PDHonline Course M196 (4 PDH)

HVAC Made Easy: A Guide to Heating & Cooling Load Estimation

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Course Content

AIR CONDITIONING SYSTEM OVERVIEW

Cooling & heating load calculations are normally made to size HVAC (heating, ventilating, and air-conditioning) systems and their components. In principle, the loads are calculated to maintain the indoor design conditions. The first step in any load calculation is to establish the design criteria for the project that involves consideration of the building concept, construction materials, occupancy patterns, density, office equipment, lighting levels, comfort ranges, ventilations and space specific needs. Architects and other design engineers converse at early stages of the project to produce design basis & preliminary architectural drawings. The design basis typically includes information on:

1) Geographical site conditions (latitude, longitude, wind velocity, precipitation etc.)
2) Outdoor design conditions (temperature, humidity etc)
3) Indoor design conditions
4) Building characteristics (materials, size, and shape)
5) Configuration (location, orientation and shading)
6) Operating schedules (lighting, occupancy, and equipment)
7) Additional considerations (type of air-conditioning system, fan energy, fan location, duct heat loss and gain, duct leakage, type and position of air return system…)

Climate data requirements

One of the most important things in building HVAC design is the climate you are designing. Let's make first distinction in terms "weather" and "climate".

"*Weather*" is the set of atmospheric conditions prevailing at a given place and time. "*Climate*" can be defined as the integration in time of weather conditions, characteristics of a certain geographical location. At the global level climates are formed by the differential solar heat input and the uniform heat emission over the earth's surface.

Climate has a major effect on building performance, HVAC design and energy consumption. It is also pertinent to the assessment of thermal comfort of the occupants. The key objectives of climatic design include:

1) To reduce energy cost of a building
2) To use "natural energy" as far as possible instead of mechanical system and power
3) To provide comfortable and healthy environment for people

Classification of climates

Many different systems of climate classification are in use for different purposes. Climatic zones such as tropical, arid, temperature and cool are commonly found for representing climatic conditions. For the purposes of building design a simple system based on the nature of the thermal problem in the particular location is often used.

1) *Cold climates*, where the main problem is the lack of heat (under heating), or excessive heat dissipation for all or most parts of the year.
2) *Temperate climates*, where there is a seasonal variation between under heating and overheating, but neither is very severe.
3) *Hot-dry (arid) climates*, where the main problem is overheating, but the air is dry, so the evaporative cooling mechanism of the body is not restricted. There is usually a large diurnal (day - night) temperature variation.
4) *Warm-humid climates*, where the overheating is not as great as in hot-dry areas, but it is aggravated by very high humidity's, restricting the evaporation potential. The diurnal temperature variation is small.

Six categories of climates:

1) Warm-humid - 15°N and South of the equator, e.g. Lagos, Mombassa, Colombo, Jakarta etc.
2) Warm-humid Island - equatorial and trade wind zones, e.g. Caribbean, Philippines and Pacific Islands etc.
3) Hot-dry desert - 15° to 30° North and South, e.g. Baghdad, Alice Springs, Phoenix etc.
4) Hot-dry maritime desert - latitudes as (3), coastal large landmass, Kuwait, Karachi etc.
5) Composite Monsoon - Tropic Cancer/Capricorn, Lahore, Mandalay, New Delhi etc.
6) Tropical uplands - Tropic Cancer/Capricorn, 900 to 1200 meters above sea level (plateau and mountains), Addis Ababa, Mexico City, Nairobi etc.

Load Calculations Methods

Before one can design an efficient and effective air conditioning system, the load must first be calculated using established techniques. There are various methods in use. The most basic of these methods is a rule-of-thumb value -- for example, square feet of floor area per ton of cooling. The "square-foot-per-ton" sizing method avoids calculating the cooling load of the building and proceeds directly from the square footage of the building. While this approach is rapid and simple, it does not account for orientation of the walls and windows, the difference in surface area between a one-story and a two-story home of the same floor area, the differences in insulation and air leakage between different buildings, the number of occupants, and many other factors. Such rules-of-thumb are useful in schematic design as a means of getting an approximate handle on equipment size and cost.

The more refined methods available in the HVAC handbooks are:

1) Total Equivalent Temperature Difference/Time Average (TETD/TA)
2) Cooling Load Temperature Difference/Cooling Load Factor (CLTD/CLF)
3) Transfer Function Method (TFM)
4) Heat Balance (HB) & Radiant Time Series (RTS)
5) Manual J Method for Residential Applications & Manual N for Commercial Buildings: These methods are simplified versions, jointly developed by Air conditioning contractors of America (ACCA) and the Air conditioning and Refrigeration Institute (ARI).

These different methods may yield different results for the same input data. This is primarily due to the way; each method handles the solar effect and building dynamics. But in true sense all the above approaches attempt to consider the fundamental principle that heat flow rates are not instantaneously converted to loads and heat addition or extraction incident upon the building do not immediately result in a change in temperature. Thermally heavy buildings can effectively delay the cooling or heating load for several hours.

Most designers use the TETD and CLTD methods because these methods are simple to use, give component loads and tend to predict load on conservative side. The most recent versions of the ASHRAE Fundamentals Handbook (2001) provide more detailed discussion on the Radiant Time Series (RTS) and Heat Balance (HB) methods. The Heat Balance method is the most accurate but is very laborious and cumbersome and is more suitable with the use of computer programs. The RTS is a simplified method derived from heat balance (HB) method and effectively replaces all other simplified (non-heat balanced) methods.

For strictly manual cooling loads calculation method, the most practical to use is the CLTD/CLF method. This course discusses CLTD/CLF method in detail in succeeding sections.

A number of handbooks provide a good source of design information and criteria to use for CLTD/CLF method; however engineering judgment is required in the interpretation of various custom tables and applying appropriate correction factors. It is not the intent of this course to duplicate information but rather to provide a direction regarding the proper use or application of the available data so that the engineers and designers can make an appropriate decision.

PART 1 SUMMER COOLING LOAD

Preface

The term summer cooling load means much more than merely cooling the air in a building. In addition to cooling the air, it also implies controlling:

1) The relative humidity
2) Providing proper ventilation
3) Filtering out contaminants (air cleaning) and
4) Distributing the conditioned air to the lived-in spaces in proper amounts, without appreciable drafts or objectionable noise

This section deals with the design aspects and the equations used for summer cooling load calculations.

Design Conditions

The amount of cooling that has to be accomplished to keep buildings comfortable in hot summer depends on the desired condition indoors and on the outdoor conditions on a given day. These conditions are, respectively, termed the "indoor design condition" and the "outdoor design condition".

Indoor Design Conditions

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. As a general guideline for summer air-conditioning design, the thermal comfort chapter of the ASHRAE fundamentals handbook (Chapter 8, 2001) provides a snapshot of the psychrometric chart for the summer and winter comfort zones.

For most of the comfort systems, the recommended indoor temperature and relative humidity are:

1) Summer: 73 to 79°F; The load calculations are usually based at 75ºF dry bulb temperatures & 50% relative humidity
2) Winter: 70 to 72°F dry bulb temperatures, 20 - 30 % relative humidity

The standards were developed for mechanically conditioned buildings typically having overhead air distribution systems designed to maintain uniform temperature and ventilation conditions throughout the occupied space. The *Psychrometric* chapter of the *Fundamentals Handbook* (Chapter 6, 2001) provides more details on this aspect.

Outdoor Design Conditions

Outdoor design conditions are determined from published data for the specific location, based on weather bureau or airport records. Basic climatic and HVAC "design condition" data can be obtained from ASHRAE handbook, which provides climatic conditions for 1459 locations in the United States, Canada and around the world. The information includes values of dry-bulb, wet-bulb and dew-point temperature and wind speed with direction on percentage occurrence basis.

Design conditions for the United States appear in Table 1a and 1b, for Canada in Tables 2a and 2b, and the international locations in Tables 3a and 3b of 1997, ASHRAE fundamentals handbook chapter 26.

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

1) Dry bulb temperatures corresponding to 99.6% and 99% annual cumulative frequency of occurrence,
2) Wind speeds corresponding to 1%, 2.5% and 5% annual cumulative frequency of occurrence,
3) Wind direction most frequently occurring with 99.6% and 0.4% dry-bulb temperatures and
4) 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:

1) 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.
2) Wet-bulb temperature corresponding to 0.4%, 1.0% and 2.0% annual cumulative frequency of occurrence and the mean coincident dry-bulb temperature
3) 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).
4) 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 might 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.

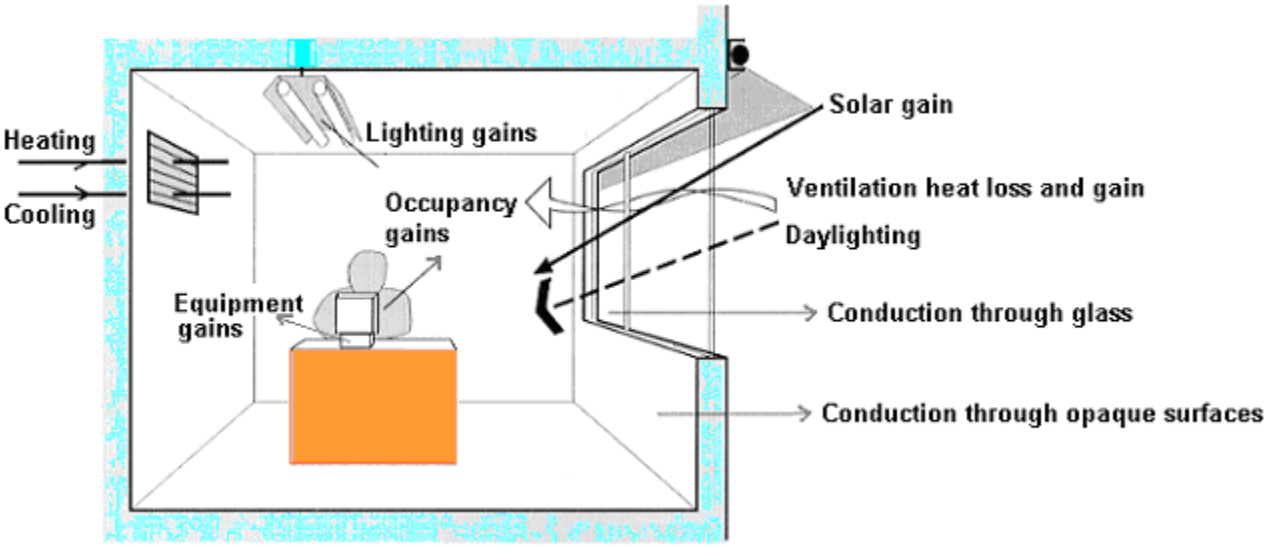
Design condition is used to calculate maximum heat gain and maximum heat loss of the building. For comfort cooling, use of the 2.5% occurrence and for heating use of 99% values is recommended.

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) or 2.5% of the time in a year, the outdoor air temperature will be above the design condition.

Cooling Loads Classified by Source

Cooling loads fall into the following categories, based on their sources:

1) Heat transfer (gain) through the building skin by conduction, as a result of the outdoor-indoor temperature difference.
2) Solar heat gains (radiation) through glass or other transparent materials.
3) Heat gains from ventilation air and/or infiltration of outside air.
4) Internal heat gains generated by occupants, lights, appliances, and machinery.



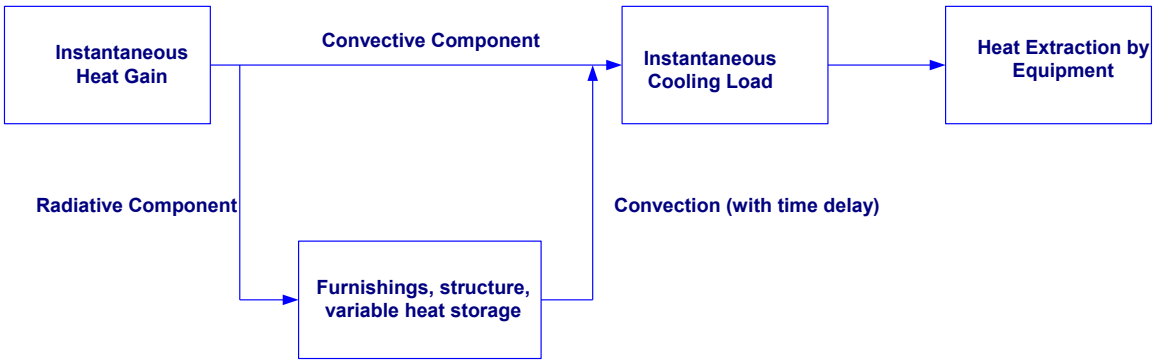
In cooling load calculation, there are four related heat flow terms; 1) *space heat gain, 2) space cooling load, 3) space heat extraction rate and 4) cooling coil load.*

What does these terms mean?

1) The ***heat gain*** for a building is a simultaneous summation of all external heat flows plus the heat flows generated inside the building. The heat gain varies throughout the 24 hours of the day, as the solar intensity, occupancy; lights, appliances etc keep varying with time.

2) The ***cooling load*** is an hourly rate at which heat must be removed from a building in order to hold the indoor air temperature at the design value. In other words, cooling load is the capacity of equipment required to account for such a load. Theoretically, it may seem logical to address that the space heat gain is equivalent to space cooling load but in practice "*Heat gain ≠ cooling load.*"

   The primary explanation for this difference is the time lag or thermal storage affects of the building elements. Heat gains that enter a building are absorbed/stored by surfaces enclosing the space (walls, floors and other interior elements) as well as objects within the space (furniture, curtains etc.) These elements radiates into the space even after the heat gain sources are no longer present. Therefore the time at which the space may realize the heat gain as a cooling load is considerably offset from the time the heat started to flow. This thermal storage effect is critical in determining the instantaneous heat gain and the cooling load of a space at a particular time. Calculating the nature and magnitude of these re-radiated loads to estimate a more realistic cooling load is described in the subsequent sections.



Schematic Relation of Heat Gain to Cooling Load

The convective heat flows are converted to space cooling load instantaneously whereas radiant loads tend to be partially stored in a building. The cooling load for the space is equal to the summation of all instantaneous heat

gain plus the radiant energy that has been absorbed by surfaces enclosing the space as well as objects within the space. Thus heat gain is often not equal to cooling load.

In heating load calculations however, the instantaneous heat loss from the space can be equated to the space-heating load and it can be use directly to size the heating equipment.

3) The ***space heat extraction rate*** is usually the same as the space-cooling load but with an assumption that the space temperature remains constant.

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

Cooling Loads Classified by Kinds of Heat

There are two distinct components of the air conditioning load; (1) the sensible load (heat gain) and (2) the latent load (water vapor gain).

Sensible Loads

*Sensible heat gain is the direct addition of heat to a space,which shall result in increase in space temperatures. The f*actors influencing sensible cooling load:

1) Solar heat gain through building envelope (exterior walls, glazing, skylights, roof, floors over crawl space)
2) Partitions (that separate spaces of different temperatures)
3) Ventilation air and air infiltration through cracks in the building, doors, and windows
4) People in the building
5) Equipment and appliances operated in the summer
6) Lights

Latent Loads

A latent heat gain is the heat contained in water vapor. Latent heat does not cause a temperature rise, but it constitutes a load on the cooling equipment. Latent load is the heat that must be removed to condense the moisture out of the air. The sources of latent heat gain are:

1) People (breathing)
2) Cooking equipment
3) Housekeeping, floor washing etc.
4) Appliances or machinery that evaporates water
5) Ventilation air and air infiltration through cracks in the building, doors, and windows

The total cooling load is the summation of sensible and latent loads.

Cooling Loads Classified by Inside-Outside Environment

Buildings can be classified as envelope-load-dominated and interior-load-dominated. The envelope heat flows are termed “external loads”, in that they originate with the external environment. The other loads are termed “internal loads”, in that they are generated from within the building itself. The percentage of external versus internal load varies with building type, site climate, and building design decisions. It is useful to identify whether internal or external loads will dominate a building, as this information should substantially change the focus of design efforts related to control and energy efficiency.

External Loads

External cooling loads consist of the following:

1) Sensible loads through opaque envelope assemblies (roofs, walls, floors)
2) Sensible loads through transparent or translucent envelope assemblies (skylights, windows, glazed openings)
3) Sensible loads through ventilation and infiltration (air leakage)
4) Latent loads through ventilation and infiltration.

Because of the inherent differences in these types of heat flows, they are calculated (estimated) using four different equations:

1) *Roofs, External Walls & Conduction through Glass*

   The equation used for sensible loads from the opaque elements such as walls, roof, partitions and the conduction through glass is:

   Q = U * A * (CLTD)

   U = Thermal Transmittance for roof or wall or glass. See 1997 ASHRAE Fundamentals, Chapter 24 or 2001 ASHRAE Fundamentals, chapter 25.

   A = area of roof, wall or glass calculated from building plans

   CLTD = Cooling Load Temperature Difference for roof, wall or glass. Refer 1997 ASHRAE Fundamentals, Chapter 28, tables 30, 31, 32, 33 and 34.

2) *Solar Load through Glass*

   The equation used for radiant sensible loads from the transparent/translucent elements such as window glass, skylights and plastic sheets is:

   Q = A * (SHGC) * (CLF)

   A = area of roof, wall or glass calculated from building plans

   SHGC = Solar Heat Gain Coefficient. See 1997 ASHRAE Fundamentals, Chapter 28, table 35

   CLF = Solar Cooling Load Factor. See 1997 ASHRAE Fundamentals, Chapter 28, and Table 36.

3) *Partitions, Ceilings & Floors*

   The equation used for sensible loads from the partitions, ceilings and floors:

   Q = U * A * (Ta - Trc)

   U = Thermal Transmittance for roof or wall or glass. See 1997 ASHRAE Fundamentals, Chapter 24 or 2001 ASHRAE Fundamentals, and Chapter 25.

   A = area of partition, ceiling or floor calculated from building plans

   Ta = Temperature of adjacent space (Note: If adjacent space is not conditioned and temperature is not available, use outdoor air temperature less 5°F)

   Trc = Inside design temperature of conditioned space (assumed constant)

4) *Ventilation & Infiltration Air*

   Ventilation air is the amount of outdoor air required to maintain Indoor Air Quality for the occupants (sees ASHRAE Standard 62 for minimum ventilation requirements) and makeup for air leaving the space due to equipment exhaust, exfiltration and pressurization.

   $Q_{sensible} = 1.08 * CFM * (T_o - T_c)$

   $Q_{latent} = 4840 * CFM * (W_o - W_c)$

   $Q_{total} = 4.5 * CFM * (h_o - h_c)$

   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)

Refer to 1997 ASHRAE Fundamentals, Chapter 25, for determining infiltration

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Internal Loads

Internal cooling loads consist of the following:

1) Sensible & latent loads due to people
2) Sensible loads due to lighting
3) Sensible loads due to power loads and motors (elevators, pumps, fans & other machinery)
4) Sensible & latent loads due to appliances

An internal load calculation is "the area of engineering judgment." The internal loads are sometimes about 60% of the load; however, these data are generally the least amount of information available to you at the design stage and therefore the generic rules are most often employed to fix the variables. The equations used in estimating internal loads are:

1) *People*

   $Q_{sensible} = N * (Q_S) * (CLF)$

   $Q_{latent} = N * (Q_L)$

   N = number of people in space.

   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.

2) *Lights*

   The lights result in sensible heat gain.

   $Q = 3.41 * W * F_{UT} * F_{BF} * (CLF)$

   W = Installed lamp watts input from electrical lighting plan or lighting load data

   F_{UT} = Lighting use factor, as appropriate

   F_{BF} = Blast factor allowance, 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.*

3) *Power Loads & Motors*

Three different equations are used under different scenarios:

   a. *Heat gain of power driven equipment and motor when both are located inside the space to be conditioned*

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

      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

b. *Heat gain of when driven equipment is located inside the space to be conditioned space and the motor is outside the space or air stream*

Q = 2545 * P * 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

F_{LM} = Motor load factor

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

c. *Heat gain of when driven equipment is located outside the space to be conditioned space and the motor is inside the space or air stream*

Q = 2545 * P * [(1.0-Eff)/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

F_{LM} = Motor load factor

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

4) *Appliances*

Q = 3.41 * W * F_u * F_r * (CLF)

W = Installed rating of appliances in watts. See 1997 ASHRAE Fundamentals, Chapter 28; 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.

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

Heat Gain from HVAC System

a. 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 * [P / (Eff_1 * Eff_2)]

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; Table 4 for motor heat gain.

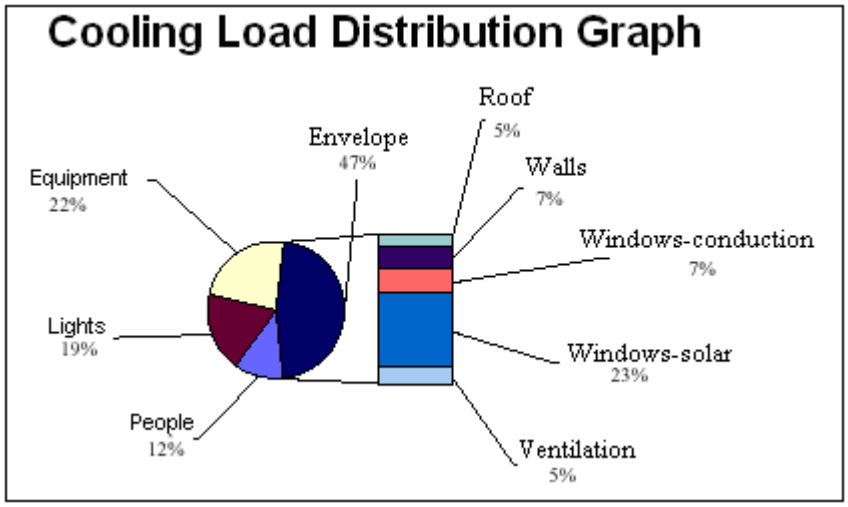
b. Duct Heat Gain

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.

TOTAL LOAD

The total load is the summation of external and internal load or both sensible and latent loads. Usually 10% safety margin is added but it all depends on how accurate are the inputs. The final load is than used to size the HVAC equipment. HVAC equipment is rated in Btuh, but is commonly expressed in tonnage. A Btu (British thermal unit) is the amount of heat needed to raise one pound of water one degree Fahrenheit. A "Ton" of cooling load is actually 12,000 Btu per hour heat extraction equipment. The term ton comes from the amount of cooling provided by two thousand pounds or one ton of ice.

Traditionally, cooling loads are calculated based on worst case scenarios. Cooling loads are calculated with all equipment & lights operating at or near nameplate values, occupant loads are assumed to be at a maximum, and the extreme outdoor conditions are assumed to prevail 24 hours per day. Real occupant loads are seldom as high as design loads. In detailed designing, the internal and external loads are individually analyzed, since the relative magnitude of these two loads have a bearing on equipment selection and controls. For example check the figure below:



Analysis of this breakup provides an idea of how much each component of the building envelope contributes to the overall cooling load and what can be done to reduce this load. Reducing solar heat gain through windows is clearly one of the key areas.

The architect must also be aware of the heat load equations and the calculation methodology as these influence the architectural design decisions that will in turn influence the energy consumption and comfort potential of the facility. The majority of these decisions are made--either explicitly or by default--during the architectural design process.

Information on each of the components of cooling load equations -- and the design decisions that lie behind these equations -- is covered in the subsequent sections.

PART 2 SOLAR HEAT GAIN THROUGH WALLS, ROOF, PARTITIONS

Preface

The equation used for sensible loads from the opaque elements such as walls, roof, partitions and the raised floors is:

Q = U * A * (CLTD)

U = Thermal transmittance for roof or wall or glass. See 1997 ASHRAE Fundamentals, Chapter 24 or 2001 ASHRAE Fundamentals, chapter 25.

A = area of roof, wall or glass calculated from building plans

CLTD = Cooling Load Temperature Difference for roof, wall or glass. Refer 1997 ASHRAE Fundamentals, Chapter 28, tables 30, 31, 32, 33 and 34.

This section deals with the interpretation of solar heat gain equations, tabulated information and the load reduction strategies through building envelope opaque surfaces.

Heat Gain through Building Walls

The heat flow through the wall is calculated by the following equation:

$Q = U \times Ax\ (T_i - T_o)$

OR

$$Q = \frac{A * (T_i - T_o)}{R_{Total}}$$

In the equation above:

1) Q describes 'Sensible heat flow' that affects HVAC equipment size and energy consumption. For overall economics and efficiency the goal is to minimize this value. Higher Q value shall impose high first- and recurring operation costs on HVAC system.

2) A is the area of wall, which is a function of building form. The area values are computed from building plans and elevations drawings.

3) U-value describes the rate of heat flow through a building element. It is the reciprocal of the R-value; $U = 1/R_{Total}$ where R_{Total} is the total resistance of the materials used in the construction of wall. The higher the R-value, the higher shall be the insulating value of the material or the lower the U-value the higher shall be the insulation value of the material.

   The units of the U-value are BTU/hr sq-ft °F. A maximum value of U-factor is often set by energy efficiency standards and is calculated from the material information shown in building drawings. The first step in estimating the heat transfer through wall is to determine the design heat transmission coefficient or overall thermal resistances of the various components that make up the envelope of the space. ASHRAE Fundamentals edition 2001, Chapter 25, or 1997 ASHRAE Fundamentals Chapter 24 provides detailed procedures and the thermal values which may be used to calculate the thermal resistances of building walls, floors, roofs and ceilings.

3) ΔT i.e. $(T_i - T_o)$ is the temperature difference between the inside design and outdoor temperature.

   The conductive heat gain equation $Q = U \times Ax\ (T_i - T_o)$ is OK for estimating heat loss in winter months but for summer months, the combined effect of convection, conduction, radiation and thermal (time) lag (for opaque surfaces) are to be considered. The time lag is the difference between the time of peak outside temperature and the time of the resulting indoor temperature. All of the transmitted solar radiation does not immediately act to increase the cooling load; some is absorbed by wall and is radiated back to indoor space even after the sunset. Therefore the time at which the space may realize the heat gain as a cooling load is considerably offset from the time the heat started to flow. This phenomenon is caller "thermal storage effect" and is dependent on the thermal mass of the material (the concept is discussed later in this section).

   The heat gain equation for cooling load is thus modified to include empirical value termed as 'Cooling load temperature difference (CLTD) that takes into account heat storage and time lag effects.

$Q = U \times Ax\ (CLTD)$

Or

$$Q = \frac{A * CLTD}{R_{Total}}$$

Where to get CLTD values?

The CLTD values can be found from tables listed in ASHRAE handbook of fundamentals.

The CLTD is determined by the type of wall (assembly construction) and is affected by thermal mass, indoor & outdoor temperatures, daily temperature range, orientation, tilt, month, day, hour, latitude, solar absorbance, wall facing direction and other variables. Corrections and adjustments are made if the conditions are different. Table- 1 & 2, below provide a snapshot of U-values, wall type & category (description & number), type of construction, mass of wall and CLTD values for Type-D wall construction at various wall facing directions and solar time.

Table -1
U-values for Walls

Wall description	Group Number	Description of construction	Weight, lb/ft²	U value, Btu/h·ft²·°F	Code numbers for layers
4-in face brick + brick	C	Airspace + 4-in face brick	83	0.358	A0 A2 B1 A2 E0
	D	4-in common brick	90	0.415	A0 A2 C4 E1 E0
	C	1-in insulation or airspace + 4-in common brick	90	0.174-0.301	A0 A2 C4 B1/B2 E1 E0
	B	2-in insulation + 4-in common brick	88	0.111	A0 A2 B3 C4 E1 E0
	B	8-in common brick	130	0.302	A0 A2 C9 E1 E0
	A	Insulation or airspace + 8-in common brick	130	0.154-0.243	A0 A2 C9 B1/B2 E1 E0
4-in face brick + (lightweight or heavyweight concrete block)	E	4-in block	62	0.319	A0 A2 C2 E1 E0
	D	Airspace or insulation + 4-in block	62	0.153-0.246	A0 A2 C2 B1/B2 E1 E0
	D	8-in block	70	0.274	A0 A2 C7 A6 E0
	C	Airspace or 1-in insulation + 6-in or 8-in block	73-89	0.221-0.275	A0 A2 B1 C7/C8 E1 E0
	B	2-in Insulation + 8-in block	89	0.096-0.107	A0 A2 B3 C7/C8 E1 E0
Heavyweight concrete wall + (finish)	E	4-in concrete	63	0.585	A0 A1 C5 E1 E0
	D	4-in concrete + 1-in or 2-in insulation	63	0.119-0.200	A0 A1 C5 B2/B3 E1 E0
	C	2-in insulation + 4-in concrete	63	0.119	A0 A1 B6 C5 E1 E0
	C	8-in concrete	109	0.49	A0 A1 C10 E1 E0
	B	8-in concrete + 1-in or 2-in insulation	110	0.115-0.187	A0 A1 C10 B5/B6 E1 E0
	A	2-in insulation + 8-in concrete	110	0.115	A0 A1 B3 C10 E1 E0
	B	12-in concrete	156	0.421	A0 A1 C11 E1 E0
	A	12-in concrete + insulation	156	0.113	A0 C11 B6 A6 E0
Metal curtain wall	G	With/without airspace + 1-in/2-in/3-in insulation	06-May	0.091-0.230	A0 A3 B5/B6/B12

Wall description	Group Number	Description of construction	Weight, lb/ft²	U value, Btu/h·ft²·°F	Code numbers for layers
					A3 E0
Frame wall	G	1-in to 3-in insulation	16	0.081-0.178	A0 A1 B1 B2/B3/B4 E1 E0

Table - 2

CLTD for Group D Walls

Wall facing	Solar time, h																							
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
N	15	13	12	10	9	7	6	6	6	6	6	7	8	10	12	13	15	17	18	19	19	19	18	16
NE	17	15	13	11	10	8	7	8	10	14	17	20	22	23	23	24	24	25	25	24	23	22	20	18
E	19	17	15	13	11	9	8	9	12	17	22	27	30	32	33	33	32	32	31	30	28	26	24	22
SE	20	17	15	13	11	10	8	8	10	13	17	22	26	29	31	32	32	32	31	30	28	26	24	22
S	19	17	15	13	11	9	8	7	6	6	7	9	12	16	20	24	27	29	29	29	27	26	24	22
SW	28	25	22	19	16	14	12	10	9	8	8	8	10	12	16	21	27	32	36	38	38	37	34	31
W	31	27	24	21	18	15	13	11	10	9	9	9	10	11	14	18	24	30	36	40	41	40	38	34
NW	25	22	19	17	14	12	10	9	8	7	7	8	9	10	12	14	18	22	27	31	32	32	30	27

*The CLTD values for all other wall groups can be referred in the 1997 ASHRAE *Fundamentals Handbook.*

Note that the values shown in the tables assume a mean of 85ºF (tm), a room temperature (t_r) of 78ºF, a daily range (DR) of 21ºF, dark surface, and a clear sky on the July 21. *When conditions are different,* CLTD values from the table must be corrected before being used in heat transfer equation. $CLTD_{Cor}$ can be found using equation as follows:

$CLTD_{Cor} = CLTD_{Table} + (78 - t_r) + (t_m - 85)$, where $t_m = (t_o + t_r) / 2 = t_o - (DR / 2)$

The revised equation $Q = U \times Ax\ CLTD_{cor}$ shall than be used for your specific project site application.

Heat Gain through Building Roof

The heat flow through the roof is calculated by the following equation:

$Q = U \times Ax\ (CLTD)$

Or

$$Q = \frac{A * CLTD}{R_{Total}}$$

Just like the case with the walls, the materials used in the construction of roof shall be used to calculate the resistance (R) and the U-value of the roof. The area of the roof is calculated from building plans. The materials parameters and the solar time shall than be used to determine the maximum CLTD value for the roof. Below is the reference Table-3 on typical CLTD values for a flat roof without suspended ceiling.

Table - 3

CLTD for Flat Roofs without Suspended Ceilings

Description of construction	Wt. lb/ft2	U value, Btu/hr·ft2·°F	Solar time

			1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
Steel sheet with 1" (or 2") insulation	7 (8)	0.213 (0.12)	1	-2	-3	-3	-5	-3	6	19	34	49	61	71	78	79	77	70	59	45
1" wood with 1" insulation	8	0.17	6	3	0	-1	-3	-3	-2	4	14	27	39	52	62	70	74	74	70	62
4" lightweight concrete	18	0.213	9	5	2	0	-2	-3	-3	1	9	20	32	44	55	64	70	73	71	66
2" heavyweight concrete with 1" (or 2") insulation	29	0.206 (0.12)	12	8	5	3	0	-1	-1	3	11	20	30	41	51	59	65	66	66	62
1" wood with 2 " insulation	9	0.109	3	0	-3	-4	-5	-7	-6	-3	5	16	27	39	49	57	63	64	62	57
6" lightweight concrete	24	0.158	22	17	13	9	6	3	1	1	3	7	15	23	33	43	51	58	62	64
2.5" wood with 1" insulation	13	0.13	29	24	20	16	13	10	7	6	6	9	13	20	27	34	42	48	53	55
8" lightweight concrete	21	0.126	35	30	26	22	18	14	11	9	7	7	9	13	19	25	33	39	46	50
4" heavyweight concrete with 1" (or 2") insulation	52 (52)	0.200 (0.12)	25	22	18	15	12	9	8	8	10	14	20	26	33	40	46	50	53	53
2.5" wood with 2" insulation	13	0.093	30	26	23	19	16	13	10	9	8	9	13	17	23	29	36	41	46	49
Roof terrace system	75	0.106	34	31	28	25	22	19	16	14	13	13	15	18	22	26	31	36	40	44
6" heavyweight concrete with 1" (or 2") insulation	75 (75)	0.192 (0.12)	31	28	25	22	20	17	15	14	14	16	18	22	26	31	36	40	43	45
4" wood with 1" (or 2") insulation	17 (18)	0.106 (0.07)	38	36	33	30	28	25	22	20	18	17	16	17	18	21	24	28	32	36

*The CLTD values for flat roof with suspended ceiling and other details can be referred in Table 5, p. 26.8, Chap 26, ASHRAE Fundamental Handbook, 1985 along with notes, limitations and adjustments)

Heat Gain through Exposed Floors & Slabs

The exposed floor is that portion of the building that has a vacant space below for example 2nd floor apartment building with open air parking garage below. Heat loss from floors to open spaces is treated as a wall or roof ($Q = U * A * \Delta T$).

Heat gain from crawl space floors and slabs- on- grade are typically neglected for summer cooling load but it's taken for heat loss during winter months. Slab-on-grade is defined as any portion of a slab floor in contact with the ground which is less than or equal to twenty-four inches below the final elevation of the nearest exterior grade.

Heat Gain through Interior Partitions & Ceilings

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

$q = U * A * (T_b - T_i)$

Where

1) A = area for interior partitions, ceilings or floors
2) U = overall heat transfer coefficient for interior partitions, ceilings or floors
3) T_b = average air temperature of the adjacent area
4) T_i = indoor air temperature

The load through ceiling shall not be added when the plenum (space above ceiling and roof) is used directly as the return plenum. If the return ducts are used than there will be temperature difference between the room air and the plenum and this load must be added.

SOL-AIR Temperature Concept

In the calculation of heat flow use is made of the concept of *sol-air temperature*, which is defined as; 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.

$$\boldsymbol{Q = h_{so} \times (T_{sol} - T_s)}$$

Where

$\boldsymbol{h_{so}}$ = heat transfer coefficient of the outer surface

$\mathbf{T_{sol}}$ = Sol-air temperature

$\mathbf{T_s}$ = temperature of outer surface

The heat gain due to the heat flow through the roof or wall is easily determined by using sol-air temperature. Both direct and diffuse solar radiations have a heating effect on the exterior surface of the wall causing higher surface temperature than outside air. Some of this heat will be stored, increasing the internal temperature of the wall. Due to temperature differential, some heat will be transferred by conduction to the cooler interior surface and then to the room, as heat gain. The process is dynamic because the exterior surface temperature is constantly changing as the angle of the sun changes. At certain times of the day and night, some of the stored heat will be transferred back to the exterior surface. Only part of the heat that enters the wall becomes cooling load, and this is delayed by storage effects. The greater the mass of the wall, the greater will be the delay.

The sol-air temperature derives an equivalent outside temperature, which is a function of time of day and orientation. The value is then adjusted for the storage effect and the time delay caused by

1) The mass of the wall or roof;
2) The daily temperature range, which has an effect on the storage;
3) The color of the outside surface, which affects the solar heat absorption rate; and
4) The latitude and month.

When the sol-air data is combined with the inside design temperature, a cooling load temperature difference (CLTD) is obtained. Then the cooling load is Q =U * A * CLTD.

The Sol-air temperature is a method of determining the heating effect of the outside surface due to solar irradiance and is substituted for the outside temperature in calculations for opaque surfaces i.e. T = Ts - Ti. The Sol-air temperature is calculated from:

Sol-air temperature Ts = To + ITHd x a/fo

Where:

1) To = outside temp
2) ITHd = solar irradiance
3) a = surface absorbance
4) fo =outside surface conductance

So, thermal conductance of solar energy through walls: $Q_c = A * U * (Ts-Ti)$

The sol-air temperature represents 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.

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

LOAD REDUCTION STRATEGIES

Interpreting the conductive heat transfer equation, Q = U * A * ΔT, a building construction with low U-value (air-to-air transmittance) will reduce all forms of conduction heat transfer through the building envelope. Insulating the walls with materials such as fiberglass, cellulose, Styrofoam, etc shall result in a much lower rate of heat transfer through the walls when the outdoor temperatures exceed the indoor temperature. But the added insulation shall also increase the retention of heat generated within the indoor space when the outdoor temperature fall below the indoor temperature.

Concept of Heat Capacity & Thermal Mass

Consider for example, an outside wall on a typical house that just happens to have no windows in it. The composite R-value of wall is probably something close to 14 (assuming R 11 insulation). The units of this R-value are Btu/sq-ft °F. This means that 1 Btu of heat will move across this wall for each sq-ft wall for every 14°F temperature difference between the outside air and the inside air. If the outside air was 40°F and the inside air was 68°F, two BTUs of heat would move. How fast this change occurs depends on the *heat capacity (thermal mass)* of the building structure and its contents.

Heat capacity in simple terms is a 'measure' of how much heat a material can hold. *Thermal mass in exterior walls will slow down heat flow through the wall allowing a reduction in insulation requirements.* Buildings constructed of concrete have a unique energy saving advantage because of their inherent thermal mass. These materials absorb energy slowly and hold it for much longer periods of time than do the low mass materials. This delays and reduces heat transfer across the wall. The thermal mass building yields three important results.

1) First, there are fewer spikes in the heating and cooling requirements, since mass slows the response time and moderates indoor temperature fluctuations.

2) Second, the high-mass materials reduce building annual heating and cooling requirements due to reduced heat transfer through the building envelope.

3) Third, thermal mass can shift energy demand to off peak time periods when utility rates are lower.

Buildings that most benefit from thermal mass are typically those with substantial cooling loads. In these cases, the thermal mass can be pre-cooled at night using outside air for free cooling or less expensive off-peak electricity for mechanical cooling. This allows the mass to absorb heat the following day, reducing the need for operation of cooling systems during peak utility demand hours.

In summary in a tropical environment, the thermal mass of the exterior wall could be used to intercept the heat and insulation placed on the inside of the thermal wall. In a colder, temperate environment, the insulation should be placed on the outside of the wall to keep heat inside. Generally, thermal mass is part of the integral construction of the building and is not added for conservation reasons. Unfortunately, there are no easy rules to determine how thermal mass will affect different buildings. It is important to note its existence because it may help you understand behavior of the mechanical systems or reasons for some comfort complaints.

Roof Insulation

A significant portion of a building's heat loss and heat gain occurs through its roof and walls. The most effective means of reducing the heat transfer rate is to maximize R-value by adding thermal insulation. Insulation serves to limit the conduction of heat through the building shell. For buildings with a large amount of roof area, such as a single-story retail facility, reducing heat gain through the roof can be an important consideration.

The type of insulation used and the cost effectiveness of adding more depends on the type of structure, the building's orientation, the amount of insulation already installed, venting of the space below the roof, and the color of the roof. Some of typical design considerations for roof insulation are:

1) For flat-roofed structures with rubber membrane roofing, it's common to use a rigid insulation board on the roof deck.

2) For structures with sloped roofs, batts or loose blown-in insulation are more common.

3) Where there is an attic or crawlspace below the roof, in most cases, roof insulation is applied to the attic floor, in the form of either fiberglass batts, blown-in loose insulation typically spray-on urethane or fiberglass foam. A building with an attic space can have insulation added at any time and in most cases, roof insulation is applied to the attic floor. Attic floor insulation is inappropriate in the presence of any type of water pipe in the attic space. Because the insulation reduces the heat flow from the occupied spaces upward to the attic, the attic space is at a lower temperature, increasing the likelihood that your pipes may freeze and burst during the heating season.

4) For buildings with un-vented attics or no attic, or in buildings where foot traffic might damage attic floor insulation, apply insulation to the inside roof surface, using either rigid board or spray on foam insulation.

5) For existing buildings, rigid board insulation, typically two inches thick, can be applied to the exterior surface of the roof prior to the application of the roof covering. This technique works well with new roof construction as well.

6) The greatest energy savings are typically made when adding insulation to a dark colored, flat, uninsulated roof directly over air-conditioned space.

The following table shows the insulating values of common types of insulation used:

Type	Form	R-value per inch
Polyurethane	Expanding Foam	6
Expanded Polystyrene	Rigid board	4 - 5
Polyisocyanurate, faced	Rigid board	7
Fiberglass	Batts or Loose	3.5
Cellulose	Loose or wet blown	3.5
Mineral Wool	Loose	2.5
Rock Wool	Loose	2.5

The effect of different roof insulation levels and color on annual heating and cooling costs is shown in the table below.

Insulation Level	Roof Color	Annual Cooling Costs ($ /sq-ft/year)
No Insulation, R-0	Dark	2.26
	Light	2.17
R-19	Dark	1.80
	Light	1.69
R-30	Dark	1.23
	Light	1.11

The following table provides some rules-of-thumb on the cost effectiveness of adding roof insulation to an existing building.

Existing Condition	Is it cost effective to add insulation?

No insulation to R-6	Yes, always
R-7 to R-19	Yes, if attic is accessible or if built-up roof is replaced
Greater than R-19	Not usually cost effective

These are general guidelines and it's always good to consult with your architect or builder about the cost-effectiveness of your application. For new buildings it is recommended to have a minimum R-30 roof insulation.

Roof & Wall Color

Lighter colors and reflective coatings reflect more of the sun's heat than darker colors. The color of a roof can affect the demand for cooling in buildings. The savings from applying a light colored or reflective roof treatment vary depending on the orientation of the roof, the ventilation of the space below it, and the roof's insulation levels. The greatest savings are expected on a flat roof with no ventilation below it and no insulation. Another method available for flat roofs is to apply a layer of white pebbles. In order to remain effective, a light or reflective roof coating or material needs to be kept clean. If it is allowed to become dark over time due to dirt, dust, and pollutants it's effectiveness will degrade.

The choice of the surface color is not of significant importance, the absorbance and surface conductance are. Surface conductance does not change by a large amount for different materials; however selection of a low absorbance material will reduce the solar heating effect.

In summary, the heat flow through the walls, roof and other parts of building envelope depends on the architectural aspects of building orientation, shape size, materials of construction, air tightness and other variables. Refer to section-10 for summary tips on architectural aspects and appendix B on the heat transfer through the building elements.

PART 3 HEAT TRANSFER THROUGH GLAZING –WINDOWS & SKYLIGHTS

Preface

Heat transfer through glazing is both conductive and transmission. It is calculated in two steps:

Step # 1

The equation used for sensible loads from the conduction through glass is:

Q = U * A * (CLTD)

U = Thermal Transmittance for roof or wall or glass. See 1997 ASHRAE Fundamentals, Chapter 24 or 2001 ASHRAE Fundamentals, chapter 25.

A = area of roof, wall or glass calculated from building plans

CLTD = Cooling Load Temperature Difference for glass. Refer 1997 ASHRAE Fundamentals, Chapter 28, tables 30, 31, 32, 33 and 34.

Step # 2

The equation used for radiant sensible loads from the transparent/translucent elements such as window glass, skylights and plastic sheets is:
Q = A * (SHGC) * (SC) * (CLF)

A = area of roof, wall or glass calculated from building plans

SHGC = Solar Heat Gain Coefficient. See 1997 ASHRAE Fundamentals, Chapter 28, table 35

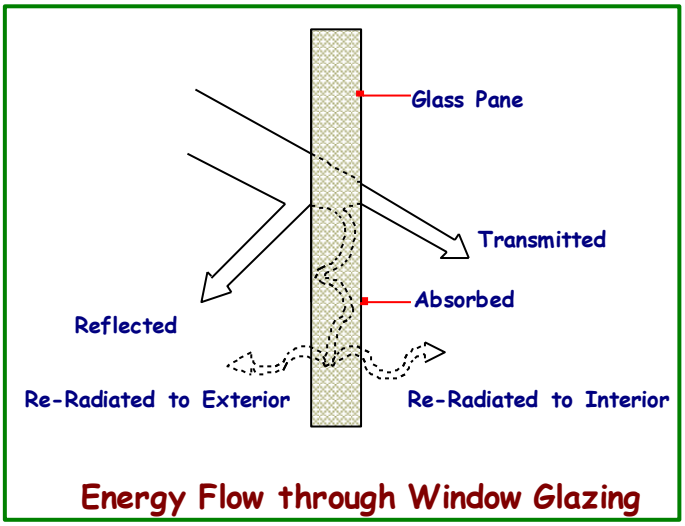
CLF = Solar Cooling Load Factor. See 1997 ASHRAE Fundamentals, Chapter 28, table 36.

Solar Heat Gain for Windows

Like walls, windows serve as separators between the uncontrolled outside environment and the controlled environment inside a building; but unlike walls, windows must also transmit light and is a poor insulator. Window represents the largest source of unwanted heat loss and heat gain in buildings. This is because even the best windows provide less insulation than the worst walls and because windows also admit solar radiation and represents a common source of air leakage. The net effect depends on a) the characteristics and orientation of the fenestration and b) the weather and solar radiation conditions. Minimizing energy transfer through the building would obviously encourage minimum glass for summer loads. *"Fenestration"* a term commonly used in architectural parlance refers to any glared aperture in the building envelope including glazing material, framing and shading devices.

But this is not to say we need to eliminate windows. Windows satisfy human needs of visual communication with the outside world, admit solar radiation to provide supplement daylight and enhance the exterior and interior appearance of a building. The thermal performance of windows can be thus be compared in terms of two indices 1) the light transmittance and 2) the heat transmittance.

An understanding of some basic energy concepts is essential to choosing appropriate glazing and skylights. Three major types of energy flow occur through windows, as shown in figure below:



When solar radiation strikes on glazing,

1) Part of the radiant energy is reflected back outdoors
2) Part of the radiant energy is absorbed within the glass
3) The remainder is transmitted directly indoor, and
4) The absorbed portion comes out again and flows either outward or inward.

For ordinary windows the absorption is quite a small fraction and transmission is the largest part. Window glass allows short-wave solar radiation get into an interior space. This radiation is absorbed by the interior of the building. The interior then radiates long-wave, thermal radiation. Glass is opaque (not transparent) to this long wave radiation. Thus energy is trapped in the building and the indoor air temperature rises. This is known as the “Green Effect.”

The heat admission through these fenestration areas is affected by:

1) Latitude, time of year, weather conditions, and orientation
2) Solar radiation intensity and incident angle,
3) Outdoor-indoor temperature difference,
4) Velocity and direction of air flow across the exterior and interior fenestration surfaces,
5) Low temperature radiation between the surfaces of the fenestration and the surroundings, and
6) Exterior and/or interior shading

SOLAR LOADS THROUGH GLAZING

Solar heat gain through glazing is a sum of the transmitted radiation and the portion of the absorbed radiation that flows inward. The solar heat gain is estimated by a two-step process. 1) For conduction and 2) the heat gain due to solar radiation transmitted through or absorbed by glass.

Step-1

The conduction heat flow for the windows is calculated closely to the same manner as described in for walls and roof.

Q = U * A * CLTD

Where:

1) Q describes ‘Sensible heat flow’ in Btu/hr
2) U is overall heat transfer coefficient for glazing and is inverse of the resistance of the glazing material. (Refer to table 1 below).

3) A is area of glazing including area of frame and is function of building design. Area is calculated from plans and elevations.

4) CLTD is the cooling load temperature difference for the window. (Refer to table 2 below).

Table- 1

Overall Heat Transfer Coefficients for Window Glasses

Window Glass	Overall heat transfer coefficient (U-values) in Btu/h.ft²°F							
	Summer (outdoor wind velocity = 3.33m/s)				Winter (outdoor wind velocity = 6.67m/s)			
	3 mm Thick	5 mm Thick	6 mm Thick	12 mm Thick	3 mm Thick	5 mm Thick	6 mm Thick	12 mm Thick
Single-glazed	0.95	0.92	0.88	0.76	1.07	1.00	0.95	0.81
Reflective			0.83				0.88	
Double-glazed 6mm airspace	0.56	0.53	0.51		0.55	0.51	0.49	
Double glazed 12mm airspace	0.49	0.48	0.46		0.48	0.46	0.42	

Table –2

Cooling Load Temperature Difference (CLTD) values through Window Glass

Solar time, hour	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
CLTD, °F	1	0	-1	-2	-2	-2	-2	0	2	4	7	9	12	13	14	14	13	12	10	8	6	4	3	2

Source ASHRAE, 1989 Fundamentals

Note: The values in the table were calculated for an inside temperature of 78F and an outdoor temperature of 95F with an outdoor daily range of 21F. The table remains approximately correct for other outdoor maximums of 93-102F and other outdoor daily ranges of 16-34F, provided the outdoor daily average temperature remains approximately 85F. If the room temperature is different from 78F and/or the outdoor daily average temperature is different from 85F, the following rules apply: a) For room air temperature is less than 78F, add the difference between 78F and room air temperature: if greater than 78F, subtract the difference. B) For outdoor daily average temperature less than 85F, subtract the difference; between 85F and the daily average temperature; if greater than 85F, add the difference.

Step- 2

Glass permits a portion of the solar radiation to be transmitted. This number is known as the Solar Heat Gain Coefficient (formerly the Shading Coefficient). Shading devices typically block the direct solar radiation, but not the diffuse.

The transmission solar heat gain through the windows is calculated separately for the different directions, the windows are facing. The heat flow through the window due to transmission is calculated by following equation:

$Q = A_s$ * max SHGF * SC * CLF

Where

1) Q describes 'Sensible heat flow'

2) A_s is the un-shaded area of window glass, plastic including area of frame and is function of building design. Area is calculated from plans and elevations

3) SHGC is the total solar heat transmission which includes both directly transmitted energy and indirectly transferred heat from energy initially absorbed by the glazing. Max SHGF is the maximum solar heat gain factor for window glass expressed in Btu/h/sq-ft. SHGF is affected by orientation, tilt, month, day, hour, and latitude. The SHGF value is a statistical data; available in ASHRAE fundamentals handbook. (Refer snapshot Table 3).

4) SC (shading coefficient) is a measure of the shading effectiveness of a glazing product. To determine shading coefficient one must first determine Solar Heat Gain Coefficient (SHGC). Shading Coefficient is the dimensionless ratio of the SHGC of a particular glazing compared to the SHGC of clear, double strength (DS or 1/8" thick) glass. As such, the shading coefficient of clear, double strength glass is 1.0, and glazing which block greater levels of solar energy than 1/8" clear will have lower SHGC values and therefore lower shading coefficients. In order to reduce the solar heat gain into a building interior, solar control glazing having a lower shading coefficient than clear glass should be utilized. Solar control glazing can be grouped into general groups such as: heat absorbing glass, reflective glass, and Low-E glass. The smaller the value of the shading coefficient the better the window is at stopping the entry of solar heat. (Refer to Table 4).

5) CLF is the cooling load factor (dimensionless). CLF factors are used to account for the fact that building thermal mass creates a time lag between heat generation from internal sources and the corresponding cooling load. CLF factors are presented in a set of tables that account for number of hours the heat has been on, thermal mass, type of furnishing or window shading, type of floor covering, number of walls, the room air circulation, the solar time, and the facing direction. The type of furnishings, the thickness of the slab, the room air circulation, the solar time, and the facing direction determine the cooling load factor (CLF). A CLF represents the fraction of the heat gain that is converted to cooling load. The cooling load factor is defined as:

$$\text{CLF} = \frac{\text{Sensible Cooling Load}}{\text{Sensible Heat Gain}}$$

CLF is used to determine solar loads or internal loads. Some CLF values are shown in Table 5.

Table 3

Maximum Solar Heat Gain Factor for Sunlit Glass on Average Cloudiness Days (Max SHGF)

Month	Maximum solar heat gain factor for 22 degree north latitude, Btu/ft²					
	North	North-east / North-west	East / west	South-east / South-west	South	Horizontal
January	26	41	180	230	203	205
February	28	77	205	221	169	236
March	31	118	217	193	116	257
April	35	150	210	151	61	262
May	41	167	200	118	41	260
June	53	172	194	104	39	257
July	43	165	196	114	41	256
August	36	146	203	145	65	257
September	33	113	206	186	114	249
October	29	76	197	214	164	231
November	26	41	177	229	200	204
December	25	29	169	231	213	192

* The SHGF is set by (a) the local latitude; (b) the date, hence the declination; (c) the time of day (solar time should be used); (d) the orientation of the window. Tabulated values of SHGF are given in the 1981 ASHRAE Handbook of Fundamentals, Chapter 27, for latitudes from 0 degrees (the equator) to 64 degrees N by 8 degrees increments and for orientations around the compass from N to NNW, by 22.5 degrees increments. Selected values from the 40 degrees table are given in an adjacent column.

Table 4

Shading Coefficient for Window Glasses with Indoor Shading Devices (SC)

Window glass	Nominal Thickness, mm	Solar transmission	Shading coefficient					
			Venetian		Roller shade, opaque		Draperies, light color	
			Medium	Light	Dark	White	Open	Close
Clear	3 - 12	0.78 - 0.79	0.64	0.55	0.59	0.25	0.65	0.45
Heat-absorbing	5 - 6	0.46	0.57	0.53	0.45	0.30	0.49	0.38
Heat-absorbing	10	0.34	0.54	0.52	0.40	0.28		
Reflective coated SCa=0.30 SCa=0.40 SCa=0.50 SCa=0.60			 0.25 0.33 0.42 0.50	 0.23 0.29 0.38 0.44			 0.23 0.33 0.41 0.49	 0.21 0.28 0.34 0.38
Insulating glass:								
Clear out-clear in SCa=0.84	6	0.80	0.57	0.51	0.60	0.25	0.56	0.42
Heat absorbing out-clear in SCa=0.55	6	0.56	0.39	0.36	0.40	0.22	0.43	0.35
Reflective SCa=0.20 SCa=0.30 SCa=0.40	6	0.80	 0.19 0.27 0.34	 0.18 0.26 0.33			 0.18 0.27 0.36	 0.16 0.25 0.29

* Values of the shading coefficient are given in Chapter 27 of the 1981 ASHRAE Handbook of Fundamentals for the most widely used glazing materials alone and in combination with internal and external shading devices. Selected values for single and double glazing are given in the "Shading Coefficient for Selected Glazing Systems" table.

Table 5

Cooling Load Factor for Window Glass with Indoor Shading Devices (CLF)

(North Latitude and All Room Construction)

Solar time, hour	1	2	3	4	5	6	7	8	9	10	11	12
Orientation:												
North	0.08	0.07	0.06	0.06	0.07	0.73	0.66	0.65	0.73	0.80	0.86	0.89
NE	0.03	0.02	0.02	0.02	0.02	0.56	0.76	0.74	0.58	0.37	0.29	0.27
East	0.03	0.02	0.02	0.02	0.02	0.47	0.72	0.80	0.76	0.62	0.41	0.27
SE	0.03	0.03	0.02	0.02	0.02	0.30	0.57	0.74	0.81	0.79	0.68	0.49
South	0.04	0.04	0.03	0.03	0.03	0.09	0.16	0.23	0.38	0.58	0.75	0.83
SW	0.05	0.05	0.04	0.04	0.03	0.07	0.11	0.14	0.16	0.19	0.22	0.38
West	0.05	0.05	0.04	0.04	0.03	0.06	0.09	0.11	0.13	0.15	0.16	0.17
NW	0.05	0.04	0.04	0.03	0.03	0.07	0.11	0.14	0.17	0.19	0.20	0.21
Horizontal	0.06	0.05	0.04	0.04	0.03	0.12	0.27	0.44	0.59	0.72	0.81	0.85

Table 5 Cont….

Cooling Load Factor for Window Glass with Indoor Shading Devices (CLF)

(North Latitude and All Room Construction)

Solar time, hour	13	14	15	16	17	18	19	20	21	22	23	24
Orientation:												
North	0.89	0.86	0.82	0.75	0.78	0.91	0.24	0.18	0.15	0.13	0.89	0.86
NE	0.26	0.24	0.22	0.20	0.16	0.12	0.06	0.05	0.04	0.04	0.26	0.24
East	0.24	0.22	0.20	0.17	0.14	0.11	0.06	0.05	0.05	0.04	0.24	0.22
SE	0.33	0.28	0.25	0.22	0.18	0.13	0.08	0.07	0.06	0.05	0.33	0.28
South	0.80	0.68	0.50	0.35	0.27	0.19	0.11	0.09	0.08	0.07	0.80	0.68
SW	0.59	0.75	0.81	0.81	0.69	0.45	0.16	0.12	0.10	0.09	0.59	0.75
West	0.31	0.53	0.72	0.82	0.81	0.61	0.16	0.12	0.10	0.08	0.31	0.53
NW	0.22	0.30	0.52	0.73	0.82	0.69	0.16	0.12	0.10	0.08	0.22	0.30
Horizontal	0.85	0.81	0.71	0.58	0.42	0.25	0.14	0.12	0.10	0.08	0.85	0.81

PERFORMANCE PARAMETERS

The key measures of window performance are a) Window Solar Heat Gain Coefficient (SHGC) or shading coefficient (SC); b) Glass Visible Transmittance and; c) Window U-value

When making a decision about purchasing windows, you should ask 3 questions:

a) How much heat are you trying to keep out?

b) How much daylight do you want to let in?

c) How much insulation must the glass assembly provide?

The first question is answered by looking at the Solar Heat Gain Coefficient (SHGC) — a figure that represents the percentage of heat transmitted through the glass. A SHGC of .55 equates to 55 percent of solar heat allowed to pass through the window into the building.

The second question is answered by visible transmittance (VT) - a figure that represents the amount of daylight that passes through glass. A VT of .7 means that 70 percent of outside light is transmitted through the window.

The third question is answered by the insulating value or U-factor- a figure that represents the rate of heat flow due to conduction, convection, and radiation through a window as a result of a temperature difference inside and outside. The lower the U-factor, the greater shall be the insulating properties of that assembly.

Solar Heat Gain Coefficient (SHGC)

The Solar Heat Gain Coefficient (SHGC) is the fraction of solar heat that enters the window and becomes heat. This includes both directly transmitted and absorbed solar radiation. SHGC is expressed as a number between 0 and 1. The higher the SHGC the more shall be the solar gain potential through a given window.

Facts about SHGC

1) The SHGC is a ratio between 0 and 1. SHGC = 0 means none of the incident solar gain is transmitted through the window as heat and SHGC = 1 means all of the incident solar energy is transmitted through the window as heat.

2) A window with a SHGC of 0.6 will admit twice as much solar heat gain as one with a SHGC of 0.3.

3) Typically, windows with low SHGC values are desirable in buildings with high air-conditioning loads while windows with high SHGC values are desirable in buildings where passive solar heating is needed.

4) The term "SHGC" is relatively new and is intended to replace the term "shading coefficient (SC)." While the terms are related, the shading coefficient of glass is defined as the ratio of the solar heat gain through a given glazing as compared to that of clear, 1/8 inch single pane glass.

5) Maximum SGHF is the maximum value of SHGF on the 21st day of each month for specific latitude. The maximum SGHFs are values on average cloudiness days. At high elevations and very clear days, the actual max. SGHF may

be 15% higher, and in very dusty industrial areas, they may be 20 to 30% lower. Since SHGF at a fixed window orientation varies through the year because of the continuous change of incident sunlight direction.

Recommendations on SHGC

1) In general, south facing windows designed for passive solar heating (with a roof overhang to shade them in the summer) should have windows with high a SHGC to allow in beneficial solar heat gain in the winter.
2) East or West facing windows that get a lot of undesirable sun in mornings and afternoons, and windows in hot climates, should have lower SHGC assemblies.
3) *Northern Climate Recommendation:* To reduce heating, select the highest SHGC you can find (usually 0.30-0.60 for the U-factor ranges required in colder climates) so that winter solar gains can offset a portion of the heating energy need. If cooling is a significant concern, select windows with a SHGC less than 0.55.
4) *Central Climate Recommendation:* If you have significant air conditioning costs or summer overheating problems, look for SHGC values of 0.40 or less. If you have moderate air conditioning requirements, select windows with a SHGC of 0.55 or less. While windows with lower SHGC values reduce summer cooling and overheating, they also reduce free winter solar heat gain.
5) *Southern Climate Recommendation:* A low SHGC is the most important window property in warm climates. Select windows with a SHGC less than 0.40.

Visible Transmittance (VT)

The visible transmittance (VT) is an optical property that indicates the amount of visible light transmitted. While VT theoretically varies between 0 and 1, most values are between 0.3 and 0.8. The higher the VT, the more light is transmitted. A high VT is desirable to maximize daylight.

Facts about VT

1) The visible transmittance (VT) refers to the percentage of the visible spectrum (380-720 nanometers) that is transmitted through the glazing.
2) Sunlight is an electromagnetic form of energy exchange between the sun and the earth. It is composed of a range of electromagnetic wavelengths, generally categorized as ultraviolet (UV), visible and infrared (IR) referred to collectively as the solar spectrum.
3) The short, UV wavelengths are largely invisible to the naked eye, but are responsible for fabric fading and skin damage. Visible light is made up of those wavelengths detectable by the human eye. This light contains about 47% of the energy in sunlight. Longer IR wavelengths are also invisible and contain about 46% of the energy in sunlight.
4) A typical clear, single-pane window has a VT of 0.90, meaning it admits 90% of the visible light.
5) For a given glazing system, the term "Coolness index (K_e)," also called Efficacy Factor is the ratio of the visible transmittance (VT) to the shading coefficient (SC).
6) The National Fenestration Rating Council (NFRC's) specifies VT for the whole window that includes the impact of the frame, which does not transmit any visible light.

Recommendations on VT

1) In general select windows with a higher VT to maximize daylight and view.
2) When daylight in a space is desirable, as in showrooms and studios, high VT glazing is a logical choice.
3) However, low VT glazing such as bronze, gray, or reflective-film windows are more logical for office buildings or where reducing interior glare is desirable.

Insulating Value (U-Factor)

Windows lose heat to the outside during the heating season and gain heat from the outside during the cooling season. The U-Factor is a measure of how easily heat travels through a material. The lower the U-value, the greater a window's resistance to heat flow and the better its insulating value. Some manufacturers rate thermal performance using R-Factors. R-Factor is the inverse of the U-Factor, i.e., 1/U = R or 1/R = U. For example: a U-Factor of 0.25 is the same as an R-Factor of 4.0.

Some typical U-Factor ranges for different window assemblies are:

1) Single glazed: 0.91 - 1.11
2) Double glazed: 0.43 - 0.57
3) Triple glazed: 0.15 - 0.33

Facts about U-Factor

1) The units of U-value are: Btu's per hour per square foot per °F (Btu/hr · ft² · °F)
2) U-factors usually range from a high of 1.3 (for a typical aluminum frame single glazed window) to a low of around 0.25 (for a multi-paned, high performance window with low-emissivity coatings and insulated frames). The smaller the U-factor, the less heat transfer between the inside and outside due to a temperature difference.
3) A window with a U-factor of 0.6 will lose twice as much heat under the same conditions as one with a U-factor of 0.3.
4) Total (or net) window U-factors can be considerably higher than the center-of-glass U-factors depending on the frame material.
5) The effects of non-solar heat flow are generally greater on heating needs than on cooling needs because indoor-outdoor temperature differences are greater during the heating season than during the cooling season. Therefore, for the heating season, the key rating parameter is the U-factor. U-factors allow consumers to compare the insulating properties of different windows and skylights.

Recommendations on U-Factor

1) ***Northern Climate Recommendation:*** Select windows with a U-factor of 0.35 or less. If air conditioning loads are minimal, windows with U-factors as high as 0.40 are also energy-efficient, if the SHGC is 0.50 or higher. Some double-glazed low-e products have U-factors below 0.30. Some three-layer products have U-factors as low as 0.15.
2) ***Central Climate Recommendation:*** Select windows with a U-factor of 0.40 or less. The larger is your heating bill, the more important a low U-factor becomes.
3) ***Southern Climate Recommendation:*** A low U-factor is useful during cold days when heating is needed. A low U-factor is also helpful during hot days when it is important to keep the heat out, but it is less important than SHGC in warm climates. Select windows with a U-factor lower than 0.75 and preferably lower than 0.60.

Representative Glass Specifications

Glass Type (Product)	Glass Thickness (Inches)	Visible Transmittance (% Daylight)	U-factor	Solar Heat Gain Coefficient (SHGC)
Single Pane glass (standard clear)	0.25	89	1.09	0.81
Single White Laminated w/Heat Rejecting Coating (Southwall California Series®)	0.25	73	1.06	0.46
Double Pane Insulated Glass (standard clear)	0.25	79	0.48	0.70
Double Bronze Reflective Glass (LOF Eclipse®)	0.25	21	0.48	0.35
Double glass, clear, $^1/_2$-inch air space		0.60		0.60
Double glass, bronze tint outer		0.45		0.49

Glass Type (Product)	Glass Thickness (Inches)	Visible Transmittance (% Daylight)	U-factor	Solar Heat Gain Coefficient (SHGC)
pane, 1/2-inch air space				
Double glass, green tint outer pane, 1/2-inch air space		0.55		0.48
Double glass, clear, E = 0.15*, 1/2-inch air space		0.54		0.50
Double glass, spectrally selective, E = 0.04*, 1/2-inch argon space		0.53		0.33
Triple Pane Insulated Glass (standard clear)	0.125	74	0.36	0.67
Pyrolitic Low-e Double Glass (LOF Clear Low-e®)	0.125	75	0.33	0.71
Soft-coat Low-e Double Glass w/Argon gas fill (PPG Sungate® 100 Clear)	0.25	73	0.26	0.57
High Efficiency Low-e (Solarscreen 2000 VEI-2M™)	0.25	70	0.29	0.37
Suspended Coated Film (Heat Mirror™ 66 Clear)	0.125	55	0.25	0.35
Suspended Coated Film w/ Argon gas fill (Azurlite® Heat Mirror SC75)	0.125	53	0.19	0.27
Double Suspended Coated Films w/ Krypton (Heat Mirror™ 77 Superglass)	0.125	55	0.10	0.34
Performance information was calculated using Lawrence Berkeley National Laboratory WINDOW 4/1 computer analysis program Azurlite® and Sungate® are registered trademarks of PPG Industries Heat Mirror™ and California Series® are trademarks of Southwall Technologies LOF Eclipse® is a registered trademark of Pilkington/Libby-Owens-Fort Co. Solarscreen 2000 VEI-2M™ is a trademark of Viracon				

Light-to-Solar Gain Ratio (LSG)

The ratio between SHGC and VT is called the light-to-solar gain ratio (LSG.) This provides a gauge of the relative efficiency of different glass types in transmitting daylight while blocking heat gains. *The higher the ratio number the brighter the room is without adding excessive amounts of heat.*

Here are typical values of SHGC, VT & LSG for the Total Window and for Center of Glass (shown in parenthesis) for different types of windows:

Window and Glazing Types	*SHGC*	*VT*	*LSG*
Single-glazed, clear	0.79 (0.86)	0.69 (0.90)	0.87 (1.04)
Double-glazed, clear	0.58 (0.76)	0.57 (0.81)	0.98 (1.07)
Double-glazed, bronze	0.48 (0.62)	0.43 (0.61)	0.89 (0.98)

Window and Glazing Types	*SHGC*	*VT*	*LSG*
Double-glazed, spectrally selective	0.31 (0.41)	0.51 (0.72)	1.65 (1.75)
Double-glazed, spectrally selective	0.26 (0.32)	0.31 (0.44)	1.19 (1.38)
Triple-glazed, new low-e	0.37 (0.49)	0.48 (0.68)	1.29 (1.39)

Air Leakage (AL)

Heat loss and gain occur by infiltration through cracks in the window assembly. It is indicated by an air leakage rating (AL) expressed as the equivalent cubic feet of air passing through a square foot of window area. The lower the AL, the less air will pass through cracks in the window assembly. The recommendation is to select the windows with an AL of 0.30 or less (units are CFM / sq-ft)

LOAD REDUCTION STRATEGIES

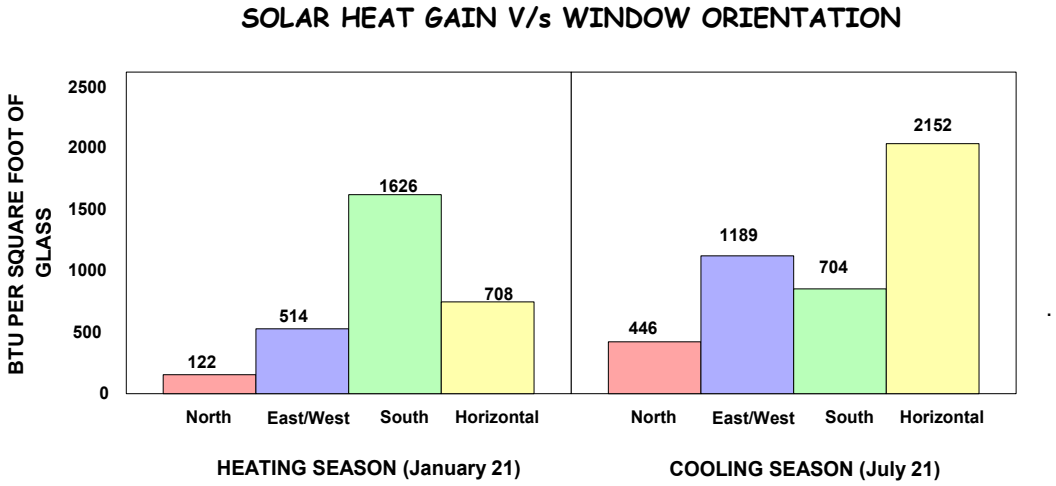
There are various methods available to the designer; among the few are:

1) Window Orientation (architectural aspect)
2) Type of Window Glasses
3) Special glass (Glazing Attributes)
4) Type of Window Frames
5) Window Dressing (Internal blinds/Roller Shades/Curtains)
6) Ventilation & Air tightness
7) External shading

Window Orientation

Solar transmission through windows and skylights can provide free heating during the heating season, but it can cause a building to overheat during the cooling season. Solar-induced cooling needs are generally greater than heating benefits in most regions. In fact, solar transmission through windows and skylights may account for 30% or more of the cooling requirements in residence/offices.

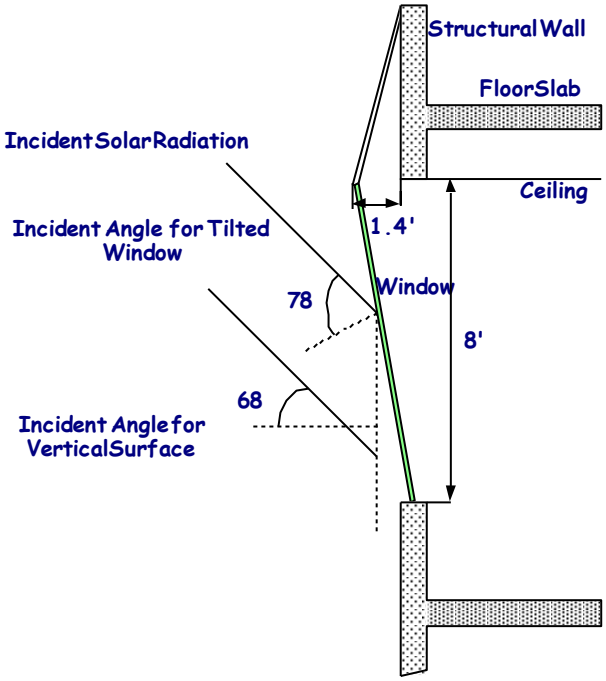
Because the position of the sun in the sky changes throughout the day and from season to season, window orientation has a strong bearing on solar heat gain. Figure below shows the solar heat gain through $^{1}/_{8}$-in clear single glass for various window orientations on very clear days in the heating and cooling seasons at 40deg. latitude (for example, Columbus, Ohio, and Boulder, Colorado).



Clearly, the windows facing south allow the greatest and potentially most beneficial solar heat gain during the heating season, while admitting relatively little of the solar heat that contributes to cooling requirements during the cooling season. The reverse is true for skylights and east- and west-facing windows. North exposures transmit only minimal solar heat at any time.

Window Tilting

The solar gain reduction can be controlled by tilting. Tilting the window glass can significantly reduce solar heat gain through a south facing window than for those facing east or west, since for the south facade the solar angle of incidence is large in summer and projections from the wall consequently cast long shadows. The energy falling on the window in this configuration is the same as would occur, if the window were vertical and had a 1.4 foot projecting shade along the lintel. The tilted glass reflects 45% of the radiation when the incident angle is 78° compared with 23% when the glass is vertical. This difference in reflectivity decreases as the season progresses toward the winter solstice, and in winter the tilted and vertical windows transmit essentially the same amount of solar energy.



TILTED WINDOW ARRANGEMENT

If shading or tilting are unacceptable for architectural reasons, heat absorbing glass can be used to advantage for south windows. Heat absorbing glasses can absorb of over 70% of the incident radiation so that transmission to the inside of a building is about 20%, when the angle of incidence is small and even less when it is large. Heat absorbing glass however has its best application; for east and west facing windows where the simple expedient of tilting has no appreciable effect.

Type of Glazing

In order to provide satisfactory visual effects, better solar heat control and thermal insulation, various types of window glazing is available. The options are:

1) ***Clear Plate or Sheet Glasses:*** These provide fine visual qualities and also a greater transmittance of solar radiation.

2) ***Tinted Heat Absorbing Glasses:*** These have bronze, grey and blue-green colors. Tinted heat absorbing glasses absorb a greater amount of infra-red radiation with some reduction of visible light.

3) ***Reflective Coated Glasses:*** These have a microscopically thin metallic layer of coating on one of the surfaces of the float glass. The reflecting coating provides reflecting characteristic in the infrared region with comparatively less

reduction of visible light transmission. Fritted glass uses baked ceramic coatings, or frits, can be applied to the surface of glass in many different patterns, colors, and densities.

4) ***Insulating Glasses:*** These are made of two or three pieces of glasses separated by metal or rubber spacer around the edge and sealed in a stainless steel structure. The dehydrated airspace between the glass panes is usually at a thickness of 6 to 12 mm, which enhances the thermal insulation of the unit. The spaces between window panes can be filled with gases that insulate better than air. Inert gases such as argon, krypton, sulfur hexafluoride, and carbon dioxide is usually injected to suppress convection and conduction through the air space in multiple glazed units. The gas fills reduce U-values without affecting shading coefficients or visible transmittance.

5) ***Switch able Optics-*** A new generation of dynamic glazing are available that change optical properties through changes in light, temperature, or voltage (i.e., photo-chromic, thermo-chromic, electro-chromic.) Currently, these materials are economically viable for niche markets only, such as two-way mirrors.

6) ***Low-emittance glazing-*** A low-e coating is a microscopically thin, virtually invisible, metal or metallic oxide layer deposited on a glazing surface. The coating limits radiative heat flow between panes by reflecting heat back into the home during cold weather and back to the outdoors during warm weather. This effect increases the insulating value of the window. Low-e coatings provide better insulation; reduce U-value, control heat gain and loss, reduce glare, minimize fabric fading, provide privacy and occasionally provide added security in wind, seismic and other high-hazard zones. The Energy Code prescribes low-e glazing with an emittance of less than 0.4.

Tints & Color

The properties of a given glass can be altered by tinting or by applying various coatings or films to the glass. Glass tints are generally the result of colorants added to the glass during production. The tints absorb a portion of the sunlight and solar heat before it can pass all the way through the window to the room. Tinting is the oldest of all the modern window technologies and, under favorable conditions, can reduce solar heat gain during the cooling season by 25% to 55%. Both glass and plastic laminate may be tinted. "Heat absorbing" tinted glass maximizes its absorption across some, or all, of the solar spectrum. Unfortunately, the absorbed energy often transfers by radiation and convection to the inside.

Tints are usually selected for aesthetic purposes but usually also help reduce solar gains.

Coatings & Films

The films are usually in form of metal oxides coatings, which transmit visible light while reflecting the long-wave infrared portion of sunlight. Films are thin layers of polyester, metallic coatings, and adhesives that save energy by limiting both the amount of solar radiation passing through the windows and the amount of internal heat escaping through windows. One layer of film is typically about 1/10,000 the diameter of a human hair. Coatings can be applied to glass during production or can be applied afterwards to the interior surfaces of all types of existing glass.

Window films typically cost between $1.35 and $3 per square foot, installed and generally last 7 to 12 years.

Most window manufacturers now use one or more layers of low-e coatings in their product lines. For buildings that primarily use daylight for lighting, a ‘spectrally selective window’ is a good choice. Spectrally selective coatings often have a light blue or green tint. These tints absorbs much of the ultraviolet (UV) portion of solar spectrum, reduce infrared light (heat) transmission while allowing relatively more visible light to pass through (compared to bronze- or gray-tinted glass).

Type of Window Frames

The insulating value of an entire window can be very different from that of the glazing alone. The whole-window U-factor includes the effects of the glazing, the frame, and, if present, the spacer. The spacer is the component in a window frame that separates glazing panes. It often reduces the insulating value at the glazing edges. A single-pane window with a metal frame has about the same overall U-factor as a single glass pane. With the wide use of multiple-pane, low-e, and gas-filled windows and skylights, frame and spacer properties have a more pronounced influence on the overall U-factors of windows and skylights.

Window frames can be made of aluminum, steel, wood, vinyl, fiberglass, or composites of these materials. Wood and vinyl frames are far better insulators than metal. Insulated fiberglass can perform better than either wood or vinyl. Advanced frame materials equipped with thermal breaks and low conductivity spacer elements improve energy performance. Some aluminum frames are designed with internal thermal breaks, non-metal components that reduce heat flow through the frame. These thermally broken aluminum frames can resist heat flow considerably better than aluminum frames without thermal breaks.

Frame geometry also strongly influences energy performance. Table below shows representative U-factors for window glazing, frame, and spacer combinations.

U-Factor (Btu/hr-ft^2-°F)

Glazing Type	Aluminum Frame without Thermal Break	Aluminum Frame with Thermal Break	Wood or Vinyl Frame with Insulated Spacer
Single glass	1.30	1.07	(n/a)
Double glass, $^1/_2$-inch air space	0.81	0.62	0.48
Double glass, E = 0.20*, $^1/_2$-inch air space	0.70	0.52	0.39
Double glass, E = 0.10*, $^1/_2$-inch air space	0.67	0.49	0.37
Double glass, E = 0.10*, $^1/_2$-inch argon space	0.64	0.46	0.34
Triple glass, E = 0.10 on two panes*, $^1/_2$-inch argon spaces	0.53	0.36	0.23
Quadruple glass, E = 0.10 on two panes*, $^1/_4$-inch krypton spaces	(n/a)	(n/a)	0.22
*E is the emittance of the low-e coated surface. Based on 3-ft-by-5-ft windows. U-factors vary somewhat with window size. Source: 1993 ASHRAE Handbook: Fundamentals			

Window Shading (Internal Blinds/Curtains etc)

Other ways to reduce the solar cooling load involve physical shading. Exterior and interior shading are among the best ways to keep the sun's heat out of a building. In warm climates, buildings in sunny areas can benefit greatly from a variety of shading techniques.

Interior Shading

Venetian blinds, roller blinds and other operable shades are a low-cost and effective solution for keeping out the sun. A light colored blind reflects some of the solar radiation and absorbs the rest. If the blind is in the room, most of the energy it absorbs is added to the room's cooling load.

More sophisticated systems, sometimes even located between two panes of window glazing. If it is between the panes of a double window, some of absorbed energy is transferred to the outside air and the room's cooling load is reduced accordingly.

Exterior Shading

External window shading devices such as awnings, roof overhang, shutters, solar screens, and internal shading devices such as curtains and blinds, can control the entry of solar heat. However, shutters, solar screens, curtains, and blinds make rooms dark. Curtains and blinds also let in some of the undesirable heat. While exterior shading devices are about 50% more effective than internal devices at blocking solar heat, they may create problems with the building's aesthetics and are sometimes expensive to build. It is also impractical to construct roof overhang to effectively shade east and west facing windows.

Properly applied, overhang and awnings can be particularly beneficial. During winter, when the sun is low in the sky, sunlight is beneficial for heating and lighting the inside of a space while the windows are not shaded. During summer, when the sun is high in the sky, overhang or awnings keep sunlight off the window. Awnings are popular exterior shading devices on low-rise commercial buildings. When selecting shading system colors, be sure to remember that light colors are better at reflecting solar radiation. A darker awning may require venting to allow heat dissipation.

An exterior roller blind is another effective method of exterior shading. Exterior roller blinds are a series of slats, typically horizontally oriented, made of wood, steel, aluminum, or vinyl. Like interior shades, they can be raised or lowered as needed to control the amount of sunlight entering a building space. In warm temperatures during sunny hours, they can be lowered to function as an insulating barrier, limiting incoming sunlight and reducing heat gain. Similarly, they can be raised in cold temperatures during sunny hours for desirable heat gain. Partially raising the blinds allows some daylight and air to enter between the slats.

The following are the percentages of the radiant energy that different types of internal shading devices transmit, reflect, or absorb:

1) Roller Shades: up to 25%, 15-80%, 20-65%
2) Vertical Blinds: 0%, 23%, 77%
3) Venetian Blinds: 5%, 40-60%, 35-55%

External shading with vegetation with natural deciduous trees is very effective at providing shade: during winter they are bare, allowing sunlight to pass through, but during summer they shade the building.

Ventilation and Air tightness

Airflow through and around windows occurs by design as ventilation and inadvertently as infiltration. The use of windows for natural ventilation is as old as architecture itself. Opening windows, particularly on opposite sides of a living space, can cool a home for free. The sash type of a window influences the ventilation airflow rate through the window relative to its size. Some common sash types and their effective open areas for ventilation purposes are shown in table below. Casement windows are especially effective for ventilation because they tend to direct the greatest airflow into the living space when fully open.

Infiltration is the leakage of air into a building from the exterior through joints and cracks around window and skylight frames, sash, and glazing. This leakage can account for 5% to 30% of the energy usage in a home. The air tightness of a window depends on the sash type as well as the overall quality of the window construction and installation. Because of the way they seal against the framing, windows with compressing seals are generally more airtight than purely sliding seals.

An air leakage rating is a measure of the rate of infiltration around a window or skylight in the presence of a strong wind. Air leakage ratings allow consumers to compare the air tightness of different windows and skylights as manufactured.

Representative Window Ventilation Areas

Sash Type	Effective Open Area*
Casement	90%
Awning	75%
Jalousie	75%
Hopper	45%
Horizontal sliding	45%
Single-hung	45%
Double-hung	45%

*The effects of window screens are not included

Source: R.K. Vieira and K.G. Sheinkopf, Energy-Efficient Florida Home Building, FSEC-GP-33-88, Florida Solar Energy Center, Cape Canaveral, FL, 1988.

In summary, it is easy to appreciate why a building designer should take great care to minimize solar heat gain when one considers the cost of the air-conditioning plant needed to remove it. The real cost of removing the heat that enters a building or lost through windows during heating season is so great that it is makes an economical decision to carefully evaluate all possible options during the conceptual stage. In nutshell, for summer cooling load SHGC should be as low as possible while for winter heating load the U-factors shall be as low as possible. (Refer to the section 10 for summary tips.)

PART 4 HEAT GAIN THROUGH LIGHTING FIXTURES

Preface

The lights result in sensible heat gain given by equation:

Q = 3.41 * W * FUT * FBF * (CLF)

W = Installed lamp watts input from electrical lighting plan or lighting load data

FUT = Lighting use factor, as appropriate

FBF = Blast factor allowance, as appropriate

CLF = Cooling Load Factor, by hour of occupancy. See 1997 ASHRAE Fundamentals, Chapter 28, and Table 38. Note: CLF = 1.0, if operation is 24 hours or if cooling is off at night or during weekends.

Heat Gain through Lights

The heat gain from lights is function of wattage (W) and 1 W = 3.414 Btus of heat. The heat gain from the light is calculated by:

*Q = 3.41 * Watts * FUT * FBF * (CLF)*

Where:

1) Q is a sensible heat gain in Btu/hr
2) Watts is the total light wattage obtained from the ratings of all fixtures
3) FUT is the diversity factor defined as the ratio of wattage in use at design condition to the installation condition
4) FBT is the special allowance factor for fluorescent fixtures accounting for ballast loss varying from 1.18 to 1.30
5) CLF is the cooling load factor for the lights. CLF is a function of
   - Number of hours that electric lights are switched on (for 24 hours continuous lighting, CLF = 1),
   - Types of building construction and furnishings

The lighting load is particularly affected by

1) The type of fixture
2) The type of furnishings provided in a room
3) The ventilation rate of the air-conditioning system
4) The placement of the fixtures and the type of HVAC system
   - A suspended fixture gives off heat through all three modes of heat transfer, radiation, convection, and conduction, and all of the heat remains in the space. The same is true for surface mounted fixtures.
   - Recessed fixtures give off half their heat above the ceiling and half below into the space. Some HVAC systems use the space between the suspended ceiling and the roof as the return air plenum. Recessed fixtures would increase the return air temperature and thus increase the cooling load.

The procedure to determine CLF involves a) finding the coefficient for your settings and b) estimating the CLF corresponding to the a-coefficient and b-classification from the tables below. The b-classification is same as we determined in group wall classification.

a-coefficients	Furnishings	Air supply and return	Type of light fixture
0.45	Heavyweight, simple furnishings, no carpet	Low rate; supply and return below ceiling (V* ≤ 0.5)	Recessed not vented
0.55	Ordinary furniture, no carpet	Medium or high ventilation rate; supply and return below ceiling or ceiling grill and space (V* ≥ 0.5)	Recessed not vented
0.65	Ordinary furniture, with or without carpet	Medium to high ventilation rates or fan coil or induction type air conditioning terminal unit; Supply through ceiling or wall diffuser; return	Vented

		around light fixtures and through ceiling space (V* ≥ 0.5)	
0.75 or greater	Any type of furniture	Ducted returns through light fixtures	Vented or free hanging in air stream

*V is room air supply rate in CFM per sq-ft of floor area

Cooling Load Factors when Lights are 'On' for 8h

a Coeff.	b Class-ifaction	Number of Hours after Lights are Turned On												
		0	1	2	3	4	5	6	7	8	9	10	11	12
0.45	A	.02	.46	.57	.65	.72	.77	.82	.85	.88	.46	.37	.30	.24
	B	.07	.51	.56	.61	.65	.68	.71	.74	.77	.34	.31	.28	.25
	C	.11	.55	.58	.60	.63	.65	.67	.69	.71	.28	.26	.25	.23
	D	.14	.58	.60	.61	.62	.63	.64	.65	.66	.22	.22	.21	.20
0.55	A	.01	.56	.65	.72	.77	.82	.85	.88	.90	.37	.30	.24	.19
	B	.06	.60	.64	.68	.71	.74	.76	.79	.81	.28	.25	.23	.20
	C	.09	.63	.66	.68	.70	.71	.73	.75	.76	.23	.21	.20	.19
	D	.11	.66	.67	.68	.69	.70	.71	.72	.72	.13	.18	.17	.17
0.65	A	.01	.66	.73	.78	.82	.86	.88	.91	.93	.29	.23	.19	.15
	B	.04	.69	.72	.75	.77	.80	.82	.84	.85	.22	.19	.18	.16
	C	.07	.72	.73	.75	.76	.78	.79	.80	.82	.18	.17	.16	.15
	D	.09	.73	.74	.75	.76	.77	.77	.78	.79	.14	.14	.13	.13
0.75	A	.01	.76	.80	.84	.87	.90	.92	.93	.95	.21	.17	.13	.11
	B	.03	.78	.80	.82	.84	.85	.87	.88	.89	.15	.14	.13	.11
	C	.05	.80	.81	.82	.83	.84	.85	.86	.87	.13	.12	.11	.10
	D	.06	.81	.82	.82	.83	.83	.84	.84	.85	.10	.10	.10	.09

**Refer to 1997 ASHRAE Fundamentals, Chapter 28, and Table 38 for further details. Note: CLF = 1.0, if operation is 24 hours or if cooling is off at night or during weekends.*

Estimating Lighting Loads

The lighting load is determined from the nameplate wattage from the lighting layout drawings. The actual lighting layout should be used whenever possible, however in practice most of time these nameplate ratings shall not be available during design phase and therefore an estimate can be made using the rule of thumb guidelines. Check the table below:

Location	Rule of thumb (Watts/sq-ft)
General Office Areas	1.5 to 3.0
Private	2.0 -5.0
Conference Rooms	2.0 – 6.0
Public Places (Banks, Post offices, Courts etc)	2.0 – 5.0
Precision Manufacturing	3.0 – 10.0
Computer Rooms/Data Processing Facilities	2.0 – 5.0
Restaurants	1.5 – 3.0
Kitchens	1.5 – 2.5
Pubs, Bars, Clubhouses, Taverns etc	1.5 – 2.0
Hospital Patient Rooms	1.0 – 2.0
Hospital General Areas	1.5 – 2.5
Medical /Dental Centers, Clinics	1.5 – 2.5
Residences	1.0 – 4.0
Hotel & Motels (public places and guest rooms)	1.0 – 3.0
School Classrooms	2.0 – 6.0
Dining halls, Lunch Rooms, Cafeterias	1.5 – 2.5
Library, Museums	1.0 – 3.0
Retail, Department & Pharmacist Stores	2.0 – 6.0
Jewelry Showrooms, Shoes, Boutiques etc	2.0 – 4.0
Shopping Malls	2.0 – 4.0
Auditoriums, Theaters	1.0 – 3.0

Location	Rule of thumb (Watts/sq-ft)
Religious Places (Churches)	1.0 – 3.0
Bowling Alleys	1.0 – 2.5

Note that the lighting values for most energy conscious construction will be lower values and can be very high if theatrical and fancy lighting is used. The cases which have not been mentioned above shall be based on the judgment of the most comparable use.

Heat Generated by Lights

As a general rule of thumb, any air-conditioned commercial or industrial building with a high percentage of incandescent lighting will find conversion to fluorescent fixtures more efficient. Only ~10 percent of the energy going into an incandescent light is converted to light, the rest of the energy is given off as heat. For a fluorescent light, twice as much of the energy is converted to light, but still this is only 20 % of the total energy.

This makes it easy to understand why improving the efficiency of the lighting can result in a 20 percent savings in cooling costs in an office building.

In manual method of calculations, the formula is simplified to calculate the cooling load for incandescent and fluorescent as:

Q_{light} = (3.413) * (watts) [Btu/hr] for incandescent lamps

Q_{light} = (3.413) * (watts) * (1.2) [Btu/hr] for fluorescent lamps

(The FUT, FBF and CLF is neglected, but with software's this is many a times accounted for)

A typical multiplier for fluorescents is 1.2 due to provision of ballast resulting in heat gain of 4.1 Btu per lighting watt. However the fluorescent lights provides more lighting as compared to incandescent and therefore as space shall require lower number of fixtures for same lighting levels. The table below can be used for estimating heat load from lights.

Installed Wattage (W)

	Illumination lux				
	200	400	600	800	1000
Incandescent lamp	38	75	110	145	180
Fluorescent tubes	15	25	36	48	60

Characteristics and Efficiency of Light Sources

A large percentage of the energy input into the lighting system shows up as heat. The efficiencies of different type of light sources can vary dramatically, so the choice of light sources can have a dramatic impact on lighting energy and the air-conditioning costs.

The performance and features of a light source must be matched to the lighting task being performed. In order to select the right light source for the job, you should consider important performance variables such as light output (Lumens), efficiency (Lumens/Watt – technically referred to as "efficacy"), lamp life, and color rendering properties measured in terms of the Color Rendering Index (CRI).

The CRI is a measure of the degree to which a light source renders colors that are close to true color. For practical purposes it is a number from 0 to 100; the higher the number, the closer to true color.

The term lighting efficacy measures the lamp's ability to convert input electric power into luminous power. Since electric power is measured in watts and luminous power in lumens, the unit of efficacy is lumen/watt. If there are auxiliary devices such as blasts, its energy consumption is charged towards the light source.

The performance characteristic of common light sources is shown below.

Type of Lamp	Lumens / Watt		Color Rendering Index	Typical Applications	Typical Life (hours)
	Range	Avg.			
Incandescent	8 - 18	14	Excellent	Homes, restaurants, general lighting, emergency lighting	1000
Fluorescent Lamps	45 - 60	50	Good w.r.t coating	Offices, shops, hospitals, homes	5000
Compact fluorescent Lamps (CFL)	40 - 70	60	Very gooc	Hotels, shops, homes, offices	8000 -10000
High pressure mercury (HPMV)	45 - 57	50	Fair	General lighting, industries, garages, parking areas, flood lighting	5000
Halogen lamps	18 - 24	20	Excellent	Display, flood lighting, stadium exhibition grounds, construction areas	2000-4000
High pressure sodium (HPSV) SON	67 -121	90	Fair	General lighting in factories, warehouses, street lighting	6000-12000
Low pressure sodium (LPSV) SOX	101 - 175	150	Poor	Roadways, tunnels, canals, street lighting	6000 -12000

Improvements in lighting efficiency shall decrease building cooling loads.

LOAD REDUCTION STRATEGIES

Each kilowatt hour of lighting requires 3,412 Btuh of air cooling. One ton of air conditioning is equivalent to heat extraction rate of 12,000 Btuh. Every 100 kilowatts removed during a lighting retrofit cuts cooling load by about 28 tons - enough capacity to cool an 8,000 square foot commercial space. Note the facts below:

1) *Installation of energy efficient fluorescent lamps (T8) in place of conventional fluorescent lamps (T12)*

   Energy efficient lamps are based on the highly sophisticated tri-phosphor fluorescent powder technology. They offer excellent color rendering properties in addition to the very high luminous efficacy.

2) *Installation of Compact Fluorescent Lamps (CFL's) in place of incandescent lamps*

   Compact fluorescent lamps are generally considered best for replacement of lower wattage incandescent lamps. These lamps have efficacy ranging from 55 to 65 lumens/watt. The average rated lamp life is 10,000 hours, which is 10 times longer than that of normal incandescent lamps. CFL's are highly suitable for places such as Living rooms, Hotel lounges, Bars, Restaurants, Pathways, Building entrances, Corridors, etc.

3) *Installation of metal halide lamps in place of mercury / sodium vapor lamps*

   Metal halide lamps provide high color rendering index when compared with mercury & sodium vapor lamps. These lamps offer efficient white light. Hence, metal halide is the choice for color critical applications where, higher

illumination levels are required. These lamps are highly suitable for applications such as assembly line, inspection areas, painting shops, etc. It is recommended to install metal halide lamps where color rendering is more critical.

4) *Installation of high frequency (HF) electronic ballasts in place of conventional ballasts*

   New high frequency (28-32 kHz) electronic ballasts have following advantages over the traditional magnetic ballasts a) Energy saving up to 35%, b) less heat dissipation, c) lights instantly, d) improved power factor, e) operates in low voltage load and f) increases the life of lamp

Additional facts

1) Compact fluorescent lamps last 10 to 13 times longer than the traditional incandescent light bulb, while consuming 75 percent less energy.
2) Fluorescent lamps produce approximately five times the lumens per watt of standard incandescent lamps.
3) T10 lamps produce 24% more light than a F40-T12 while consuming 3% more energy.
4) T8 lamps with a CRI of 85 produce 7% more light than a conventional T12 while consuming 16% less energy.
5) The high frequency electronic ballasts increase the efficacy of the lamps by 10 to 12 percent, compared to magnetic ballasts.
6) At average national electric rates, replacing a 150-watt incandescent lamp with a 35-watt metal halide lamp (drawing a total of 45 watts) would save about $30 a year.
7) Halogen lamps can save 60 watts per lamp in retail situations, while providing better lighting quality than incandescent lamps.
8) Increasing light levels can be achieved by upgrading the lens or louver, cleaning the luminaries, or adding reflectors.
9) Fixture efficiency can be significantly improved by replacing translucent diffusers or small-cell louvers with clear acrylic lenses or large-cell parabolic louvers. A facility lit with parabolic luminaries can lit at half the watts per square foot normally used in conventional lighting designs.
10) Decreasing light levels can be achieved by delamping, using lower wattage lamps, power reducers, partial output electronic ballasts, and dimming systems.
11) Use auto controller strategies such as occupancy sensors, motion sensors, photoelectric sensors, dimming devices etc for switching on-off the lighting fixtures.
12) When designing a new installation the lamp type selected should have as high an efficacy as possible and with characteristics that suit the requirements of the installation – color properties, appearance and rendering, life or service period, etc.

In general, the light should be brightest on your immediate work area, but should not over-illuminate. Lighting levels should decrease as you move into the general environment of the room.

PART 5 HEAT GAIN THROUGH POWER EQUIPMENT & APPLIANCES

Preface

Much of the electric use by the indoor electric equipment ultimately ends up in space as heat. The heat gain is given by following equation:

$Q = 3.41 * W * F_u * F_r * (CLF)$

W = Installed rating of appliances in watts. 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, and Tables 8, 9, & 10.

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

Note: The large power equipment is handled differently.

Heat Gain from Industrial Machines & Power Equipment

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 is in the room the heat transferred can be calculated as

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

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

Eff = Equipment 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 #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

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

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

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

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

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

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.

Heat Gain from Equipment

Heat gain from equipment; appliances, computers, printers, fax machines, TV, refrigerator, washing machines, video, vending machines, cleaning equipment and kitchen equipment etc is handled in a similar manner as lighting. The heat gain by the equipment is determined by the wattage of the equipment and is calculated by:

$Q = 3.41 * W * F_u * F_r * (CLF)$

Where:

1) Q is the sensible heat gain in Btu/hr
2) W is the wattage of the equipment
3) F_u = Usage factor.
4) F_r = Radiation factor.
5) CLF is the cooling load factor for the equipment and is a function of number of hours the equipments are on. (For 24 hours continuous running, CLF = 1). The CLF is found by the operational hours and the time the equipment is turned on.

Equipment and Appliances

The heat gain from electrical equipment is based on the data from tables 20 and 21 of chapter 26 of 1985 ASHRAE fundamentals handbook or chapter 28 of 1997 edition. Check the table below:

			Recommended Rate of Heat Gain, Btu-hr			
			Without Hood			With Hood
Appliances	**Input Rating Btu-hr**	**Probable Max. Input Btu-hr**	**Sensible**	**Latent**	**Total**	**All Sensible**
Cooking Range(gas type) Burner (Top) Oven	 64000 40000	 32000 20000	Hood Required			 6400 4000
Coffer brewer (240 cs/h)	17000	8500	6500	2026	8500	1700
Deep-fat fryer (14 lb)	18750	9400	2800	6600	9400	3000
Toaster (360 slices/hr)	7500	3700	2026	1740	3700	1200
Steam kettle, per gal cap	2026	1000	600	400	1000	320
Helmet type, hair dryer	2400		1870	330		2200
Instrument Sterilizer	3750		650	1200		1850
Bunsen Burner	3000		1680	420		2100

Heat Gain from Office Equipment

Computers, printers, fax machines and copiers consume energy even when these are not in use. With the widespread use of desktop computers, printers and other devices, the commercial establishment can have hundreds of units and the heat gain can just add up. The table below provides a sample of typical electrical power requirements for common office equipment. For details refer to *2001 ASHRAE Fundamentals, Chapter 29, and Tables 8, 9, & 10.*

Heat Gain Rates for Equipment - Watts (1 Watt = 3.41 Btu/h)

	Continuous	Average	Idle
Computer – 15" Monitor	110	-	20
- 17" Monitor	125	-	25
- 19" Monitor	135	-	30
Laser Printer - Desktop	130	100	10
- Small Office	320	160	70
- Large Office	550	275	125
Fax Machine		30	
Other Office Equipment		25% Nameplate (Watts ≈ volts×amps)	
Coffee Maker – 10 cup		1050 + 1540 Btu/h Latent	
Microwave Oven – 1 ft^3		400	
Refrigerator – 15 ft^3		300	
Water Cooler – 8 gal/hr		350	
¼ hp Motor	270		
¾ hp Motor	750		
1 hp motor	930		
10 hp Motor	8500		
(Watts=746×hp/η_{Motor})			

LOAD REDUCTION STRATEGIES

Reducing equipment heat gain offers very profitable opportunities for cooling load reductions and energy savings. Here are few tips

1) Always buy equipment with 'Energy Star' label. The energy use can be reduced by 50% or more. A typical non-Energy Star-compliant computer and color monitor draw a continuous electrical load of 150 watts or more (*ASHRAE Journal*, Sept. 1991).

2) Energy Star-labeled monitors automatically enter two successive low-power modes of 15 watts and 8 watts. In addition to reducing wasted energy, Energy Star-compliant monitors emit fewer electromagnetic fields in sleeping mode because most of their electronic components are turned off.

3) Typically, printers and fax machines are left on 24 hours a day, although they are active for only a small percentage of that time. To conserve energy, consider a combination printer/fax machine, which consumes half as much energy when idle as two stand-alone products.

4) Plug-in timers automatically turn equipment off at the power sources at certain times of day. They are especially useful for copiers and printers.

5) Laser printers consume more energy than inkjet printers.

6) Color printers use more energy than black and white.

7) Liquid crystal displays use less energy than conventional monitors.

8) Laptops draw about one-tenth the power of a conventional desktop computer. You can connect a laptop computer to a conventional monitor and still save almost half the energy of a standard computer.

9) If you use screen savers make sure they are compatible with your computer's power management features, as many will actually prevent your computer from going into the power-saver mode.

10) Power management features reduce energy consumption, but energy is still used in the power down mode. For greatest savings, turn equipment off at the power source when not in use overnight or on weekends. If the computer must be left on at night and weekends save energy costs by turning off the monitor. Monitors typically use more energy than the computer itself.

PART 6 HEAT GAIN DUE TO PEOPLE - OCCUPANCY LOADS

Preface

Human beings release both sensible heat and latent heat to the conditioned space when they stay in it. The space sensible ($Q_{sensible}$) and latent (Q_{latent}) cooling loads for people staying in a conditioned space are calculated as:

The heat gain from the occupancy or people is given be equation:

$Q_{sensible} = N * SHG * (CLF)$

$Q_{latent} = N * LHG$

N = number of people in space.

SHG, LHG = 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.

Heat Gain from Occupancy

You might have noticed that when a small room is filled with people, it tends to become warmer. People emit heat primarily through breathing and perspiration, and, to a lesser extent, through radiation. An average adult will generate 400 to 600 Btu of heat per hour. This heat translates into an increased cooling load on your cooling systems.

The heat gain by the occupants in the building is separated into sensible and latent heat. The number of people, the type of activity they are performing, and the CLF determines sensible and latent heat. The CLF is determined by the time the occupants come into the building and for how long they stay in the building. The sensible heat gain by the occupants is calculated by the following equation:

$Q_{sensible} = N * SHG * (CLF)$

Where:

1) $Q_{sensible}$ is the sensible heat gain in Btu/hr
2) N is the number of people
3) SHG is the sensible heat gain per person
4) CLF is the cooling load factor for the occupants. CLF for people is a function of a) the time people spending in the conditioned space, and b) the time elapsed since first entering. CLF is equal to 1 if the space temperature is not maintained constant during the 24-hour period.

The latent gain is assumed to immediately translate onto the cooling load and for this reason there is no CLF. The latent heat gain by the occupants is calculated by the following equation:

$Q_{latent} = N * LHG$

Where:

1) N is the number of people
2) LHG is the latent heat gain per person

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 slowly,	800	640	315	325
Light bench work	880	780	345	435
Light machine work, walking 3mi/hr	1040	1040	345	695
Moderate dancing	1360	1280	405	875

Activity	Total heat, Btu/h		Sensible heat, Btu/h	Latent heat, Btu/h
	Adult, male	Adjusted		
Heavy work, lifting	1600	1600	565	1035
Athletics	2026	1800	635	1165

The 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. Adjusted values for total heat along with SHG and LHG are shown in table below for normal percentage of men, women and children of which heat released from adult female is 85% of adult male, and that from child is 75%.

Adjusted total heat value for eating in a restaurant, includes 60 Btu/h for food per individual (30 Btu/h sensible and 30 Btu/h latent.)

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.

The entire sensible heat rate from people is not immediately converted in to cooling load because of thermal mass effects. The CLF correction must be applied and values are provided in the table below. However, the latent component is immediately converted to cooling load so no CLF correction is necessary.

Total hrs in Space	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
2	.49	.58	.17	.13	.10	.08	.07	.06	.05	.04	.04	.03	.03	.02	.02	.02	.02	.01	.01	.01	.01	.01	.01	.01
4	.49	.59	.66	.71	.27	.21	.16	.14	.11	.10	.08	.07	.06	.06	.05	.04	.04	.03	.03	.03	.02	.02	.02	.01
6	.50	.60	.67	.72	.76	.79	.34	.26	.21	.18	.15	.13	.11	.10	.08	.07	.06	.06	.05	.04	.04	.03	.03	.03
8	.51	.61	.67	.72	.76	.80	.82	.84	.38	.30	.25	.21	.18	.15	.13	.12	.10	.09	.08	.07	.06	.05	.05	.04
10	.53	.62	.69	.74	.77	.80	.83	.85	.87	.89	.42	.34	.28	.23	.20	.17	.15	.13	.11	.10	.09	.08	.07	.06
12	.55	.64	.70	.75	.79	.81	.84	.86	.88	.89	.91	.92	.45	.36	.30	.25	.21	.19	.16	.14	.12	.11	.09	.08
14	.58	.66	.72	.77	.80	.83	.85	.87	.89	.90	.91	.92	.93	.94	.47	.38	.31	.26	.23	.20	.17	.15	.13	.11
16	.62	.70	.75	.79	.82	.85	.87	.88	.90	.91	.92	.93	.94	.95	.95	.96	.49	.39	.33	.28	.24	.20	.18	.16
18	.66	.74	.79	.82	.85	.87	.89	.90	.92	.93	.94	.94	.95	.96	.96	.97	.97	.97	.50	.40	.33	.28	.24	.21

LOAD REDUCTION STRATEGIES

Traditionally, cooling loads are calculated based on worst case scenarios. Real occupant loads are seldom as high as design loads. Observations of conference rooms reveal that they are used about half of the time during which the building is occupied. The actual occupant load is usually about one third of the maximum seating capacity. An office plan may show a chair at every desk, as well as one or two other chairs for visitors. In most cases, the visitors' chairs will be empty most of the time. Analyze carefully.

PART 7 HEAT GAIN DUE TO VENTILATION & INFILTRATION

Preface

Ventilation air is the amount of outdoor air required to maintain Indoor Air Quality for the occupants (see ASHRAE Standard 62 for minimum ventilation requirements). The ventilation air is also required for positive pressurization of air-conditioned spaces and is used for makeup of air leaving the spaces due to forced exhaust and exfiltration due to opening of doors etc. The equation used for ventilation & infiltration air has two components 1) Sensible Heat and 2) Latent Heat

$Q_{sensible} = 1.08 * CFM * (T_o - T_c)$

$Q_{latent} = 4840 * CFM * (W_o - W_c)$

$Q_{total} = 4.5 * CFM * (h_o - h_c)$

Where

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

Heat Gain from Ventilation & Infiltration

An outside air requirement for maintaining occupant health and comfort vary depending on the type of facility and the level of occupancy. Ventilation loads are often driven by code and the designer's understanding of the maximum number of occupants that will be in any given space. Lacking occupancy information, designers may assume all chairs in an office or conference room are filled. When outside air enters a building, it has to be cooled or heated to maintain comfort. The higher is the unconditioned air entering the building, the greater the load on the heating and cooling system and the greater the cost.

Air can enter the building in three ways:

1) Intentionally via the HVAC system to provide fresh air to the occupants and the value is generally dictated by the codes (ASHRAE Ventilation Standard 62-1989)
2) Unintentionally through cracks and crevices in the building
3) Unintentionally through doors and windows as they are opened and closed throughout the day

Uncontrolled infiltration may not provide fresh air where needed, and it cannot be turned off when the building is unoccupied.

Building codes dictate minimum ventilation rates. Interestingly, code-mandated rates have changed significantly over the years in response to events and new understandings about the impact of outside air quantities on energy consumption and occupant comfort.

Prior to 1973, ASHRAE recommended a standard of 15 to 25 cubic feet per minute (CFM) of outside air per person. After the oil crisis of the mid 1970s, an energy conservation drive led to ASHRAE Standard 62-1981.

ASHRAE Standard 62-1981 redefined ventilation air requirements and allowable contaminant levels. Striking a compromise between indoor air quality concerns and energy consumption concerns, this standard recommended a minimum of 5 CFM of outside air per person in a nonsmoking environment and 20 to 35 CFM of outside air per person in a smoking environment. Partly because of the lower ventilation rates set in ASHRAE Standard 62-1981, complaints of discomfort and poor health, phenomena now referred to as Sick Building Syndrome and Building Related Illnesses, increased. This led ASHRAE to raise its ventilation requirements closer to its previous levels.

Depending on space use, the current ASHRAE Ventilation Standard 62-1989 now requires minimum ventilation rate of 20 CFM per occupant, while 35 to 60 CFM of outside air per occupant is required for designated smoking areas. The standard also specifies that maximum allowable CO_2 concentration should not exceed 1,000 ppm.

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1) Estimating Ventilation Rates by Actual Cooling Load

The volumetric flow rate of the air that is required must be determined for each person in the building. This can be based upon space use, ceiling height, the number of people expected to occupy that space, and whether smoking is permitted in the area. Required values air flow rates (CFM) for both commercial and residential structures are provided in the 1989 version of ASHRAE Standard 62, *Ventilation Air for Acceptable Indoor Air Quality*. These are then added to find the total airflow in cubic feet per minute (CFM). The mass flow rate of the air is than determined by the following equation:

$m = CFM * \rho_{air}$

Where:

CFM is the volumetric flow rate of the air in cubic feet per minute

ρ_{air} is the density of the air

Once this is determined the sensible heat flow rate of the ventilation is calculated by the following equation:

$Q = m * C_p * (T_o - T_i)$

Where:

a) m is the mass flow rate of the air

b) C_p is the specific heat of air

c) T_i is the inside temperature

d) T_o is the outside temperature

---

2) Estimating Ventilation Rates by Air-changes

Special ventilation criterion is available for commercial kitchens, dishwashing areas, bathrooms, swimming pools, health clubs, etc. The air-change method is mainly used for preliminary estimation of ventilation load. Usually, the air-change method is applied for determining the exhaust requirements, which varies from 2 CFM per sq-ft for toilets, restrooms, locker rooms to 6 to 10 CFM per sq-ft for hazardous areas such as laboratories, battery room etc. The mass flow rate of the make-up air required to compensate the forced exhaust is than calculated by the following equation:

CFM = Vol * ACH / 60

Or

$m = Vol * ACH * \rho_{air}$

Where:

a) CFM is cubic feet per minute of air flow rate

b) m is the mass flow rate of the air

c) Vol is the volume of the room

d) ACH is the estimated air changes per hour

e) ρ_{air} is the density of the air

Once the mass flow rate of the air is found then the sensible heat flow is calculated by the same equation:

$\boldsymbol{Q = m * C_p * (T_o - T_i)}$

Where:

a) m is the mass flow rate of the air

b) C_p is the specific heat of air

c) T_i is the inside temperature

d) T_o is the outside temperature

The ventilation is going to be a higher value for hazardous operations. The guidelines for the contaminated, hazardous and manufacturing areas having mandatory exhaust requirements are tabulated in ASHRAE standards.

Calculating Ventilation Loads

Empirically, the ventilation load is given by equation:

$Q_{sensible} = 1.08 * CFM * (To – Ti)$

$Q_{latent} = 4840 * CFM * (Wo – Wi)$

$Q_{total} = 4.5 * CFM * (ho – hi)$

Where,

1) CFM = Infiltration airflow rate

2) To, Ti = Outside/Inside dry bulb temperature, °F

3) Wo, Wi = Outside/Inside humidity ratio, lb (water) per lb (dry air)

4) ho, hi = Outside/Inside air enthalpy, Btu per lb (dry air)

Space cooling not only lowers the temperature, but usually also has to remove water vapor. This dehumidification effect is called the latent load and can be significant in comparison to sensible cooling since every pound of water removed requires about 1000 Btu. You can actually measure this water removal by collecting water from the drain pans in the air conditioning system and weighing the amount over a time period. The cooling load imposed by the ventilation system is the heat required to be removed from the ventilation air stream as it is conditioned from the outdoor air temperature and humidity to the temperature and humidity level of the indoor air. Obviously the ventilation load shall be high if the differential (To – Ti) is high and if the ventilation mass flow rate (CFM) is high.

Though the computation of differential (To – Ti) looks to be simple, it is not straight forward. The design comfort indoor temperature (Ti) is normally fixed at 75°F, the design value of outdoor temperature (To) vary for geographical locations and is usually taken as ‘design dry bulb with coincident wet bulb temperature’ of chapter 26 of ASHARE fundamentals.

The design outdoor weather conditions used to calculate the space loads would often not apply to the calculation of the design ventilation load. In many cases, the outdoor design conditions of "design wet bulb with coincident dry bulb" may dictate the ventilation load. Following is an example of four locations in the United States of quite differing climates. Assuming in all locations, indoor conditions at 75° F Dry bulb and 50% RH (h_i = 28.11 BTU/lb), the Δh in table below is the difference between the enthalpy of the outdoor air and the indoor air.

Design Conditions Comparison

Location	Design DB with Mean Coincident WB					Design WB with Mean Coincident DB			
	DB	WB	h	Δh		DB	WB	h	Δh
Los Angeles	85	64	29.10	0.99		78	70	33.95	5.84
Denver	93	60	26.20	0.09		81	65	29.88	1.77
St. Louis	95	76	39.29	11.18		90	79	42.40	14.29

Miami	91	77	40.33	12.22		87	80	43.51	15.04

Note that the enthalpy difference (Δh) is proportional to the ventilation load and in Los Angeles, St. Louis and Miami using the design wet bulb conditions increases the ventilation cooling loads by 49%, 28% and 23% respectively. Thus the computation of ventilation load should not be carried out blindly without judging the consequences of design wet bulb temperature. This is particularly more important for the facilities such as hospital operation theaters, smoking lounges, industrial clean rooms, electronic & pharmaceutical facilities, public gathering places and manufacturing units where large amount of fresh air is needed to ensure positive pressurization.

Infiltration

No building is 100% tight and all buildings allow some level of air flow through the building envelope. The infiltration is the uncontrolled airflow through cracks and openings in the building. It is very difficult to estimate the heat gain or loss through infiltration as there are numerous factors:

Windows leak air around the frames, doors leak air with increased traffic movement, walls are porous, metal panel walls are never airtight, there are numerous utility penetrations and the vertical air movement in multistoried buildings takes place through elevator shafts, stairwells, utility chases, ducts and other openings.

Wind will increase infiltration and tall buildings have a “stack” or “chimney” effect that draws air into the bottom of the building and forces it out the top. The chimney effect is related to variations in the air density due to temperature and height and is aggravated by wind. The effect is minor during warm weather but significant in winter.

Infiltration through door traffic is the main culprit. One of the accurate means of computing infiltration into an air-conditioned space is by means of the velocity of airflow through an open door. When the door of an air-conditioned space is opened, the difference in density between cold and warm air will create a pressure differential, causing cold air to flow out the bottom of the doorway and warm air to flow in the top. Velocities will vary from maximum at the top and bottom to zero in the center. The estimated average velocity in either half of the door is 100 feet per minute for a doorway 7' high at a 60^{O}F temperature differential. The velocity will vary as the square root of the height of the doorway and as the square root of the temperature difference.

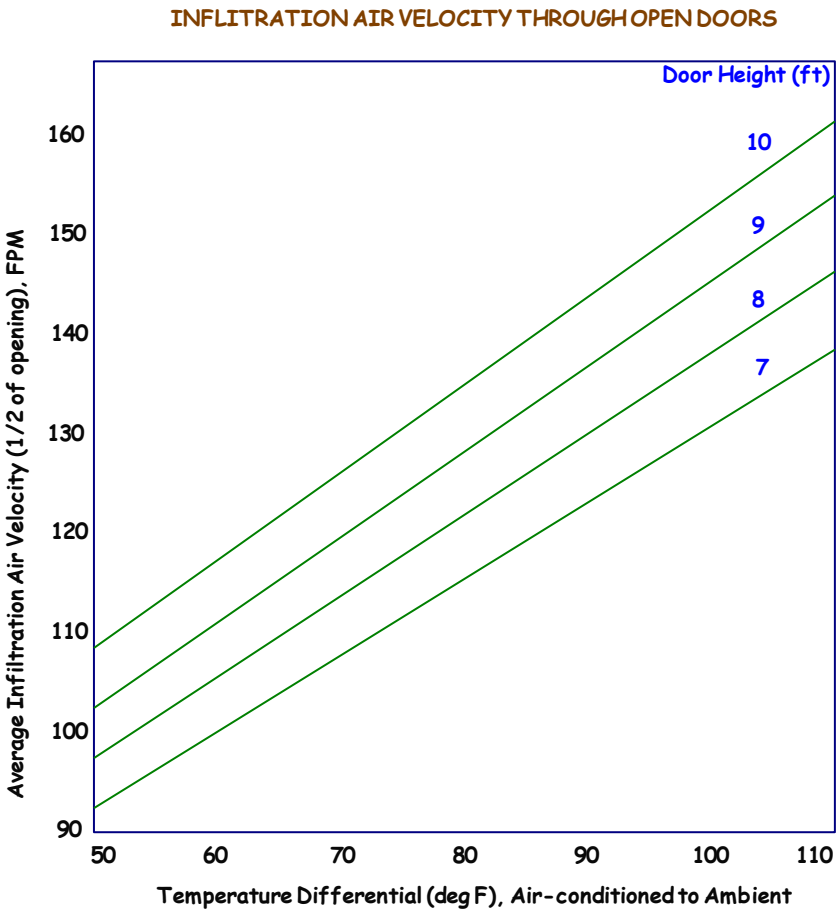
For example, the rate of infiltration through a door 8' high and 4' wide, with a 100^{O}F TD between the air-conditioned space and the ambient can be estimated as follows:

$$\text{Velocity} = 100\text{Fpm} \times \frac{\sqrt{8} \times \sqrt{100}}{\sqrt{7} \times \sqrt{60}}$$

$$= 138\,\text{fpm}$$

$$\text{Estimated Rate of Infiltration per door opening} = 138\ \text{fpm} \times \frac{8\ \text{ft} \times 4\ \text{ft}}{2} = 2208\text{CFM}$$

Infiltration velocities for various door heights and ΔT are plotted in figure below:



From the chart above, once the rate of infiltration has been determined, the heat load can then be calculated. The sensible component ($q = 1.08 \times cfm \times \Delta T$) and latent component ($q = 4840 \times cfm \times \Delta W$) are applied to infiltration load in similar manner as for controlled ventilation.

Makeup Air and Building Pressurization

The ventilation air is also used for positive pressurization of building particularly for clean room and hazardous areas applications. Positive pressurization is the basis of assuring that uncontrolled and untreated air does not infiltrate the protected area. The recommended minimum amount of positive pressurization gradient is 0.03" to 0.05" (~0.75 to 1.25mm) water column for clean room applications. This would normally equate to 3- 8% of gross room volume.

Optimizing Makeup Air Requirements

Careful attention needs to be paid 'not to' over-pressurize the area.

With pressurization, the requirement for make up air and the treatment costs due to cooling /dehumidifying and chemical filtration also increases. The cost of treating the make up air shall be very high, particularly for the extreme ambient environment conditions.

The amount of outside air required is a function of

1) Equipment exhausts and exhaust through toilets/kitchen/pantry/battery rooms etc.
2) Leakage through pass through, conveyor openings, strip curtains, air locks, door under cuts etc
3) Duct leakage, wall and ceiling leakages
4) Level of positive pressurization required

The HVAC design must optimize the use of make up air and shall minimize the uncontrolled air leakages while maintaining the controlled ventilation.

Impact on Energy Use

Over pressurization is waste of energy that not only entails high capital costs but also increases the operating costs. One-inch water gauge pressure is equivalent to wind velocity of 4005 feet per minute (~45 miles/hr).

The makeup air requirements depend on the level of positive pressure required in the room. High positive pressure requirement require high makeup air quantities. With higher pressurization the leakage velocity, leakage rates and the processing costs shall also increase.

Leakage through the fixed openings should be restricted as much as possible. The amount of expected leakage can be calculated from the following:

Leakage in CFM = $\sqrt{\text{Room Pressure in wg}}$ x 4005

Assuming 0.05" wg,

Leakage = 0.223 x 4005

= 895 feet per minute

With a total of 2 square feet opening size

Leakage = 2 x 895 = 1800 CFM

Higher positive pressure of say 0.1" wg (2.5 mm) shall mean higher velocity pressure of 1266 fpm (~6.4 m/s). The amount of leakage for 2 square feet opening shall be 2532 CFM an increase of 40%. Higher the velocity pressure higher shall be the ex-filtration or the leakages.

Assuming an ASHARE design condition of 95°F DB/72°F WB (~35°C DB/22° C WB) and room conditions of 72°F DB/60°F WB (~22°C DB/15.5°C WB, ~50% RH), the enthalpy difference is 9.5 BTU/lb (~22 kJ/Kg) of air.

For 1800 CFM leakage: this corresponds to energy loss of

= 1800 x 9.5 x 4.5

= 76950 BTU's/hr or 6.4 TR

For 2532CFM leakage: this corresponds to energy loss of

= 2532 x 9.5 x 4.5

= 108234 BTU's/hr or 9.0 TR

This is not only the extra capital cost but also the recurring energy costs of nearly 6 kWh @ 1kWh per TR (3.5 kW) of cooling load.

The room pressure should be limited to 0.03" to 0.05" (~0.75 to 1.25mm) as pressure above this is very inefficient (high energy and treatment costs on chemical filtration)

Air Tightness of Building Shell

Positive pressurization can be maintained only if the sealing integrity of the building is maintained. The building should be air tight for low air leakage performance. There are areas with in the facility that require negative exhausts such as toilets, pantry, laboratory or battery room but these are controlled ventilation areas having fixed amount of exhaust. Uncontrolled leakages areas in the building are door undercuts; pass through, walls, ceilings and duct joints etc; that should be restricted as far as possible. Remember a slogan;

"Build tight –ventilate right"

The building shall be optimally pressurized to achieve low capital costs, overall energy conservation and treatment costs on filtration.

LOAD REDUCTION STRATEGIES

1) Control ventilation rates to minimum requirements. The ventilation rate should be calculated based on the actual number of occupancy expected and not on air-changes.
2) Don't over pressurize. The recommended level of positive pressurization is 0.03" increasing to 0.05" for critical applications. This would normally equate to 3- 8% of gross room volume.
3) The mechanical exhaust systems should be interlocked to the fresh air supply systems.
4) Wherever possible, maximize return air re-circulation. Where not possible evaluate possibilities of recovering energy from the exhaust air through heat wheels or heat pipes etc.
5) Carbon dioxide-based demand- controlled ventilation systems vary the ventilation rate based on carbon dioxide (CO_2) levels in the building. For spaces with extreme variations in occupancy, such as banquet halls or meeting rooms, carbon dioxide sensors located in each zone adjacent to the room thermostat or in the common return air automatically control the amount of outside air. The controls are set such that the CO_2 level do not exceed ASHRAE permissible levels of 1000ppm.
6) Provide Time Clocks: Time clocks that automatically reduce ventilation rates during unoccupied periods can greatly reduce the energy load in buildings. If your building does not currently have night-time setback of the ventilation system, consider investing in time clocks.
7) Use of building automation system to utilize ambient air temperature variations is an effective means of utilizing free energy. At times, many buildings require air conditioning although the outside air is relatively cool and dry. During these times, increased amounts of outside air can reduce the cooling load.
8) In buildings with mechanical ventilation systems, it is desirable to minimize uncontrolled air leakage to reduce cooling loads. There are several methods to address unwanted infiltration:
    a) Caulking and weather-stripping should be in place for doors and windows.
    b) For open doorways (such as are often used at loading docks and warehouses), clear vinyl strips can be used.
    c) HVAC system outside air dampers should seal tightly when closed. Replacement with good quality opposed blade dampers with seals at the blade edges and ends will reduce infiltration.
    d) Exhaust hoods should be examined and adjusted to ensure they are exhausting the minimum air necessary to remove contaminants. Baffles can be added to the exhaust ducting or inside the hood to reduce flow.
    e) The orientation of the building, placement of trees and structures as windshields, and even the floor-to-floor open corridors will influence infiltration.

PART 8 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

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

1) CFM = air flow in cubic feet per minute
2) 1.08 = conversion constant = 0.244 X (60/13.5)
3) 0.244 = specific heat of moist air, Btu/lb of dry air
4) 13.5 = specific volume of moist air, cu-ft. per lb of dry air (@70°F, 50% RH)
5) $Q_{sensible}$ = total room sensible heat gain, BTU per hr.
6) T_R = Room dry bulb temperature, °F
7) T_S = Room supply air dry bulb temperature, °F (not necessarily the same as the temperature of the air leaving the cooling coil).

The selection of temperature differential ($T_R - T_S$) is typically routine and casual. But actually it is very critical because the real operating temperature differential is determined by the laws of "Psychrometrics" governing the performance of air system. With your basic knowledge of psychrometrics, consider an example below followed by a brief discussion on the subject.

Example:

Consider an interior core space of a building has a sensible load of 50000 Btu/h; a latent load of 10000 Btu/h and is conditioned by a single zone AHU. The inside conditions to be maintained are 75°F dry bulb and 50% RH and the design outside conditions are 95°F dry bulb & 75°F wet bulb. Assume the ventilation (outside) rates are 20%. Determine the air flow requirements and the supply air properties.

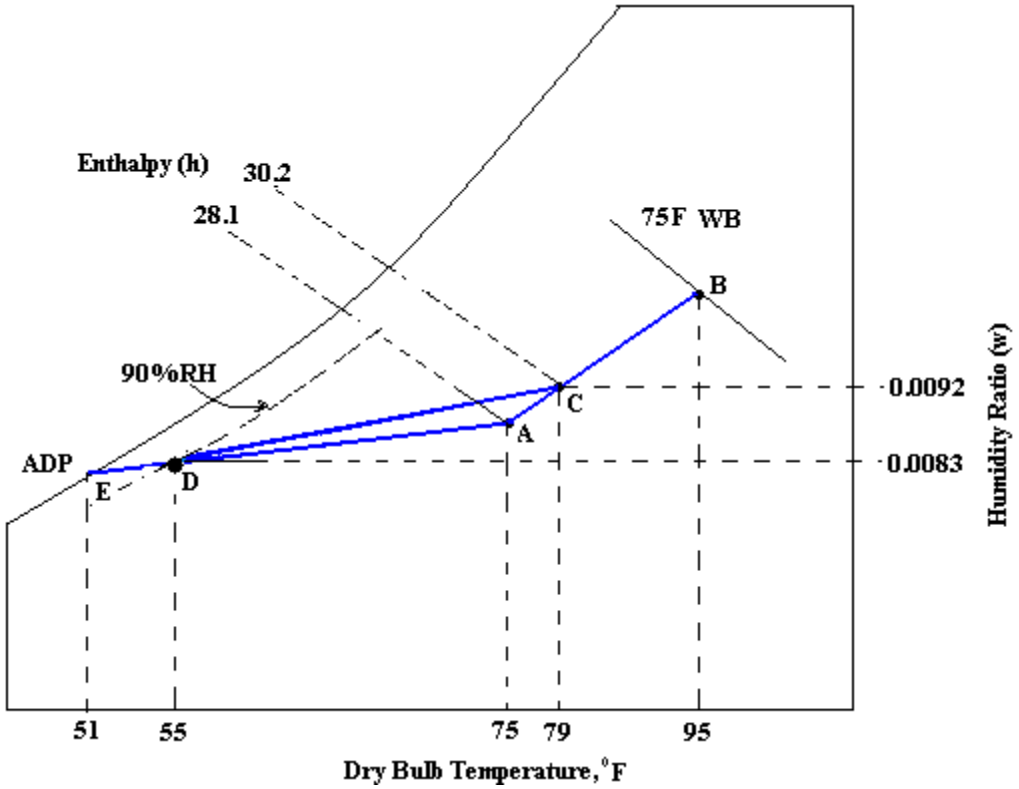
Solution:

Sensible load = 50000 Btu/h

Latent load = 10000 Btu/h

Sensible to total load ratio (S/T) = 50000/60000 = 0.833

In the psychrometric chart below, the returned room air at 75°F dry bulb and 50% RH (Condition A) is mixed with the required outdoor air at 95°F dry bulb & 75°F wet bulb (Condition B) at the air handling unit. This mixed air (Condition C) is passing through the cooling coil. Typically, the outdoor air condition is always warmer and more humid than the return air; therefore, the cooling process generally involves both cooling and dehumidification, with the conditioned air leaving the cooling coil at Condition D. Check the figure below for steps and analysis.



From Psychrometric Chart determine:

1) Locate indoor air conditions (75°F dry bulb & 50% RH) as point A. The room air enthalpy at this point is 28.1 Btu/lb of dry air and the room air specific humidity (w) is 0.0092 lb moisture/lb of dry air.
2) Locate outdoor air conditions (95°F dry bulb & 75°F wet bulb) as point B
3) Determine the mixed-air condition by ratio proportion as 79°F dry bulb and 65.3°F wet bulb with an enthalpy of 30.2 Btu/lb of dry air. Locate this point as C.
4) If the psychrometric chart include a 'protractor", (as in ASHRAE chart) a line may be drawn through the room state point with a slope equal to the sensible / total (S/T) ratio (0.833). In the figure below this line is DA. Theoretically, the supply air point may be anywhere on this line.
5) Determine Air flow rates: On the psychrometric chart, the slope of the supply air process line may also be determined by assuming a temperature differential ($T_R - T_S$) and calculating the resulting Δw. Assuming a temperature differential of 20°F, so that the supply point 'D' is at 55°F. Than the air flow rate shall be 50000/ (20 x 1.08) = 2315 CFM.
6) Determine specific humidity differential (Δw) using the calculated supply air flow rate:
   10000 / (2315 x 60 x 0.075 x 1059) = 0.0009
   Where
   a) 60 = min/hr
   b) 0.075 = air density, lb/ft3 (saturated air)
   c) 1059 = latent heat of vaporization at 60F, Btu/lb
7) The Δw of 0.0009 subtracted from the room w of 0.0092 equals 0.0083, the needed w of the supply air at point D. Corresponding to the point S, locate the supply air properties as 55°F dry bulb, 53.3°F wet bulb, h = 22.2, w = 0.0083 and RH = 90%. By projecting to the saturation line, the apparatus dewpoint (ADP) is 51°F.

Analysis:

This figure of 90% RH presents a problem because it implies a coil bypass factor of about 14% (refer to Psychrometric Analysis- Coil discussion below). This condition indicates that if the supply air temperature is controlled at 55°F, the resulting room condition will be somewhat higher humidity than the designed value. With chilled water system, the ADP of 51°F shall require a supply water temperature of about 45°F. At part load conditions the throttling of chilled water flow by two-way control valve allows the bypass factor to increase and supply air temperature to modulate.
With a direct expansion (DX) cooling coil, the ADP will tend to be between 40 and 45°F, which will pull the humidity downwards, increasing the load due to dehumidification of outside air. It will lower the supply air temperature so that

temperature differential will be 25°F or more. The air flow rate shall be 1850 CFM. This could result in cold drafts; rapid cycling of the refrigeration system and the associated control problems.
As a sanity check on the supply air values, use the following thumb of rules:

1) Supply air values below 0.7 CFM per sq-ft should be viewed as caution and should not be allowed. Lower supply air values may create distribution and balancing problems along with the complaints of stuffiness and high humidity.
2) For a variable volume (VAV) system allow for minimum supply air rates of 0.75 to 1.0 CFM per sq-ft of floor area.
3) Values above 3 CFM per sq-ft can create distribution problems, with high velocity drafts on the occupants. Recheck the calculations and if check reveals that these are required, special attention needs to be paid to the air distribution techniques.
4) Clean room application requires typically very high supply rates in range of 8 to as high as 20 CFM per sq-ft.

Psychrometric Analysis

Space design condition

When a design engineer performs a cooling load calculation, one of the first things that must be done is to set the indoor design condition. For most of the comfort environments, this condition is generally 75 F dry bulb (DB) and 50 percent RH, or 62.5 F wet bulb (WB) temperature.1 While this condition is a design goal, in reality the space dry bulb temperature is the only parameter that is maintained by a room thermostat setting. The RH or WB of the conditioned space, on the other hand, is not controlled unless specifically addressed in the system design. Extra air conditioning system apparatus and/or controls will be needed if the room RH or WB is also to be maintained. (This may entail the use of reheat coils and/or humidifiers. For very low RH requirements, desiccant dehumidification may be required.)

On the Psychrometric Chart

In a typical air conditioning system (Figure below), the returned room air (Condition A) is mixed with the required outdoor air (Condition B) at the air handling unit. This mixed air (Condition C) is passing through the cooling coil. In the conventional or typical case, the outdoor air condition is always warmer and more humid than the return air. Therefore, the cooling process generally involves both cooling and dehumidification, with the conditioned air leaving the cooling coil at Condition D.

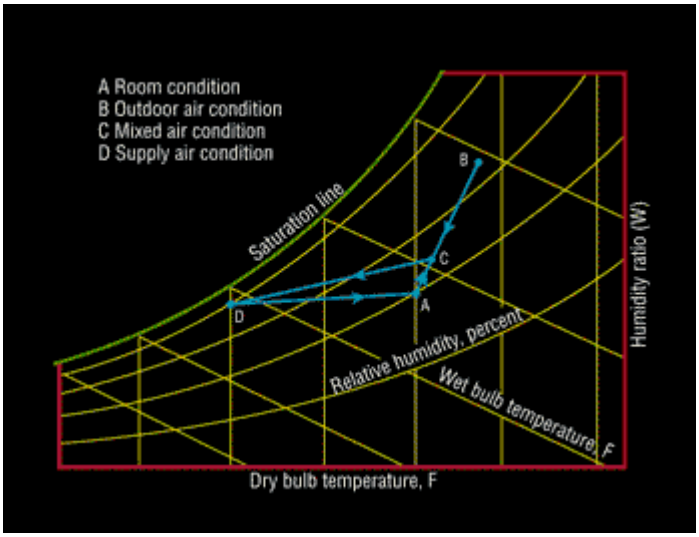


Fig- Psychrometric process of a simple air conditioning system

This cooled and dehumidified air leaving the coil at D is then supplied into the conditioned space at the room condition (A) to complete the cycle. The slope of the process line D--A is determined by the room sensible heat factor (RSHF), which is influenced by the amount of latent load in the conditioned space. While the dry bulb temperature in the conditioned space is controlled by a room thermostat, the humidity ratio in the conditioned space is a function of the cooling coil leaving condition and the room sensible heat factor. Note that the humidity ratio in the space is not under control of the air conditioning system.

Coil leaving condition

The RH of the coil leaving condition is a function of both physical and operating characteristics of the cooling coil. Decreasing the coil heat-transfer surface--*i.e.*, fewer rows, fewer fins per inch, or increasing the air velocity through the coil--will result in a lower RH coil leaving condition. Conversely, a coil having a larger heat-transfer surface and/or lower face velocity will have a coil leaving air condition closer to the saturation line--*i.e.*, a higher RH.

Because the leaving condition of a cooling and dehumidification process generally falls near the saturation line, it is often, by convenience only, assumed that the leaving RH is at 90 percent. While this assumption is generally adequate for illustrating cooling coil operation, it is definitely erroneous to use this assumed RH for determining the cooling capacity of the coil.

For a cooling and dehumidification process, one can determine the coil leaving condition using the coil bypass factor. The bypass factor is defined as the fraction of the incoming air that passes through the cooling apparatus completely unaltered, with the balance of the supply air completely saturated at the apparatus dew point (Point E in figure below). The bypass factor typically is expressed as a decimal.

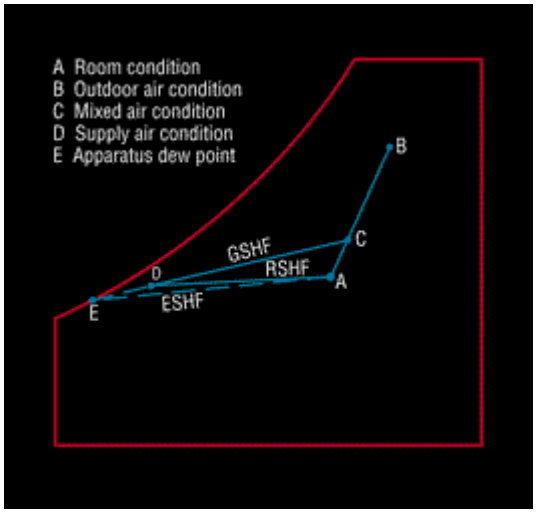


Fig - RSHF, ESHF, and GSHF relationship on a psychrometric chart

For a given cooling coil where bypass factor (BF), coil entering temperature condition at point C (T_C), and the required leaving dry bulb temperature condition at point D (T_D) are known, one can determine the coil leaving condition and the final condition of the room air (A) through an iterative process using the apparatus dewpoint temperature of the coil, the effective sensible heat factor (ESHF), and the grand sensible heat factor (GSHF). The GSHF is the ratio of the total sensible heat to the total sensible and latent heat.

By the definition of bypass factor, apparatus dew point temperature (T_E) can be calculated as follows:

$T_E = [T_D - (BF \times T_C)]/ (1 - BF)$ ---------- (Eq. 1)

The ESHF is the ratio of effective sensible heat to the effective sensible and latent heat:

ESHF = ESH/ (ESH + ELH) ---------- (Eq.2)

Effective sensible heat (ESH) is defined as the room sensible heat plus the appropriate fan heat and that portion of the outside air sensible heat and heat to the return air that is considered as being bypassed, unaltered, through the cooling coil. Effective latent heat (ELH) is defined as the room latent heat plus the portion of the outside air latent heat and latent heat to the return air, if any, that is considered as being bypassed, unaltered, through the cooling coil. A more detailed discussion of this calculation process is explained in the references.

The humidity ratio of T_E at saturation (W_E) can be calculated using psychrometric subroutines. Knowing T_E and W_E, one can calculate the humidity ratio of the room condition (W_A)--in lb of moisture per lb of dry air--as follows:

$W_A = W_E + (T_A - T_E) X [(1/ESHF) - 1)/4410]$ ---------- (Eq.3)

The calculated value of WA will differ from the originally assumed WA because of the imposed limitation of the cooling coil. An iterative process using the calculated WA can be set up to recalculate the latent loads, sensible heat factors, and a new value of WA, until the difference between the two successive values of WA is within a preset limit. The humidity ratio of the coil leaving condition can be calculated as follows:

WD = WC - (TC - TD) X [(1/GSHF) - 1)/4410] ---------- (Eq.4)

The constant 4410 in Equations 3 and 4 are used when W_A and W_D are in English units.

It is important to recognize that the psychrometric process loop shown in Fig. 1 must be closed and in equilibrium. In other words, the process line D--A, with the slope determined by the room sensible heat factor, must end at the same Condition A that was the room air condition at the beginning of the mixing process. When the Condition A at the end of Process D--A does not coincide with the targeted room air condition, an iterative process will have to take place. At the end of the process when the psychrometric polygon reaches its equilibrium, Condition A will be shifted away from the originally assumed condition.

Air system affects RH

The interrelating factors that can affect the RH (or WB) of the conditioned space in an air conditioning system include:

1) Latent load of the conditioned space
2) Outdoor air condition
3) Cooling coil leaving air condition
4) Type of air handling equipment selected for the air conditioning system

It is important to recognize the fact that unless the calculation involves a single air conditioned space, the RH conditions for various spaces served by the same air handling system will not be the same.
The following is a brief summary of how different air systems affect space RH:

1) For a reheat system serving multiple spaces, the RH of the space is a function of the amount of latent heat in each occupied space--*i.e.*, sensible heat factor.
2) For both double-duct and multi-zone systems, the RH of the conditioned space is a function of the room sensible heat factor and the condition of the air that bypasses the cooling coil.
3) With a variable air volume system, the RH of the conditioned space also is a function of the room sensible heat factor. The psychrometric process is different, however. The humidity ratio of the return air for any system serving multiple spaces is the weighted average of the humidity ratio of each space served by the system.

PART 9 WINTER HEATING LOAD

The heat loss is divided into two groups:

1) The heat transmission losses through the confining walls, floor, ceiling, glass, or other surfaces, and
2) The infiltration losses through cracks and openings, or heat required to warm outdoor air used for ventilation.

Normally, the heating load is estimated for winter design temperature usually occurring at night; therefore, internal heat gain is neglected except for theaters, assembly halls, industrial plant and commercial buildings. Internal heat gain is the sensible and latent heat emitted within an internal space by the occupants, lighting, electric motors, electronic equipment, etc.

As a basis for design, the most unfavorable but economical combination of temperature and wind speed is chosen. The winter month heating load conditions are based on annual percentiles of 99.6 and 99%.

The 99% and 99.6% cold values 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. Use of 99% values is recommended.

Heat Transmission Loss

Heat loss by conduction and convection heat transfer through any surface is given by:

$Q_{sensible} = A * U * (Ti – To)$

Where

1) Q = heat transfer through walls, roof, glass, etc.
2) A = surface areas
3) U = air-to-air heat transfer coefficient
4) Ti = indoor air temperature
5) To = outdoor air temperature

Floors on Slab

The slab heat loss is calculated by using the following equation:

$Q = F^* P * (T_i - T_o)$

Where:

1) F is the Heat Loss Coefficient for the particular construction and is a function of the degree days of heating.
2) P is the perimeter of slab
3) T_i is the inside temperature
4) T_o is the outside temperature

Heat loss from slab-on- grade foundations is a function of the slab perimeter rather than the floor area. The losses are from the edges of the slab and insulation on these edges will significantly reduce the heat losses.

The 2001 Fundamentals refers us to the 1981 Fundamentals for this data. Fortunately, it is also provided in Principles of HVAC, pages 4.16-4.17.The portion of heat transmission from basement is usually neglected unless the weather in winter is severe and the values are significant in comparison with other forms of heat transmission.

Infiltration and Ventilation Loss

The heat loss due to infiltration and controlled natural ventilation is divided into sensible and latent losses. The energy associated with having to raise the temperature of infiltrating or ventilating air up to indoor air temperature is the sensible heat loss, which is estimated by:

$Q_{sensible} = V * \rho_{air} * C_p * (Ti - To)$

Where:

1) V = volumetric air flow rate
2) ρ_{air} is the density of the air
3) C_p = specific heat capacity of air at constant pressure
4) Ti = indoor air temperature
5) To = outdoor air temperature

The energy quantity associated with net loss of moisture from the space is latent heat loss which is given by:

$Q_{latent} = V * \rho_{air} * h_{fg} * (Wi - Wo)$

Where

1) V = volumetric air flow rate
2) ρ_{air} is the density of the air
3) Wi = humidity ratio of indoor air
4) Wo = humidity ratio of outdoor air
5) hfg = latent heat of evaporation at indoor air temperature

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Cooling Load Vs Heating Load Calculations

Summer cooling-load calculations are similar in many ways to winter heating load calculations. However, there are some very important differences and cooling load calculation is relatively more complex. This is because in determining the heating load, credit for solar heat gain or internal heat gains (which are more complex to determine) are not usually included and the thermal storage effects of building structure or content are generally ignored.

The variables affecting cooling load calculations are numerous and often difficult to define precisely. The heat storage and time lag aspects of the load make the cooling load calculations more complex. Briefly the factors are:

1) First, there is usually a much greater outdoor temperature variation over a 24-hour period in summer than there is in winter.
2) Second, solar heat gain is a plus factor in winter heating, but it may be a major part of the load for summer cooling.
3) Third, there is the matter of the moisture content of summer air-latent heat-which has a great deal to do with human comfort. Much of this moisture must be removed from the indoor air in order to attain a comfortable condition, and this moisture load is a load on the cooling equipment.
4) Fourth, internal heat sources such as lights, machinery, appliances, and people constitute cooling loads in summer, whereas in winter the heat from these sources is a plus factor.

PART 10 CONSERVATION STRATEGIES – ARCHITECTURAL & MECHANICAL

The following load reduction strategies & checkpoints are listed from HVAC engineer's perspective.

1) Shape

The shape of the building has influence on the cooling and heating load. Ideally the building has to have the least aspect ratio (length/width ratio). The lower aspect ratio means the building has the least surface area of the building envelope (least wall area, glazing area and the roof area). Consider for example a 900 square feet area can be built as ~ 34 ft diameter room or can be made as 30 ft x 30 ft square or 60 ft x 15 ft rectangular. For a 10 ft height, the circular room shall have the surface area of 1067 sq-ft; for square the surface area is 1200 sq-ft and for rectangular the surface area is 1500 sq-ft. The lower surface area shows that not only the building will use less concrete, brick, wood, glazing or insulation but shall also have lower cooling and heating loss from the building envelope. Therefore the building shall be designed for least aspect ratio where possible.

2) Orientation

The orientation of a building often is determined by siting considerations. However, for those sites where there is a choice, analyzing the effect of orientation on energy and equipment costs can lead to a more energy-efficient building. While it is important to look at each project on an individual basis, as a general guide, long, narrow buildings facing south with their long axis running east/west will have lower peak cooling loads and may be able to utilize smaller cooling equipment. Conversely, buildings facing east or west with their long axis running north/south will have higher peak cooling loads and electricity demand costs, and may require larger cooling equipment. Orientation as well as directional emphasis changes with latitude in response to solar angles.

Zone	Building's main orientations	Directional emphasis
Tropical	On an axis 5° north of east	North-south
Arid	On an axis 25° north of east	South-east
Temperate	On an axis 18° north of east	South-south-east
Cool	On an axis facing south	Facing south

Research has shown that the preferred length of the sides of the building, where the sides are of length x: y is:

1) Tropical zone - 1:3
2) Arid zone - 1:2
3) Temperate zone - 1: 1.6
4) Cool zone - 1:1

Analysis of these ratios indicates that an elongated form to minimize east and west exposure is needed at the lower latitudes. This form slowly transforms to a ratio of 1:1 (cylindrical) at the higher latitudes. This is a direct response to the varying solar angles in the various latitudes.

3) Landscaping

Well designed landscaping can reduce cooling costs from summer heat gains in building. Trees planted on the east, west and south sides of a one-or two-story building can effectively reduce summer solar heat gains through windows, which is one of the major contributors to the cooling load on an air conditioning system. External shading with vegetation with natural deciduous trees is very effective at providing shade and cooling by evaporating water through their leaves: during winter they are bare, allowing sunlight to pass through, but during summer they shade the building.

4) Day lighting

Day lighting with skylights and other types of architectural glazing features can provide natural lighting creating a pleasant working atmosphere. Day lighting strategies may by particularly effective using skylights in large open areas such as warehouses and manufacturing plants, and in office spaces where the electrical lighting system output can be efficiently varied over a wide range of light levels.

In architectural design, climatic graphs and charts are useful to determine the position of the sun and optimize the built form, orientation and exposure of elements (windows, roof and walls) for maximum or minimum solar gain.

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5) Shading

a) Shading devices are designed from knowledge of the suns azimuth and altitude along with the wall-solar azimuth. In the lower latitudes there is total overheating, whereas in the higher latitudes overheating only occurs during the summer months.

b) Tropical regions need both vertical and horizontal shading throughout the year. In higher latitudes, horizontal and vertical shading is only needed during the summer on the south-facing sides of buildings.

c) There are obviously seasonal variations near the equator. Solar heating becomes more important than in the upper latitudes. Beginning at the equator and moving north, the need for solar heating increases whiles the need for solar shading diminishes.

d) Any breeze in the lower latitude (tropical and arid climates) is beneficial for most of the year whereas in higher latitudes most wind is detrimental and has to be screened.

e) Generally, for the tropical zones as much ventilation as possible is desired. For the arid zone cross ventilation is required, but care has to be taken to filter out high-velocity winds. In the temperate zone, cross ventilation and shielding are both necessary (for summer and winter, respectively). In the cool region, the building should be protected from cold, high-velocity winds, although cross ventilation is still required.

f) In the arid zone, the low level of humidity can be beneficial for evaporative cooling. In the tropical zone the high level of humidity can be very uncomfortable.

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6) Zoning for transitional spaces

Transitional areas are one that does not require total climate control and natural ventilation may be sufficient. These include lobbies, stairs, utility spaces, circulation, balconies and any other areas where movement take place. For the tropical and arid zones, the transitional spaces are located on the north and south sides of the building where the sun's penetration is not as great. An atrium can also be used a transitional space. In temperate and cool zones the transitional spaces should be located on the south side of the building to maximize solar gain.

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7) Use of atrium

In the tropical zone the atrium should be located in a way to provide ventilation within the built form. In the arid zone the atrium should be located at the center of the building for cooling and shading purposes. For the cool and temperate zones the atrium should be at the center of the building for heat and light.

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8) Potential of roof/ground floor as useable exterior space

In tropical and arid climates there is a high potential to make use of all external spaces, whereas moving towards the northern latitudes the external spaces have to be covered to be used.

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9) Vertical cores and structure

The arrangement of primary mass can be used as a factor in climatic design as its position can help to shade or retain heat within the building form. For the tropical zone, the cores are located on the east and west sides of the building form, so as to help shade the building from the low angles of the sun during the major part of the day. In arid zone, the cores should also be located on the east and west sides, but with major shading only needed during the summer. Therefore, the cores are located on the east and west sides, but primarily on the south side. The arrangement of the primary mass in the temperate zone is on the north face, so as to leave the south face available for solar heat gain during the winter. The cool zone requires the maximum perimeter of the building to be open to

the sun for heat penetration. Therefore the primary mass is placed in the center of the building so as not to block out the sun's rays and to retain heat within the building.

10) Mechanical Design Considerations

Thermal Zoning: 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 is known as thermal zoning. 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. In practice the corner rooms and the perimetric spaces of the building have variations in load as compared to the interior core areas. East facing zone will normally peak at 10 to 12 AM whole most building loads will peak at 3 to 4 PM. South facing zones are similar but will peak usually at noon to 2 PM and may peak in winter. Therefore the building shall be divided into smaller zones to control comfort levels in each zone. The buildings may be zoned into individual floors, rooms, or spaces with distinct loads, such as perimeter and interior zones. All zones should be calculated at both zone peak (for sizing air handling equipment) and building peak (for sizing central equipment).

Smaller buildings are usually divided into two major zones. These two zones may contain multiple sub-zones. Large projects should consider exposure zoning and velocity of prevailing winds as well as the requirement of interior zones.

a) 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.

b) Interior Zone: The area contained by the external zone. The interior zone is only slightly affected by outdoor conditions. Thus, the interior zone usually has uniform cooling. Heating is generally provided from the exterior zone.

Identifying the thermal zones is the first step in the design of any HVAC system. For large building footprints, assume a minimum of five zones per floor: one zone for each exposure and an interior zone. Single-zone models should be limited to open floor plans with perimeter walls not exceeding 40 feet in length.

If specific requirements are met, zonal control may earn a credit towards compliance with whatever building energy efficiency standards are applicable.

11) Window Solar Control Tips

The key recommendations include:

a) In general cases, specify low U-factors (< 0.40) for residential applications. Even lower values may be desired in extreme heating climates.

b) In climates with significant air conditioning loads, specify windows with low SHGC values (< 0.40).

c) In general, high (> 70%) Glass Visible Transmittance is desired, especially for day lighting applications.

d) For commercial buildings in conjunction with day lighting strategies, analyze the trade-offs between standard glazing and high coolness index (also called spectrally selective) glass. Spectrally selective glass has a relatively high visible transmittance and a relatively low SHGC.

e) Low SHGC windows should be considered for east- and west-facing glazing as a means of controlling solar heat gain and increasing occupant comfort. For large commercial and industrial structures, specify low SHGC windows on the east, south, and west facades. SHGC for north-facing windows is not critical for most latitudes in the continental United States.

f) For buildings where passive solar heating energy is desired, south-facing windows with high SHGC values coupled with low U-factors should be specified.

g) Select windows with comfort in mind. The proper specification of windows can result in higher Mean Radiant Temperature (MRT) in winter and lower MRT in summer, improving occupant comfort and productivity. MRT represents the average temperature an occupant feels from radiant heat exchange with their surroundings.

h) Single-pane windows are impractical in heating-dominated climates. In these regions, multiple-pane, low-e, and gas-filled window configurations are advisable.

i) Specify aluminum-frame windows with thermal breaks or should be avoided at all. Even in milder climates, these windows tend to have low inside surface temperatures during the heating season, giving rise to condensation problems. Wood, vinyl, and fiberglass are the best frame materials for insulating value.

j) Window solar heat gain coefficients should be selected according to orientation. If south exposures are to admit beneficial solar heat during the heating season, their solar heat gain coefficients should be high. These high solar heat gain coefficients will not usually result in overheating problems during the cooling season because of the lower solar radiation levels on south-facing windows, especially those with overhang, at that time.

k) Skylights and east- and west-oriented windows may warrant lower solar heat gain coefficients since they transmit the most solar heat during cooling periods. There isn't much point in spending more money to obtain lower solar heat gain coefficients for north-facing windows.

l) Windows with spectrally selective or low-e coated glazing with low solar heat gain coefficients are often effective in hot, sunny climates. Darker glazing tints also provide lower solar heat gain coefficients, but they may yield somewhat decreased visibility.

m) If exterior or interior shading devices, such as awnings, louvered screens, sunscreens, Venetian blinds, roller shades, or drapes, will be used on windows, lower window solar heat gain coefficients may not be necessary, depending on individual circumstances. Many shading devices can be adjusted to admit more or less solar heat according to the time of day and the season, but windows with lower solar heat gain coefficients require less maintenance.

n) Exterior shading devices are more effective than interior devices in reducing solar heat gain because they block radiation before it passes through a window. Light-colored shades are preferable to dark ones because they reflect more, and absorb less, radiation. Horizontally oriented adjustable shading devices are appropriate for south-facing windows, while vertically oriented adjustable devices are more effective for shading windows on east and west orientations.

o) Low-e windows and skylights are the best options for decreasing the transmission of ultraviolet radiation.

p) Buy windows with energy efficient label. The window energy label lists the U-factor, solar heat gain coefficient, visible light transmittance, and air leakage rating.

q) Operable, rather than fixed, windows should be installed in household areas with high moisture production, such as bathrooms, kitchens, and laundry rooms, and in other areas where natural ventilation is desired.

r) Select windows with air leakage ratings of 0.2 cubic feet per minute per square foot of window area (cfm/ft^2) or less. Check the seals between window components for air tightness. To minimize infiltration around installed windows, caulk and weather-strip cracks and joints

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12) Other Miscellaneous Tips

a) The heating & cooling load for exterior and interior zones should be calculated in different zones and should have separate HVAC systems

b) Design multi-story buildings with typical floor HVAC design and configuration whenever applicable

c) Stores, kitchens, cafeterias, and entertainment areas may have their own HVAC systems due to differing design criteria

d) Consider separate HVAC systems for areas which directly separate the interior from the exterior (i.e., main entrances and lobbies). These areas may be designed 4 to 6°F above interior temperature during summer to reduce the temperature differential shock when entering or leaving the building.

e) Explore passive solar strategies and non-energy intensive HVAC and lighting opportunities. Use the following approach in performing the analysis of different systems.

f) Consider the building envelope when examining HVAC strategies.

g) Consider the building orientation and footprint.

h) Landscaping is a natural and beautiful way to shade and block the sun. A well-placed tree, bush, or vine can deliver effective shade and add to the aesthetic value to the property.

i) Consider thermal mass appropriately placed.

j) Light-colored roofs and walls reflect heat away from your home.

k) Carefully analyze the building's application (occupancy hours, intended use, etc.) and maximum occupancy

l) Evaluate lighting and equipment loads including special allowance factors, average from 2 to 5 watts/sq ft. lighting and heavy equipment loads (i.e., computers) may have higher loads from 5 to 10 watts/sq ft. Fluorescent light wattage is multiplied by 1.25 to include heat gain due to ballast.

m) Consider light troffers for suspended ceilings with ducted supply and plenum return.

n) Reflective window coatings reflect heat away from windows, as well as cutting glare and reducing fading of furniture, drapes, and carpeting inside the house. Note: Do not place reflective coatings on south-facing windows if you want to take advantage of heat gain during the winter.

o) Weatherization measures--such as insulating, weather-stripping, and caulking--help seal and protect the building against the summer heat in addition to keeping out the winter cold. The attic is a good place to start insulating because it is a major source of heat gain.

p) A properly installed awning can reduce heat gain up to 65% on southern windows and 77% on eastern windows. Also effective are louvers and shutters.

q) Double glass is most effective in areas where the conduction component is quite large.

r) Solar films are more effective in areas of moderate, sunny climates, i.e. Florida.

s) Draperies and curtains made of tightly woven, light-colored, opaque fabrics reflect more of the sun's rays than they let through.

t) Ventilate the building during the coolest parts of the day or night, and seal it up during the hottest part of the day.

u) Ventilated attics are about 30°F cooler than unventilated attics. Properly sized and placed louvers and roof vents help prevent moisture buildup and over heating the attic.

v) Optimize energy benefits of glazing through appropriate selection, placing, and design of the building façade.

w) Consider day lighting strategies to reduce HVAC requirements.

x) Design the HVAC system with the outdoor air rates required by ASHRAE Standard-62 to maintain indoor air quality. "Build Tight & Ventilate Right".

Appendix - A

DESIGN FACTORS & INPUTS

Summarizing, a building experiences a range of cooling & heating loads at any point of time in any given year, ranging in magnitude from zero (no cooling required) to whatever the maximum load happens to be that year. Design cooling load is a load near the maximum magnitude, but is not normally the maximum. This should become clear when the design factors and assumptions behind the calculations are understood.

DESIGN FACTORS

1) Conduction/convection of heat through walls, roofs, floors, doors and windows
2) Radiation through windows and heating effects on wall and roof surface temperatures
3) Thermal properties of buildings (Insulation, glass transmittance, surface absorbtivity)
4) Building thermal mass and corresponding delay of indoor temperature change
5) Construction quality in preventing air, heat, and moisture leakage
6) Heat added/lost with ventilation air needed to maintain air quality (code compliance)
7) Heat generated by lights, people, appliances, and equipment
8) Heat added/lost by air, water, and refrigeration distribution systems
9) Heat generated by air and water distribution equipment
10) Moisture added/lost with ventilation air to maintain air quality and code compliance
11) Moisture movement through building envelope
12) Moisture generated by occupants and equipment
13) Activity level, occupancy patterns, and make- up (male, female, child) of people
14) Acceptable comfort and air quality levels of occupants
15) Weather conditions (temperature, moisture, wind speed, latitude, elevation, solar radiation, etc.)

DESIGN INPUTS

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

A. Obtain building characteristics

Materials of construction for external walls, roofs, windows, doors, internal walls, partitions, ceiling, insulating materials and thick nesses, external wall and roof colors

1) Architectural plans, sections and elevations
2) Building size, orientation (N, S, E, W, NE, SE, SW, NW, etc), dimensions, location, fenestrations, ground reflectance etc.
3) External/Internal shading
4) Occupancy type and time of day

Check the following….

a. Type of structure, heavy, medium or light
b. Is structure insulated?
c. Is structure exposed to high wind?

5) Amount of glass
6) Length of reduced indoor temperature
7) What type of cooling or heating devices will be used?
8) Select and/or compute U-values for walls, roof, windows, doors, partitions, etc.

B. Select Outdoor Design Weather Conditions

The building information noted below is required. In most projects, this information is normally provided in the project design criteria. However, in the absence of such criteria, consult ASHRAE Handbook of Fundamentals to ascertain summer and winter design conditions for most locations. For computer programs, weather data is usually included with the program.

1) Design outdoor temperature
2) Location and Latitude

3) Altitude
4) Weather data (coincident dry bulb and wet bulb temperatures, daily range)
5) Wind direction and speed
6) Precipitation

Check the following….
a) What is daily temperature range, minimum/maximum?
b) Are there significant variations from ASHRAE weather data?

C. Select Indoor Design Conditions

The information noted below is to be determined.
1) Temperature and relative humidity for each space/room
2) Permissible variation or control limits of the temperature and relative humidity
3) Room Pressurization requirements
4) Ventilation rate: Determine if there is special equipment such as kitchen or lab hoods that require a minimum exhaust rate and eventually shall affect the ventilation rate.
5) Room function, number of occupants and the period of occupancy in each room

Check the following….
a) Estimate temperatures in un-conditioned spaces
b) Infiltration or ventilation load in accordance with ASHRAE Standard 62

D. Operating Schedule

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:
1) Lighting requirements, types of lighting fixtures
2) Appliances requirements such as computers, printers, fax machines, water coolers, refrigerators, microwave, miscellaneous electrical panels, cables etc
3) Heat released by the HVAC equipment.
4) Number of occupants, time of building occupancy and type of building occupancy
5) Determine area of walls, windows, floors, doors, partitions, etc.
6) Compute conduction heat gains for all walls, windows, floors, doors, partitions, skylights, etc.
7) Compute solar heat gains for all walls, windows, floors, doors, partitions, skylights, etc.
8) Infiltration heat gains are generally ignored unless space temperature and humidity tolerance are critical.
9) Compute ventilation heat gain required.
10) Compute internal heat gains from lights, people, and equipment.
11) Compute sum of all heat gains indicated in items above
12) Consider equipment and materials, which will be brought into building above inside design temperature.
13) Cooling load calculations should be conducted using industry accepted methods to determine actual cooling load requirements.

Typical Assumptions

Design cooling load is intended to summarize all the cooling loads experienced by a building under a specific set of assumed conditions. The typical assumptions behind design cooling load are as follows:
1) Weather conditions are selected from a long-term statistical database. The conditions will not necessary represent any actual year, but are representative of the location of the building. ASHRAE has tabulated such data. The designer may select a severity of weather that seems appropriate for the building type in question--although energy codes often specify what data shall be used (to minimize over-sized systems).
2) 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.
3) The building occupancy is assumed to be at full design capacity.
4) The ventilation rates are either assumed on air changes or based on maximum occupancy expected.
5) All building equipment and appliances are considered to be operating at a reasonably representative capacity.
6) Lights and appliances are assumed to be operating as expected for a typical day of design occupancy.
7) Latent as well as sensible loads are considered.
8) Heat flow is analyzed assuming dynamic conditions, which means that heat storage in building envelope and interior materials is considered.

Appendix - B

THERMAL TRANSMISSION THROUGH BUILDINGS

All the materials that are used in the construction absorb and transfer heat. By knowing the resistance of heat flow through a building component (R-value), you can calculate the total amount of heat entering the building. The basic equation to determine the heat loss or gain through an opaque surface such as walls, roof, etc. is given by relationship:

$Q = U \times A \times \Delta T$

Where:

Q = the heat flow through the walls, etc., in BTU per hour

U = the U-value in BTU per (hour) (square feet) (°F)

A = the area in square feet

ΔT = Difference in outside and inside temperatures in °F.

Q is the rate of heat flows through a medium. For example, the heat transfer in 24 hours through 2 sq-ft. of material, 3" thick, having a thermal conductivity factor of 0.25, with an average temperature difference across the material of 70°F would be calculated as follows:

Q = 0.25(k) x 2 sq. ft x 24 hours x 70° ΔT = 280 BTU

Before we go further, let's refresh few basic fundamentals and definitions.

Heat is transferred from a high temperature zone towards a low temperature zone by three mechanisms;

1. Conduction
2. Convection
3. Radiation

For HVAC load calculation purposes, conduction and radiation are primarily considered. Conduction is the transfer of heat through an object and radiation is the transfer of heat through electromagnetic waves, in this case sunlight. Heat travels from hot to cold and construction materials resist the flow of heat through them differently. For example, heat passes through glass much easier than wood siding.

With buildings, we refer to heat flow in a number of different ways. The most common reference is "R-value," *resistance* to heat flow or "U-value", which is a measure of flow of heat through a material. The higher the R-value of a material, the better it is at resisting heat loss (or heat gain).

R-value and U-factor are the inverse of one another: U = 1/R. Materials that are very good at resisting the flow of heat (high R-value, low U-factor) can serve as insulation materials.

"R" values, "k" values, "C" values, "U" values, what it all means?

Basically all these letter symbols are used to denote heat transfer factors. All of these terms describe the same phenomenon; however, some are described as determined by material dimensions and boundaries.

Building envelope is typically composed of various elements. A wall may be constructed of hardboard (facing outdoors), plywood (facing indoors) and sandwich insulation in between.

When a building structure is composed of various layers of construction elements having resistances R1, R2, R3…. Rn, the overall resistance value is sum of all individual resistances for whole wall, internal air spaces, insulation materials and air films adjacent to solid materials. Individual R-values are used in calculating overall heat transfer coefficients.

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k = Thermal Conductivity

"k Value" is the material property, which measures conductivity and is the quantity of heat (BTU's/hr) that passes through one inch of a homogeneous material. A material is considered homogeneous when the value of its thermal conductivity does not depend on its dimension. It is the same number regardless of the thickness of insulation. Thermal Resistivity, or "R" is the reciprocal of thermal conductivity i.e. R = 1/k. Thermal conductivity is expressed in (Btu-in/hr ft^2 °F). Materials with lower k-values are better insulators.

Insulation materials usually have K-factors less than one and are reported at what is called mean temperature. To determine the mean temperature, measure the surface temperatures on both sides of the insulation, add them together and divide by two. “As mean temperatures rises, so does the K-factor”

R = Thermal Resistance

The R-factor is the thermal resistance factor and is a measure of the ability to retard heat flow in a given thickness of material. R is the numerical reciprocal of C (R = 1/C). The higher the R-value, the higher (better) the insulating value. R Values change as the thickness of the insulating material changes. Primarily building insulation products and plans are measured and specified by the material's R factor. R-value for material only deals with conductive heat transfer. Since the total heat transferred by conduction varies directly with time, area, and temperature difference, and varies inversely with the thickness of the material, it is readily apparent that in order to reduce heat transfer, the ‘k’ factor should be as small as possible, and the material as thick as possible. For example, a wall with a U-value of 0.25 would have a resistance value of R = I/U = 1/0.25=4.0. The value of R is also used to represent Thermal Resistivity, the reciprocal of the thermal conductivity. Thermal Resistivity is expressed in (hr ^{0}F ft^2)/(Btu in)

C = Thermal Conductance.

The C-factor (thermal conductance factor) is the number of B t u’s that will pass through a square foot of material with a one-degree Fahrenheit temperature difference for a specified thickness. C factor is similar to ‘k’, except it is the rate of heat flow through an actual thickness of material, where ‘k’ is a factor per inch. The C-factor is the K-factor divided by the thickness of the insulation. The formula is the reciprocal of the R-factor formula. The lower the C value, the better the insulator.

Note that the conductance of an air space is dependent on height, depth, position, character and temperature of the boundary surfaces. Therefore, the air space must be fully described if the values are to be meaningful. For a description of other than vertical air spaces, see the 1981 ASHRAE *Handbook of Fundamentals,* Chapter 23. Thermal Conductance is expressed in Btu/ (hr ^{0}F ft^2)

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h = Film or Surface Conductance.

The rate of heat exchange between a unit or surface area and the air it is in contact with. Subscripts i and o are used to denote inside and outside conductances, respectively. Film or surface conductance is expressed in Btu/(hr ^{0}F ft^2).

U = Overall Coefficient of Heat Transmission.

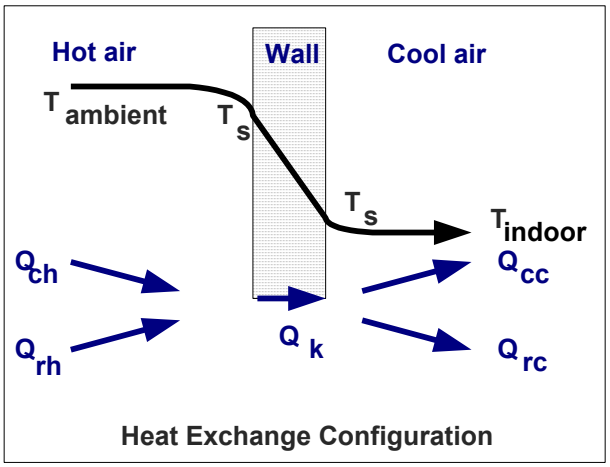
The U-value is the rate of heat flow passing through a square foot of the material in an hour for every degree Fahrenheit difference in temperature across the material (Btu/ft^{2}hr°F). This is the property that should be determined when figuring the heat loss or gain through walls, floors, ceilings, etc. The U-value or conductance flows through a material and the R-value denotes the resistance, or how slowly heat flows. The two terms are reciprocal. *(R=1/U, U=1/R).* The Overall Coefficient of Heat Transmission is expressed in Btu/(hr ^{0}F ft^2).

Windows are commonly described by their U-values. Descriptions of building walls, floors, or ceilings, often use R-values instead of U-values.

U factor has to be computed for each part of the structure. Most good insulating materials have a thermal conductivity (k) factor of approximately 0.25 or less, and rigid foam insulations have been developed with thermal conductivity (k) factors as low as 0.12 to 0.15.

Combined Modes of Heat Transfer

1) Heat transfer by convection Q_{ch} and radiation Q_{RH} from the hot air and surrounding surfaces to the wall surface,
2) Heat transfer by conduction through the wall Q_k
3) Heat transfer by convection Q_{cc} and radiation Q_{rc} from the wall surface to the cold air and surrounding surfaces.



Heat Exchange Configuration

When one side of the wall is warmer than the other side, heat will conduct from the warm side into the material and gradually move through it to the colder side. A temperature gradient is established across the thickness of the wall. The temperature gradient is linear between the two surfaces for a homogenous wall and the slope of temperature gradient is proportional to the resistances of individual layers for a composite structure.

If both sides are at constant temperatures--say the inside surface at 77°F (25°C) and the outside surface at 95°F (35°C)--conductivity will carry heat inside the building at an easily predicted rate.

Under steady state conditions, the total rate of heat transfer (Q) between the two fluids is:

$$Q = Q_{ch} + Q_{rh} = Q_k = Q_{cc} + Q_{rc}$$

In real-life situations, however, the inside and outside temperatures are not constant. In fact the driving force for conductive heat flow can reverse during the course of a day. As night falls, the outside air temperature may drop to 50°F (10°C). As the temperature difference across the wall is reversed, the heat flow is also reversed--drawing heat back towards the outside of the building.

Another scenario is when the outside temperature fluctuates but never crosses the indoor set point temperature. In this case, the direction of heat flow never changes, but the *thermal lag* or *time delay* in heat flow can still be beneficial by delaying the peak heating or cooling load.

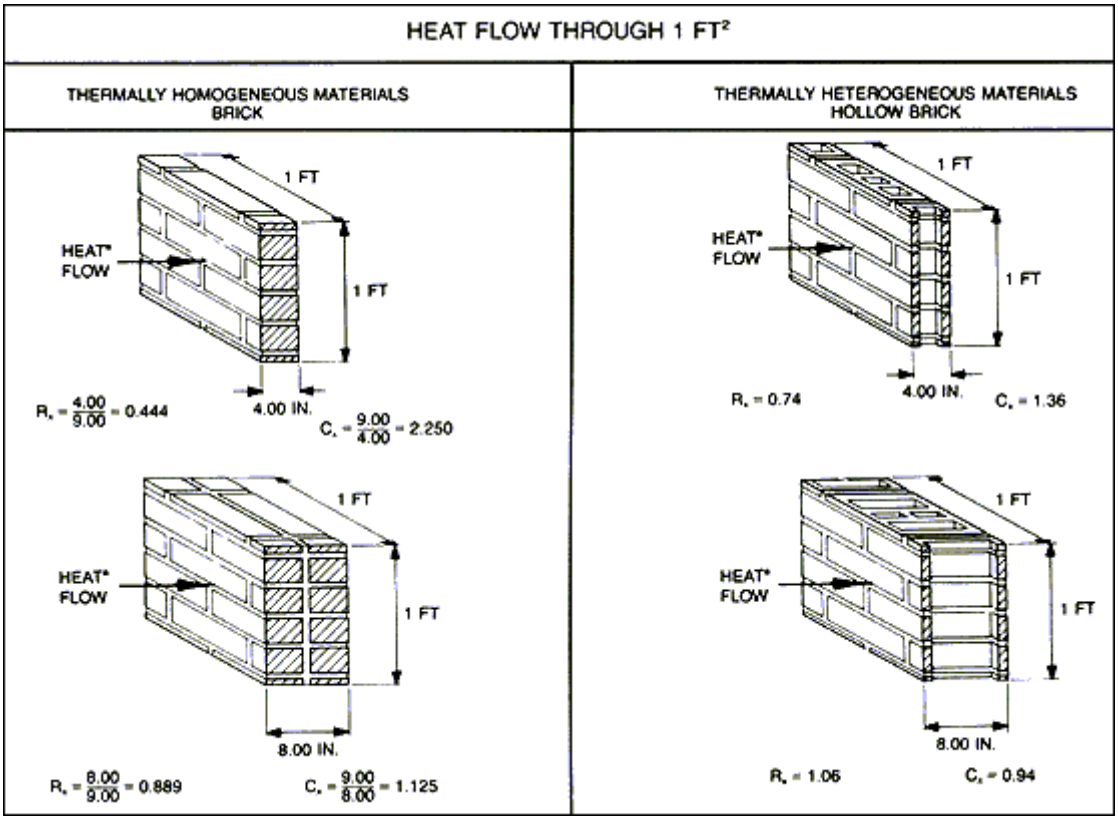
Calculation Methods

Conductance and resistances of homogeneous material of any thickness can be obtained from the following formula: $C_x=k/x$, and $R_x=x/k$

Where:
x=thickness of material in inches.

Materials in which heat flow is identical in all directions are considered thermally homogeneous.

This calculation for a homogeneous material is shown in Fig. below. The calculation only considers the brick component of the wall assembly. Whenever an opaque wall is to be analyzed, the wall assembly should include both the outside and inside air surfaces. The inclusion of these air surfaces makes all opaque wall assemblies layered construction.



Thermal Transmittance through Materials

In computing the heat transmission coefficients of layered construction, the paths of heat flow should first be determined. If these are in series, the resistances are additive, but if the paths of heat flow are in parallel, then the thermal transmittances are averaged. The word "series" implies that in cross-section, each layer of building material is one continuous material. However, that is not always the case. For instance, in a longitudinal wall section, one layer could be composed of more than one material, such as wood studs and insulation, hence having parallel paths of heat flow within that layer. In this case, a weighted average of the thermal transmittances should be taken.

Series heat flow

For layered construction, with paths of heat flow in series, the total thermal resistance of the wall is obtained by:

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

Or $R_{Total} = 1/C + x_1/k_1 + x_2/k_2 \ldots$

Where

C is the conductance

x_1 is the thickness of material one

x_2 is the thickness of material two

k_1 is the thermal conductivity of material one

k_2 is the thermal conductivity of material two

And the overall coefficient of heat transmission is:

$U = 1/R_{Total}$

or

$$U = \frac{1}{R_i + R_1 + R_2 + \ldots R_o}$$

Where:

R_i = the resistivity of a "boundary layer" of air on the inside surface.

R_1, R_2 …= the resistivity of each component of the walls for the actual thickness of the component used. If the resistance per inch thickness is used, the value should be multiplied by the thickness of that component.

R_o = the resistivity of the "air boundary layer" on the outside surface of the wall.

The formula for calculating the U factor is complicated by the fact that the total resistance to heat flow through a substance of several layers is the sum of the resistance of the various layers. The resistance to heat flow is the reciprocal of the conductivity. Therefore, in order to calculate the overall heat transfer factor, it is necessary to first find the overall resistance to heat flow, and then find the reciprocal of the overall resistance to calculate the U factor.

NOTE:

Note that in computing U-values, the component heat transmissions are not additive, but the overall U-value is actually less (i.e., better) than any of its component layers. The U-value is calculated by determining the resistance of each component and then taking the reciprocal of the total resistance. Thermal resistances (R-values) must first be added and the total resistance (R-Total) divided into 1 to yield the correct U-factor.

Correct:

$$U = \frac{1}{R_1 + R_2 + R_3 + \ldots + R_n} = \frac{1}{R_{Total}}$$

Incorrect:

$$U = \frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_3} + \ldots + \frac{1}{R_n} = U_1 + U_2 + U_3 + \ldots + U_n$$

The total R-value should be calculated to two decimal places, and the total U-factor to three decimal places.

Parallel Heat Flow

Average transmittances for parallel paths of heat flow may be obtained from the formula:

$U_{avg} [A_A (U_A) + A_B (U_B) + \ldots] / A_t$

Or

$U_{avg} = [1/ (R_A / A_A) + 1/(R_B/A_B) \ldots]/A_T$

Where:

A_A, A_B, etc. = area of heat flow path, in Ft^2,
U_A, U_B, etc.= transmission coefficients of the respective paths,
R_A, R_B, etc.=thermal resistance of the respective paths.
A_t= total area being considered (A_A+A_B+...), in Ft^2

A_t = total area being considered (A_A+A_B+...), in ft^2

Such an analysis is important for wall construction with parallel paths of heat flow when one path has a high heat transfer and the other a low heat transfer, or the paths involve large percentages of the total wall with small variations in the transfer coefficients for the paths.

Example #1

Determine the U-value for a layered wall construction assembly composed of three materials:

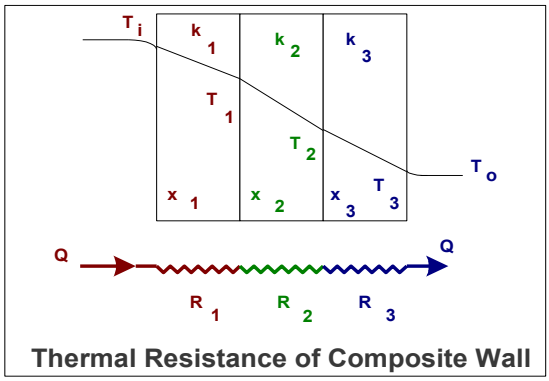
1) Plywood, 3/4-inch thick (R_1 = 3/4 X 1.25 = 0.94)

2) Expanded polystyrene, 2-inches thick (R_2 = 2" X 4.00 = 8.00)

3) Hardboard, 1/4-inch thick (R_3 = 0.18)

4) Ri = 0.68 ("still" air)

5) Ro = 0.17 (15 MPH wind, winter conditions)



Thermal Resistance of Composite Wall

The U-values is:

$$U = \frac{1}{R_i + R_1 + R_2 + R_3 + R_{o_o}}$$

$$= \frac{1}{0.68 + 0.18 + 8.00 + 0.94 + 0.17}$$

$$= \frac{1}{9.97} = 0.10 \frac{BTU}{hr. - sq.\ ft. - °F}$$

To calculate heat loss for say for 100 square feet of wall with a 70° F temperature difference would be:

$$Q = (.10)\ (100)\ (70) = 700\ \ BTU/\ HR$$

In the calculations above the TD is taken as 70°F, which is temperature difference between indoor and outside air. If the sun shines on a wall or roof of a building and heats the surface much hotter than the air (as typical in the summer), the heat flow through the wall or roof would be greatly influenced by the hot surface temperature; hence, use a surface temperature rather than air to obtain a more realistic heat flow rate. Similarly, when calculating the heat flow through a floor slab resting on the ground, there will not be an air boundary-layer resistance underneath (Ro = 0) and the temperature (t_o) will be the ground temperature (not the outside air temperature).

Example #2

Calculate the U factor of a wall composed of 2" of material having a 'k' factor of 0.80, and 2" of insulation having a conductance of 0.16.

U value is found as follows:

R total = 1/C + X1/k1 or

R total = 1/0.16 + 2/0.80

R total = 8.75

U = 1/R or 1/8.75 = 0.114 Btu/hr ft^2 °F

Once the U factor is known, the heat gain by transmission through a given wall can be calculated by the basic heat transfer equation. Assuming an area of 100 square feet wall with an inside temperature of 85°F and an outside temperature of 115°F, the heat transmission would be:

Q = U x A x TD

Q = 0.114 x 100 x 30

Q = 342 Btu/hr

HEAT TRANSMISSION COEFFICIENTS OF COMMON BUILDING MATERIALS

Material Description	*Density Lb/ft³*	*Conduction (k)* Btu-in/hr ft² °F	*(C)* Btu/hr ft² °F	*Resistance (R) Per inch thickness L/k*	*For thickness listed L/C*
Masonary Units					
Face Brick	130	9.00		0.11	
Common Brick	120	5.00		0.20	
Hollow Brick					
4” (62.9% solid)	81		1.36		0.74
6” (67.3% solid)	86		1.07		0.93
8” (61.2% solid)	78		0.94		1.06
10” 60.9% solid)	78		0.83		1.20
Hollow Brick vermiculite fill					
4” (62.9% solid)	83		0.91		1.10
6” (67.3% solid)	88		0.66		1.52
8” (61.2% solid)	80		0.52		1.92
10” 60.9% solid)	80		0.42		2.38
Lightweight concrete block-100 Lb density concrete					
4”	78		0.71		1.40
6”	66		0.65		1.53
8”	60		0.57		1.75
10”	58		0.51		1.97
12”	55		0.47		2.14
Lightweight concrete block vermiculite fill - 100 Lb density concrete					
4”	79		0.43		2.33
6”	68		0.27		3.72
8”	62		0.21		4.85
10”	61		0.17		5.92
12”	58		0.15		6.80
Building Board					
3/8” -Drywall Gypsum	50		3.10		0.32
1/2” -Drywall Gypsum	50		2.25		0.45
Plywood	34	0.80		1.25	
½” Fiberboard sheathing	18		0.76		1.32
Siding					
7/16” harDBoard	40			1.49	0.67
½” by 8” Wood bevel	32			1.23	0.81
Aluminum or steel over sheathing				1.61	0.61
Insulating Material					
Batt or Blanket					
• 2 to 2 ¾”					7.0
• 3 to 3 ½”	1.20				11.0
• 5 ½” to 6 ½”	1.20				19.0
Boards					
• Expanded Polystrene	1.80	0.25		4.00	
• Expanded Polyurethane	1.50	0.16		6.25	
• Poly isocyanurate	2.0	0.14		7.14	
Loose Fill					
• Vermiculite	4 - 6	0.44		2.27	

Material Description	*Density Lb/ft³*	*Conduction (k) Btu-in/hr ft² °F*	*(C) Btu/hr ft² °F*	*Resistance (R) Per inch thickness L/k*	*For thickness listed L/C*
• Perlite	5 - 8	0.37		2.70	
Woods					
Hard woods	45.0	1.1		0.91	
Soft woods	32.0	0.80		1.25	
Metals					
Steel	-	312		0.003	
Aluminum	-	1416		0.0007	
Copper	-	2640		0.0004	
Air Space					
¾" to 4"- winter			1.03		0.97
¾" to 4" - summer			1.16		0.86
Air Surfaces					
Inside – Still air			1.47		0.68
Outside – 15 mph wind-winter			6.00		0.17
Outside – 7.5 mph wind -summer			4.00		0.25

Appendix - D

RULE OF THUMB FIGURES

Only for rough estimating purposes
The following table can be referred for quick but rough estimate of cooling requirements for commercial applications.

Application	Average Load
Residence	400-600 sq. ft. floor area per ton
Apartment (1 or 2 room)	400 sq. ft. of floor area per ton
Church	20 people per ton
Office Building	
Large Interior	340 sq. ft. of floor area per ton
Large Exterior	250 sq. ft. of floor area per ton
Small Suite	280 sq. ft. of floor area per ton
Restaurant	200 sq. ft. of floor area per ton
Bar or Tavern	9 people per ton
Cocktail Lounge	175 sq. ft. of floor area per ton
Computer Room	50 – 150 sq. ft. of floor area per ton
Bank (main area)	225 sq. ft. of floor area per ton
Barber Shop	250 sq. ft. of floor area per ton
Beauty Shop	180 sq. ft. of floor area per ton
School Classroom	250 sq. ft. of floor area per ton
Bowling Alley	1.5 – 2.5 tons per alley
Department Store	
Basement	350 sq. ft. of floor area per ton
Main Floor	300 sq. ft. of floor area per ton
Upper Floor	400 sq. ft. of floor area per ton
Small Shop	225sq. ft. of floor area per ton
Dress Shop	280 sq. ft. of floor area per ton
Drug Store	150 sq. ft. of floor area per ton
Factory (precision manufacturing)	275 sq. ft. of floor area per ton
Groceries – Supermarket	350 sq. ft. of floor area per ton
Hospital Room	280 sq. ft. of floor area per ton
Hotel Public Spaces	220sq. ft. of floor area per ton
Motel	400 sq. ft. of floor area per ton
Auditorium or Theater	20 people per ton
Shoe Store	220 sq. ft. of floor area per ton
Specialty & Variety Store	200 sq. ft. of floor area per ton

Caution… these figures are for estimating purposes only! Many conditions such as orientation, sun loads, number of occupants and light wattage can greatly affect the total tonnage requirements. A detailed load calculation must be made to ensure accuracy. These figures are only for sanity check.

Appendix - E

DESIGN Software's

Computerized simulations are now commonly used to estimate design-cooling load in practice. The good software's available today contain all database information on the weather data, CLTD, CLF, SC and SCL tables, descriptions of walls and roof types, heat gains of lighting, appliances, and people, duct losses, ventilation air requirements, and building materials thermal properties with calculators to determine overall R and U values. Learning load calculation software is not difficult, but taking a class can help. Many software manufacturers offer technical support, as well.

It is hard to make a case for manual calculations in current IT environment but still basic understanding of design principles won't hurt before software can be profitably used. Basic knowledge about the subject is required to judiciously evaluate the inputs and perform the sanity check on the output results of computer analysis. The most common manual load computation method is cooling load temperature difference CLTD method and with advent of computer programs, CLTD manual method is suppressed, but not invalidated. Manual J method is used as a baseline because it is the most widely accepted load calculation methodology and is generally recognized as providing a safe estimate of cooling load.

HVAC Load Calculations and Psychrometric Analysis

1) Trace 700 by Trane
2) E-20II by Carrier
3) Hevacomp by Hevacomp Ltd.
4) Htools & RHVAC by Elite software
5) Loadsoft by Carmel software
6) HVAC-calc by HVAC computer systems Ltd

HVAC Load/Energy/Economic analysis

1) DOE EnergyPlus
2) Trace 700 by Trane

Appendix - F

DEFINITIONS OF USEFUL TERMS

1) Ambient Air - The air surrounding a building; outside air
2) Air Change - The term air change is a rate at which outside air replaces indoor air in a space. It can be expressed in one of two ways: the number of changes of outside air per unit of time air changes per hour (ACH); or the rate at which a volume of outside air enters per unit of time - cubic feet per minute (CFM).
3) Building Envelope - The term building envelope indicates the surfaces that separate the inside from the outdoors. This includes the parts of the building: all external building materials, windows, walls, floor and the roof. Essentially the building envelope is a barrier between the conditioned indoor environment and the outdoors.
4) Building Location Data- Building location data refers to specific outdoor design conditions used in calculating heating and cooling loads.
5) British thermal unit (BTU): Theoretically, it is approximate heat required to raise 1 lb. of water 1 deg Fahrenheit, from 59^0F to 60^0F. Its unit of heat and all cooling and heating load calculations are performed in Btu per hour in US.
6) Cooling load: The rate at which heat is removed from a space to maintain the constant temperature and humidity at the design values
7) Cooling Load Temperature Difference (CLTD) – A value used in cooling load calculations for the effective temperature difference (delta T) across a wall or ceiling, which accounts for the effect of radiant heat as well as the temperature difference. CLTD value calculates the instantaneous external cooling load across a wall or roof. CLTD value is used to convert the space sensible heat gain to space sensible cooling load.
8) Cooling Coil Load – The rate at which heat is removed at the cooling coil that serves one or more conditioned spaces and is equal to the sum of all the instantaneous space cooling loads.
9) Cubic feet per minute (CFM) - The amount of air, in cubic feet, that flows through a given space in one minute. 1 CFM equals approximately 2 liters per second (l/s). A typical system produces 400 CFM per ton of air conditioning.
10) Comfort Zone- The range of temperatures, humidity's and air velocities at which the greatest percentages of people feel comfortable.
11) Design Conditions- Cooling loads vary with inside and outside conditions. A set of conditions specific to the local climate is necessary to calculate the expected cooling load for a building. Inside conditions of 75°F and 50% relative humidity are usually recommended as a guideline. Outside conditions are selected for the 2.5% climate occurrence.
12) Exfiltration- Uncontrolled air leakage out of a building through window and door openings
13) Exhaust - The airflow leaving the treated space from toilets, kitchens, laboratories or any hazardous area where negative pressure is desired.
14) Enthalpy - Heat content or total heat, including both sensible and latent heat.
15) Fenestration – is an architectural term that refers to the arrangement, proportion and design of window, skylight and door systems within a building. Fenestration consists of glazing, framing and in some cases shading devices and screens.
16) Heating load: The heating load is a rate at which heat is added to the space to maintain the indoor conditions.
17) Infiltration- Leakage of air inward into a space through walls, crack openings around doors and windows or through the building materials used in the structure.
18) Latent Cooling Load- The net amount of moisture added to the inside air by plants, people, cooking, infiltration, and any other moisture source. The amount of moisture in the air can be calculated from a combination of dry-bulb and wet-bulb temperature measurements. The latent loads will affect absolute (and relative) humidity.
19) 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.
20) Radiant Heat Gain – is the rate at which heat absorbed by the surfaces enclosing the space and the objects within the space is transferred by convection when the surface or objects temperature becomes warmer than the space temperature.

21) Sensible Cooling Load- The heat gain of the building due to conduction, solar radiation, infiltration, appliances, people, and pets. Burning a light bulb, for example, adds only sensible load to the house. This sensible load raises the dry-bulb temperature.

22) Space Heat gain: The rate at which heat enters to and/or is generated within a space during a time interval.

23) Space Heat loss: The rate at which energy is lost from the space during a time interval.

24) Sensible Heat Gain or Loss – is the heat directly added to or taken away the conditioned space by conduction, convection and/or radiation. The sensible loads will affect dry bulb air temperature.

25) Space Cooling Load – the rate at which energy must be removed from a space to maintain a constant space air temperature. Note that "space *heat gain* ≠ *space-cooling load.*"

26) Space Heat Extraction Rate: The rate at which energy is removed from the space by the cooling and dehumidification equipment. Space heat extraction rate is usually the same as the space-cooling load if the space temperature remains constant.

27) Shading- The effectiveness of a fenestration product plus shade assembly in stopping heat gain from solar radiation is expressed as the Solar Heat Gain Coefficient (SHGC). SHGC values range from 0 to almost 1. The more effective at stopping heat gain, the lower the SHGC value.

28) Solar Heat Gain Coefficient (SHGC) - Solar heat gain coefficient (SHGC) is the ratio of the solar heat gain entering the space through the fenestration area to the incident solar radiation. Solar heat gain includes directly transmitted solar heat and absorbed solar radiation, which is then reradiated, conducted, or convected into the space. Solar Heat Gain Coefficient (SHGC) replaces the Shading Coefficient (SC) used in earlier versions of the standards as a measure of the solar heat gain due to windows and shading devices.

29) Temperature, Dry Bulb – is the temperature of a gas or mixture of gases indicated by an accurate thermometer after correction for radiation.

30) Temperature, Wet Bulb – is the temperature at which liquid or solid water, by evaporating into air, can bring the air to saturation adiabatically at the same temperature.

31) Temperature, Dewpoint – is the temperature at which the condensation of water vapor is a space begins for a given state of humidity and pressure as the temperature of the air is reduced.

32) Thermal conductivity – is the time rate of heat flow through a unit area and unit thickness of a homogenous material under steady conditions when a unit temperature gradient is maintained in the direction perpendicular to the area.

33) Thermal Transmittance or Coefficient of Heat Transfer (U-factor) – is the time rate of heat flow per unit area under steady conditions from the fluid on the warm side of a barrier to the fluid on the cold side, per unit temperature difference between the two fluids.

34) Thermal Conduction – is the process of heat transfer through a material medium in which kinetic energy is transmitted by particles of the material from particle to particle without gross displacement of the particles.

35) Thermal Convection – is the transfer of heat by movement of fluid. Forced convection is the transfer of heat from forced circulation of fluid as by a fan, jet or pump. Natural convection is the transfer of heat by circulation of gas or liquid due to differences in density resulting from temperature changes.

36) Thermally Light Buildings- A building whose heating and cooling requirements are proportional to the weather is considered a thermally light building. That is, when the outdoor temperature drops below the desired room temperature, heating is required and when the outdoor temperature goes above the desired room temperature, cooling is needed. In a thermally light building, the thermal performance of the envelope becomes a dominant factor in energy use and can usually be seen as seasonal fluctuations in utility consumption.

37) Thermally Heavy Buildings- When factors other than weather determine the heating and cooling requirements, the building can be considered thermally heavy. The difference between thermally light and thermally heavy buildings is the amount of heat generated by people, lighting, and equipment within the building. Thermally heavy buildings typically have high internal heat gains and, to a certain extent, are considered to be self-heating and more cooling dominated. This need to reject heat makes them less dependent on the thermal performance of the building envelope.

38) Thermal Weight- A simple "rule of thumb" for determining the thermal weight of a building is to look at heating and cooling needs at an outdoor temperature of 60°Ft. If the building requires heat at this temperature, it can be considered thermally light and if cooling is needed, it is thermally heavy.

39) Ton - A unit of measure for cooling capacity; One ton = 12,000 BTUs per hour

40) U-Factor- The U-factor is the “overall coefficient of thermal transmittance of a construction assembly, in Btu/(hr ft^2 ºF), including air film resistances at both surfaces."

41) Zone- Occupied space or spaces within a building which has its heating or cooling controlled by a single thermostat or zone is a 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.

42) Zoning - A system in which living areas or groups of rooms are divided into separate spaces and each space's heating/air conditioning is controlled independently.

Window Glossary

43) Air leakage rating - Air leakage rating is a measure of the rate of infiltration around a window or skylight. It is expressed in units of cfm/ft^2 of window area or CFM/ft of window perimeter length. The lower a window's air leakage rating, the greater is its air tightness.

44) Conduction- the flow of heat from one particle to another in a material, such as glass or wood, and from one material to another in an assembly, such as a window, through direct contact.

45) Convection - the flow of heat through a circulating gas or liquid, such as the air in a room or the air or gas between windowpanes.

46) Cooling Load Factor (CLF) - CLF is the ratio of actual total cooling compared with total steady-state cooling during the same period at constant ambient conditions.

47) Gas fill - a gas other than air placed between windowpanes to reduce the U-factor by suppressing conduction.

48) Glazing - the glass or plastic panes in a window or skylight.

49) Infiltration - the inadvertent flow of air into a building through breaks in the exterior surfaces of the building. It can occur through joints and cracks around window and skylight frames, sash, and glazing.

50) Low-emittance (low-e) coating - a microscopically thin, virtually invisible, metal or metallic oxide layer deposited on a window or skylight glazing surface to reduce the U-factor or solar heat gain coefficient by suppressing radiative heat flow through the window or skylight.

51) Radiation - the transfer of heat in the form of electromagnetic waves from one separate surface to another. Energy from the sun reaches the earth by radiation, and a person's body can lose heat to a cold window or skylight surface in a similar way.

52) R-value - a measure of the resistance of a material or assembly to heat flow. It is the inverse of the U-factor (R = 1/U) and is expressed in units of hr-ft^2°F/Btu. The higher a window's R-value, the greater are its resistance to heat flow and its insulating value.

53) Shading coefficient - a measure of the ability of a window or skylight to transmit solar heat, relative to that ability for $^1/_8$-in clear, double-strength, single glass. It is equal to the solar heat gain coefficient multiplied by 1.15 and is expressed as a number without units between 0 and 1. The lower a window's shading coefficient, the less solar heat it transmits, and the greater is its shading ability.

54) Solar heat gain coefficient - the fraction of solar radiation admitted through a window or skylight, both directly transmitted and absorbed and subsequently released inward. The solar heat gain coefficient has replaced the shading coefficient as the standard indicator of a window's shading ability. It is expressed as a number without units between 0 and 1. The lower a window's solar heat gain coefficient, the less solar heat it transmits, and the greater is its shading ability.

55) Spectrally selective glazing - a specially engineered low-e coated or tinted glazing that blocks out much of the sun's heat while transmitting substantial daylight.

56) U-factor (U-value) - a measure of the rate of heat flow through a material or assembly. It is expressed in units of Btu/hr-ft^2-°F. Window manufacturers and engineers commonly use the U-factor to describe the rate of non-solar

heat loss or gain through a window or skylight. The lower a window's U-factor, the greater are its resistance to heat flow and its insulating value.

57) Visible transmittance - the percentage or fraction of visible light transmitted by a window or skylight.