480	3480
Zone-5(Office Services Room)									
Sensible	7115	12322	12947	13306	13585	13743	13897	14036	14170
Latent	400	400	400	400	400	400	400	400	400
TOTAL (Sensible)	72165	177201	188637	194684	204386	212206	218708	223022	224987
TOTAL (Latent)	11080	11080	11080	11080	11080	11080	11080	11080	11080

***The highlighted area in each table shows the Maximum Loads.**

Total Sensible & Latent Loads due to Ventilation

Load of the Zones (Btu/hr)	Ventilation Load per zone	No of Rooms	Ventilation Load
Zone-1 (Three Room)			
Sensible	504	3	1512
Latent	1044	3	3132
Zone-2(Four Rooms)			
Sensible	504	4	2016
Latent	1044	4	4176
Zone-3(Three Room)			
Sensible	504	3	1512
Latent	1044	3	3132
Zone-4(Four Rooms)			
Sensible	504	4	2016
Latent	1044	4	4176
Zone-5(Four Rooms)			
Sensible	504	4	2016
Latent	1044	4	4176
Zone-5(Kitchen Room)			
Sensible	504	1	504
Latent	1044	1	1044
Zone-5(Office Services Room)			
Sensible	504	1	504
Latent	1044	1	1044
TOTAL (Sensible)			10080
TOTAL (Latent)			20880

Maximum Sensible & Latent Loads for the Building

<u>Time</u>	9	10	11	12	13	14	15	16	17
Load of the Zones (Btu/hr)									
Zone-1 (Three Room)									
Sensible	7041	23133	25554	27009	28188	28749	29175	29538	29973
Latent	4332	4332	4332	4332	4332	4332	4332	4332	4332
Zone-2(Four Rooms)									
Sensible	26772	45372	43812	42176	43096	43432	43620	43624	43224
Latent	5776	5776	5776	5776	5776	5776	5776	5776	5776
Zone-3(Three Room)									
Sensible	8088	25734	29619	31905	33210	33057	32289	31482	31308
Latent	4332	4332	4332	4332	4332	4332	4332	4332	4332
Zone-4(Four Rooms)									
Sensible	10860	32316	35256	37044	41668	47796	53528	57448	58748
Latent	5776	5776	5776	5776	5776	5776	5776	5776	5776
Zone-5(Four Rooms)									
Sensible	5356	26184	28684	30120	31236	31868	32484	33040	33576
Latent	5776	5776	5776	5776	5776	5776	5776	5776	5776
Zone-5(Kitchen Room)									
Sensible	16509	21716	22341	22700	22979	23137	23291	23430	23564
Latent	1444	1444	1444	1444	1444	1444	1444	1444	1444
Zone-5(Office ServicesRoom)									
Sensible	7619	12826	13451	13810	14089	14247	14401	14540	14674
Latent	1444	1444	1444	1444	1444	1444	1444	1444	1444
TOTAL (Sensible) Btu/hr	82245	187281	198717	204764	214466	222286	228788	233102	235067
TOTAL (Latent) Btu/hr	28880	28880	28880	28880	28880	28880	28880	28880	28880

CONCLUSIONS:

Cooling Loads for Individual Zones are calculated. Also the cooling loads for room in the zones are calculated. The maximum load of the building is found at 17.00 hours. The maximum space loads which required for the building are 224987 Btu/hr sensible, 11080 Btu/hr latent.

Also the maximum total loads for the building (included space and fresh air loads) are 235067 Btu/hr "sensible", 28880 Btu/hr "latent".

11.2 Cooling Load Example in Metric Units

A school classroom is 6 m long, 6 m wide and 3 m high. There is a 2.5 m x 4 m window in the east wall. Only the east wall/window is exterior. Assume the thermal conditions in adjacent spaces (west, south, north, above and below) are the same as those of the classroom. Determine the cooling load at 9:00 am, 12:00 noon on July 21.

Other known conditions include:

Latitude = 40° N

Ground reflectance = 0.2

Clear sky with a clearness number = 1.0

Overall window heat transmission coefficient = 7.0 W/m²K

Room dry-bulb temperature = 25.5°C

Permissible temperature exceeded = 2.5%

Schedule of occupancy: 20 people enter at 8:00 am and stay for 8 hours

Lighting schedule: 300 W on at 8:00 AM for 8 hours

Exterior wall structure:

- Outside surface, A0
- Face brick (100 mm), A2
- Insulation (50 mm), B3
- Concrete block (100 mm), C3
- Inside surface, E0

Exterior window:

- Single glazing, 3 mm
- No exterior shading, SC = 1.0

SOLUTION:

Cooling Load due to Exterior wall:

Layer	Unit Resistance (R value) m ² K/W
A0	0.059
A2	0.076
B3	1.173
C3	0.125
E0	0.121
Total	1.554

$$U = 1/R = 0.643 \text{ W/m}^2 \text{ K}$$

From Table A28-33A, find wall type 13.

From Table A28-32, CLTD_{9:00} = 9 and CLTD_{12:00} = 14 for east wall.

$$\text{CLTD}_{\text{Corrected}} = \text{CLTD} + (25.5 - T_i) + (T_m - 29.4)$$

T_i = inside temperature

T_m = mean outdoor temperature

$$T_m = (\text{maximum outdoor temperature}) - (\text{daily range})/2$$

$$\text{At 9:00 am } \text{CLTD}_{\text{Corrected}} = 9 + (25.5 - 25.5) + (31 - 9/2 - 29.4) = 6.1 \text{ K}$$

$$\text{At 12:00 am } \text{CLTD}_{\text{Corrected}} = 14 + (25.5 - 25.5) + (31 - 9/2 - 29.4) = 11.1 \text{ K}$$

$$Q = U A (\text{CLTD})$$

$$Q = 0.643 \text{ (W/m}^2 \text{ K)} \times (6 \times 3 - 4 \times 2.5) \text{ m}^2 \times 6.1 = 34 \text{ W (at 9 am)}$$

$$Q = 0.643 \text{ (W/m}^2 \text{ K)} \times (6 \times 3 - 4 \times 2.5) \text{ m}^2 \times 11.1 = 62 \text{ W (at 12 noon)}$$

Window conduction:

From Table A28-34, CLTD_{9:00} = 1 and CLTD_{12:00} = 5

$$\text{CLTD}_{\text{Corrected}} = \text{CLTD} + (25.5 - T_i) + (T_m - 29.4)$$

$$\text{CLTD}_{\text{corrected}} = 1 + 0 + (31 - 9/2 - 29.4) = -1.9 \text{ K (at 9 am)}$$

$$\text{CLTD}_{\text{Corrected}} = 5 + 0 + (31 - 9/2 - 29.4) = 2.1 \text{ K (at 12 noon)}$$

$$Q = U A (\text{CLTD})$$

$$Q = 7.0 \text{ W/m}^2 \text{ K} \times 4 \times 2.5 \text{ m}^2 \times (-1.9 \text{ K}) = -133 \text{ W} \quad (\text{at 9 am})$$

$$Q = 7.0 \text{ W/m}^2 \text{ K} \times 4 \times 2.5 \text{ m}^2 \times 2.1 \text{ K} = 147 \text{ W} \quad (\text{at 12 noon})$$

Solar load through glass:

From Table A28-35B, find zone type is A

From Table A28-36, find CLF = 576 at 9 am and CLF = 211 at 12 noon

$$Q = A (\text{SC}) (\text{SCL})$$

$$Q = (2.5 \times 4 \text{ m}^2) \times 1.0 \times 576 = 5760 \text{ W} \quad (\text{at 9 am})$$

$$Q = (2.5 \times 4 \text{ m}^2) \times 1.0 \times 211 = 2110 \text{ W} \quad (\text{at 12 noon})$$

Cooling load from partitions, ceiling, floors:

$$Q = 0$$

People:

From Table A28-35B, find zone type is B

From Table A28-3, find sensible/latent heat gain = 70 W

From Table A28-37, find CLF_{9:00} (1) = 0.65 and CLF_{12:00} (3) = 0.85

$$Q_{\text{Sensible}} = N (\text{sensible heat gain}) \text{ CLF}$$

$$Q_{\text{Sensible}} = 20 \times 70 \text{ W} \times 0.65 = 910 \text{ W} \quad (\text{at 9 am})$$

$$Q_{\text{Sensible}} = 20 \times 70 \times 0.85 = 1190 \text{ W} \quad (\text{at 12 noon})$$

$$Q_{\text{Latent}} = N (\text{latent heat gain})$$

$$Q_{\text{Latent}} = 20 \times 45 = 900 \text{ W} \quad (\text{at 9 am and 12 noon})$$

Lighting:

From Table A28-35B, find zone type is B

From Table A28-38, find CLF_{9:00} (1) = 0.75 and CLF_{12:00} (3) = 0.93

$$Q_{\text{Lighting}} = W F_{\text{ul}} F_{\text{sa}} (\text{CLF})$$

$$Q_{\text{Lighting}} = 300 \text{ W} \times 1 \times 1 \times 0.75 = 225 \text{ W} \quad (\text{at 9 am})$$

$$Q_{\text{Lighting}} = 300 \times 1 \times 1 \times 0.93 = 279 \text{ W}$$

(at 12 noon)

Appliances:

$$Q_{\text{Sensible}} = 0$$

$$Q_{\text{Latent}} = 0$$

Infiltration:

Since room air pressure is positive, we have:

$$Q_{\text{Sensible}} \approx 0$$

$$Q_{\text{Latent}} = 0$$

TOTAL COOLING LOAD:

Component	9.00 AM	12.00 Noon
Wall	34	62
Window conduction	-133	147
Window solar transmission	5760	2110
Partitions	0	0
People	910	1190
Lights	225	279
Appliances	0	0
Infiltration	0	0
Total	6796 W	3788 W

12.0 COMPUTER PROGRAMS

The usual need to look up values from various tables and charts makes the conventional hand calculation quite laborious, time consuming and prone to errors and inaccuracies because of the tendency to simplify truncate or interpolate tabulated values. Listed below are the suggested applications and methodology of two computer programs Elite CHVAC and TRACE 700 that may be used in the cooling and heating load calculations.

12.1 Elite CHVAC

CHVAC is a commercial Heating, Ventilation and Air Conditioning software platform developed by Elite Software. This computer program calculates the maximum heating and cooling loads in commercial and industrial buildings.

Listed below are capabilities of CHVAC:

- Calculates peak heating and cooling loads
- Calculates both heating & cooling airflow CFM requirements
- Calculates run out and main trunk duct sizes
- Automates compliance with ASHRAE Standard 62
- Provides overall building envelope report
- Spreadsheet file compatibility
- Performs complete psychrometric analysis
- Prints bar graphs and exploded pie charts
- Exterior shading handles overhangs, fins, & glass tilt
- Uses exact ASHRAE CLTD procedures
- Built-in design weather data for hundreds of cities
- Analyzes up to 12 months per calculation
- Calculates 24 hours per design day
- Allows unlimited number of zones per project
- Zones may be grouped under 100 air handlers
- Zones may be optionally grouped under VAV boxes
- Allows 12 walls, 12 windows, and 5 roofs per zone
- Allows simultaneous infiltration and ventilation

- Allows different summer and winter air rates
- Allows varying indoor conditions within a project
- Allows 6 master roof types, 8 master wall types, 8 master partition types, and 20 master glass types
- Allows glass to be tilted from 0 to 180 degrees
- Allows for roof and wall color effects
- Provision for both VAV and constant volume systems
- Proper handling of return air plenum loads
- Accounts for people diversity in total building load
- Computes supply fan horsepower and heat gains
- Accounts for both draw-thru and blow-thru fans
- Calculates reheat requirements if necessary
- Computes supply and return duct gains and losses
- Allows direct specification of supply CFM quantities
- Allows specification of minimum supply air quantities
- Allows heating only, cooling only, or both
- Excess supply air can be handled as reheat, reserve capacity, or by adjusting the leaving coil conditions
- Leaving coil conditions can be specified with a desired dry bulb temperature or a relative humidity
- Calculates chilled and hot water coil flow rates
- Allows for pretreated outside air
- Allows heating and cooling safety factors
- Lighting & equipment watts along with no. of people can be entered directly or on a per square foot basis

12.1.1 Calculation Method

CHVAC performs calculations using the CLTD/CLF procedures described in the ASHRAE Handbook of Fundamentals. The programs use exact CLTD and MSHGF table values where possible, otherwise direct

calculations are made. This calculation technique allows the programs to calculate for any building orientation and still produce output results that can be easily verified by hand.

12.1.2 Program Input

CHVAC is a true Windows program that uses all the latest data entry techniques such as toolbars, hyper linked help, and form tabs. All input data is checked at the time of entry so that no improper data can be entered. Five types of data are requested: general project data, outdoor design data, building material data, air handler data, and specific zone data. The general project data includes the project and client name, designer, building opening and closing hours, internal operating load schedules, and any desired safety factors. The outdoor design data includes the summer and winter outdoor design conditions (automatically looked up for you if a city reference is given) and the desired ventilation and infiltration rates. The building material data includes the definition of master building material types for roofs, walls, partitions, glass sections, and exterior shading. A user defined material library is available for saving the data on common material types. The air handler data includes the fan and terminal type, the desired heating and cooling supply air temperatures and data for duct heat gains and losses. The zone data includes the zone name, floor length and width, number of people, equipment watts, lighting watts, external shading data, and specific roof, wall, partition, floor, and glass data.

12.1.3 Program Output

The CHVAC program provides eleven types of reports, which can be selectively previewed on screen or printed as desired. CHVAC supports all printers that work with Windows and numerous full color reports are available. The reports are: General Project Data, Air Handler Input Data, Zone Input Data, Detailed Project Zone Loads, Air System Zone Summary, Total Building, Air System, and Zone Load Profiles, Air System Total Load Summary, Air System Psychrometric Analysis, Overall Building Envelope Report, Pie Charts, Bar Graphs, and the Total Building Load Summary. Air system summary data can be exported to your favorite spreadsheet.

12.2 TRACE® 700

The TRACE Load® 700 program is a commercial Heating, Ventilation and Air Conditioning software platform developed by Trane's CDS Group.

The Load phase of the program computes the peak sensible and latent zone loads, as well as the block sensible and latent loads for the building. In addition, the hourly sensible and latent loads, including weather-dependent loads, are calculated for each zone, based on the weather library. The building heating/cooling load calculations, used in the load phase of the program for annual energy consumption

analysis, are of sufficient detail to permit the evaluation of the effect of building data such as orientation, size, shape and mass, heat transfer characteristics of air and moisture, as well as hourly climatic data.

The Design phase of the TRACE program calculates the design supply air temperatures, heating and cooling capacities, and supply air quantities given the peak load files generated by the Load phase. For applications where the building design parameters are known, you can override the calculation of these values using optional entries to the System phase. This gives you the ability to simulate existing buildings with installed equipment that may not be sized according to the loads calculated in the program's Load Phase.

Beyond this, the calculations used to simulate the operation of the building and its service systems through a full-year operating period, are of sufficient detail to permit the evaluation of the effect of system design, climatic factors, operational characteristics and mechanical equipment operating characteristics on annual energy usage. Manufacturers' data is used in the program for the simulation of all systems and equipment.

The calculation procedures used in TRACE are based upon 8,760 hours of operation of the building and its service system. These procedures use techniques recommended in the appropriate ASHRAE publications or produce results that are consistent with such recommended techniques. The following are the program features:

Project Navigator View

- Organizes entries by task to lead you through the modeling process
- Displays the status of each modeling step
- Accommodates up to 4 alternatives per project

Project Tree View

- Organizes all rooms, systems, and plants in a hierarchical list
- Displays all information about a system, zone, or room on 1 screen
- Supports cut, copy, and paste to save entry time

Component Tree

- Displays cooling set point for every room in the project on 1 screen
- Makes it easy to check and edit your work

Task-oriented display guides you through the modeling process as follows:

Select weather information

- Provides both design and typical weather data by location
- Choose from 400 climate locations

- Import standard weather files for a full-year (8760) analysis

Create rooms

- Describe the construction, airflows, thermostat settings, heat sources, and schedules by room

Create airside systems

- Choose from more than 30 methods of air distribution
- Add energy recovery, economizers, and dedicated ventilation/makeup air

Assign rooms to systems

- Create thermal zones and assign them to systems
- Determine airflows, coil loads, and fan sizes for each airside system

If the only requirement is to calculate the cooling and/or heating loads and the project does not require energy analysis and economic evaluation, it is recommended that the Program Load® 700 be utilized instead of Full TRACE 700. The only difference is that Trace Load® 700 users only have access to the Load Design section (from Project Information to Assign Rooms to Systems). Full TRACE 700 users will have full access to the Load, Energy and Economic sections.

The advantage of using only Trace Load® 700 is that all the added features and capabilities (applicable to load design) in full TRACE 700 program are also available to the Trace Load® 700 users. Also, same file extensions and libraries will enable users of both programs to transfer archived files back and forth without any additional steps needed.

13.0 REFERENCES

- 2001 ASHRAE Handbook of Fundamentals
- 1997 ASHRAE Handbook of Fundamentals
- ASHRAE Cooling and Heating Load Calculation Manual
- ASHRAE Standard 62, Indoor Air Quality
- Trane C.D.S. Program Description and Capabilities
- Elite Software Program Description and Capabilities