STEADY STATE QUANTIFICATION OF ENERGY GAIN/LOSS FROM A BUILDING ENVELOPE

Author: **Devang Chauhan**
Master in Environmental Building Design-13'
cdev@design.upenn.edu

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Advisor: **Yun Kyu Yi**, Ph.D.
Assistant Professor,
Department of Architecture.
School of Design,
University of Pennsylvania

ABSTRACT

In present, when architects are pushed to develop energy saving design strategies, the emphasis on validating the design strategies for their environmental performance is more than ever. Today, many analysis tools are available to architects; however, the tools usually appear as black box for a user who is not familiar with the back side calculation engine that the tool uses. Energy simulation is one of the many types of analysis that helps the architect to test his/her design for its probable energy consumption and thermal behavior. This paper discusses two calculation methods which provide the basis for understanding a calculation engine that a typical energy simulation program uses, and makes one familiar with the variables involved in energy simulation.

KEYWORDS

Heat Balance Method, Radiant Time Series Method (RTS), Conduction Time Series, Time Delay Effect

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CONTENT

1. INTRODUCTION

Today, in design and construction industry, when the emphasis on sustainable design is greater than ever, there has been a push for a holistic design approach, where a design is checked against various environmental, social, and economic indices, for its performance. Being a leading stakeholder in the process of designing and constructing buildings, the architects shall take responsibility of testing their designs across sustainability related indices.

Since inception, architects as professionals have addressed the social aspect of design. Within the construction industry, exists the domain of construction planning and management, which address the economic aspect of design. Now with the advent of mechanical systems for buildings, the emphasis on environmentally sensitive buildings subsided. However, in today's age, when energy savings in buildings, co2 emissions from construction activity, and occupants health are some of the pressing issues, the need for environmental design is imminent.

In order to address the environmental aspect of designs, computational tools are used to evaluate and validate the performance of a design. These computations programs use calculation engines to evaluate the design across a series of variables. In order to make the best use of available computation tools, the user is expected to have fundamental knowledge of thermodynamic heat (energy) exchange between the environment and buildings.

A user familiar with the dynamic nature of energy balance of a building, will be more efficient in setting the variables such as; temperature set points, U-value of envelope, air change ration of AHU, VAV turn down ration, occupant density, lighting density, equipment density, to its optimum value, and will be able to run a successful energy evaluation of building.

This paper will look into two of the load calculation methods for a typical building example.

2. RESEARCH GOAL

To be familiar with the heat balance equation for a building and the variables that affect the heat balance, for us to be able to have a better understanding of a calculation engine used in a typical energy simulation program.

3. BACKGROUND REVIEW

Steven V Szokolay; Introduction to Architectural Science- The Basis of Sustainable Design. 2nd Ed provided a substantial description of various sources of heat gain to the building. Also, an initial heat balance formula was derived and discussed in the book which formed the scope of research. The solar angle calculation equations were used from Szokolay, which were later on used to calculate the solar air temperature.

Chapter 18_Nonresidential cooling and heating load calculation from ASHRAE book of fundamentals provided a detailed discussion on two methods of deriving heat balance equation for a building. Also, detailed equation to calculate the variables affecting the hear balance of a building were provided, which were referred to throughout the research.

Chapter 14_Climate Design Information provided detailed discussion on calculating solar irradiance on horizontal, vertical and tilt surfaces. The calculation was utilized to calculate the solar air temperature for a vertical surface facing south west.

William Bobenhausen; Simplified Design of HVAC systems provided a substantial discussion on fundamental principles of thermodynamics and the passages of heat into buildings. That helped in defining conductive, convective and radiative heat transfer.

4. SCOPE OF RESEARCH

The paper will discuss the steady state heat balance equation for a typical building and will discuss in detail the variables that affect the heat balance.

5. METHOD OF RESEARCH

Two load calculation methods and variables important in both the methods will be discussed in detail, and limitations of both the methods will be identified.

6. RESEARCH

6A. Simplified Steady State Heat Balance   METHOD 01

To be able to calculate the total cooling load of a space, conductive, convective, and radiative energy gain and loss from all the surfaces of space, and convective energy gain for air contained in the space shall be calculated. This makes the calculation process rigorous, and thus historically computational algorithms have been developed to perform these calculations as part of a computer program. Since mid-twentieth century, various programs have been developed with similar heat balance method calculations. The equation for the heat balance method can be understood as discussed below:

In the process of defining the heat balance equation for a space, conduction, convection, and radiation heat transfer from building envelop, and internal heat gains from occupants, lights, and equipment shall be considered. In addition to that, heat migration through air infiltration, moisture migration from openings, and heat gain from system equipment such as; fans and pumps shall also be considered in calculation. The load calculations should describe the building to the fullest. The estimation of loads shall be as accurate as possible and without the use of safety factors.

In order to make the load calculation as accurate as possible, before initiating the calculation procedure, detailed information about the building, its location, construction, occupancy schedule, equipment used, and weather information for pertinent location are gathered. However, due to limitation of human labor skills, the constructed building does not fully reflect the specifications mentioned in the construction drawings and schedules. Therefore, in load calculations, always remains building specification uncertainty. This specification uncertainty results in some approximation of input information while calculating the loads, and thus, it begets calculation uncertainty. The presence of such uncertainties in the calculation process, makes the results approximate.

In order to understand the thermal behavior of buildings, a building can be imagined as a steady state thermal envelope with a series of heat inputs and outputs. Such as;

Q_c = Conduction heat gain or loss
Q_{cv} = Convection heat gain or loss
Q_r= Radiation heat gain between two parallel surfaces
Q_s= Solar heat gain (Radiative)
Q_v = Ventilation heat gain or loss
Q_e= Evaporative heat loss
Q_i= Internal heat gain

Where the system can be depicted by the following equation:

$$\pm Q_c + Q_{cv} + Q_r + Q_s + Q_v \pm Q_e + Q_i = \Delta S$$

Where ΔS is the heat stored in the building
ΔS=0 indicates a thermal balance in the building
ΔS=+ve indicates the building is receiving heat

ΔS= -ve indicates the building is losing heat

In order to understand the equation and the variables involved in the calculation, basic physics of heat and thermodynamics needs to be referred.

What is Heat?
Heat is a form of energy, contained in substances as molecular motion or appearing as electromagnetic radiation in space. In order to fully understand the nature of variables involved in the equation, basic two types of heat gain shall be understood.

Sensible Heat Gain
It is the quantity of heat required to elevate the temperature of unit mass of a substance by one degree. It is measured in units of $\frac{J}{KgK}$ and its value differs for different materials.

For example: The specific heat of metals would be in range of 100-800 $\frac{J}{KgK}$

Whereas, the specific heat for the bricks will be in the range of 800-1200 $\frac{J}{KgK}$

Latent Heat Gain
It is the amount of heat absorbed by a unit mass of a substance at the change of state (from solid to liquid or form liquid to gas), without any change in temperature.

Latent heat fusion (ice to water) at 0℃ = 335 $\frac{KJ}{Kg}$

Whereas, latent heat evaporation at 100℃ = 2261$\frac{KJ}{Kg}$

Now, thermodynamics is the science of heat flow and its relationship with mechanical work. Here, since the discussion is about the heat flow through the building envelope, it becomes important to look at the basic principles of thermodynamics.

The First Law of Thermodynamics
It is the principle of energy conservation. Energy cannot be created or destroyed (except in sub-atomic processes), but only converted form one form to another. It is usually believed that the heat and work are inter-convertible. However, there is usually some loss of energy when work in converted into heat due the efficiency of engine.

The Second Law of Thermodynamics
Heat transfer can take place simultaneously in one direction only. From hotter to cooler body, or generally speaking, from higher to lower grade. In any machine, in order to have heat flow, there is a heat source (higher grade) and a heat Sink (lower grade). Heat flow from higher grade to lower grade can take place in three forms: Conduction, Convection, and Radiation.

Conduction occurs within a body or bodies in contact, when the heat is transmitted by movement of molecules. Conduction heat gain can be calculate as,

$$\mathbf{Q_c = A * U * \Delta T}$$

Where, A = Area of surface (m^2)

U = u-value of envelope ($\frac{W}{m^2K}$)

ΔT = The difference of outside temperature and inside temperature (K)

The solar heat that the surfaces receive, does not immediately released inside the building. Usually there is some hours of delay before the received heat is emitted back into the building. This emission of heat is usually convective and thus it can be added into cooling load calculations. In order to take delay effect into consideration, conduction time factors must be used in calculations. RTS method (Radiation Time Series) effectively takes conduction factors into consideration.

Convection occurs from a solid body to a fluid (liquid or gas) or vice versa. It is also transfer of heat from one surface to another via medium of a fluid. The magnitude of convection heat transfer depends on

- Area of contact (m^2)
- The difference in temperature of surface and body ΔT (K),
- Convection coefficient h_c ($\frac{W}{m^2K}$)

A convection coefficient depends on the viscosity of fluid and its flow velocity, as well as on the physical configuration that will determine whether the flow is laminar or turbulent.

Convection heat gain can be calculated as,

$$\mathbf{Q_{cv} = A \times h_c \times \Delta T}$$

Where, A = Area of surface (m^2)

h_c = Convection coefficient ($\frac{W}{m^2K}$)

$\Delta T =$ The difference of outside temperature and inside temperature (K)

For vertical surfaces h_c = 3 $\frac{W}{m^2K}$

For horizontal surfaces,

When heat flow is upward (from floor to ceiling), h_c = 4.3 $\frac{W}{m^2K}$

When heat flow is downward (ceiling to floor), h_c = 1.5 $\frac{W}{m^2K}$

In above all situations, the air is assumed to be still. If the surface is exposed to wind, or mechanically generated air movement, or if it is forced convection, the convection coefficient is relatively higher:

h_c = 5.8 + 4.1V (V = velocity in$\frac{m}{s}$)

Radiation is heat transfer form a body with a warmer surface to another which is cooler. The radiation usually has electromagnetic wavelength and the wavelength depends on the surface temperature of heat source. Usually radiation heat transfer is proportional to the difference of the 4th power of absolute temperatures of the emitting and receiving surfaces and depends on their surface qualities, such as; reflectance, absorptance, and emittance.

Since the solar heat gain to building envelope form sun is mainly radiation, it could be calculated as

$$\mathbf{Q_{gain} = A \times G \times \alpha}$$

Where, A = Area of surface (m^2)

G = Incident solar radiation ($\frac{W}{m^2}$)

$\alpha =$ Absorptance of surface (Non-dimensional)

The heat input will elevate the surface temperature and later on this heat will be dissipated to the environment. This loss of heat will depend on the conductance of the surface.

Therefore, $\mathbf{Q_{loss} = A * h \times (T_s - T_o)}$
As the surface temperature increases, equilibrium will be reached when $\mathbf{Q_{gain} = Q_{loss}}$

Therefore, $\mathbf{G \times A \times \alpha = A \times h \times (T_s - T_o)}$
Therefore, $\mathbf{T_s = T_o + G + \frac{\alpha}{h}}$
Or, $\mathbf{T_s = T_o + G \times (\alpha \times R_{so})}$

The equations neglect any heat flow from the surface into the body of the element, therefore $\mathbf{T_s}$ is a notional sol-air temperature $\mathbf{T_{sa}}$, which is the driving force of heat flow.
For a surface exposed to sky, a radiant emission term should be included in the sol-air temperature expression.

Therefore, $\mathbf{T_s = T_o + (G * (\alpha - \varepsilon)) \times R_{so}}$

The radiant emission is usually taken as $\mathbf{\varepsilon = 90 \frac{w}{m^2}}$ for a cloudless sky and $\mathbf{\varepsilon = 20 \frac{w}{m^2}}$ for a cloudy sky.
The air temperature $\mathbf{T_o}$ is same all around the building, but the solar-air temperature $\mathbf{T_{sa}}$ is different for all the side due to different orientation of different surfaces in relation with the sun.

The heat gain due to $\mathbf{dT_e}$ will be $\mathbf{G \times (\alpha \times R_{so})}$ for walls, and $\mathbf{(G \times (\alpha - \varepsilon)) \times R_{so}}$ for surfaces exposed to sky

The heat gain due to $\mathbf{T_o}$ is calculated as conduction heat transfer $\mathbf{Q_c = A \times U \times \Delta T}$
Therefore, solar heat gain from an opaque surface will be

$$\mathbf{Q_s = Q_c + \ dT_e}$$

Solar heat gain though a transparent surface is relatively simple, it could be calculated as

$$\mathbf{Q_s = A \times G \times SHGC}$$

Where, A = Area of glass ($\mathrm{m^2}$)
G = Incident solar radiation ($\frac{W}{m^2}$)
SHGC = Percentage of radiation received on the other side of glass (percentage value)

Now, in case of double skin façade, or where there's cavity in façade, radiative heat transfer can occur between outside and inside surfaces, that heat transfer can be calculated as,

$$\mathbf{Q = A * \sigma \times \varepsilon \times [\left(\frac{T'}{100}\right)^4 - \left(\frac{T''}{100}\right)^4]}$$

Where, A = Area of surface ($\mathrm{m^2}$)
$\boldsymbol{\sigma}$ = Stefan Boltzman Constant (Non-Dimensional)

$\boldsymbol{\varepsilon} =$ Effective emittance
$\mathbf{T'}$= Temperature of outside surface
$\mathbf{T''}$ = Temperature of inside surface

Effective emittance $\boldsymbol{\varepsilon}$ can be derived as $\frac{\mathbf{1}}{\boldsymbol{\varepsilon}} = \frac{\mathbf{1}}{\boldsymbol{\varepsilon'}} + \frac{\mathbf{1}}{\boldsymbol{\varepsilon''}} - \mathbf{1}$
Similar to a convection coefficient, a radiative coefficient can also be derived.

$$\mathbf{h_r = 5.7 \times \boldsymbol{\varepsilon} \times \frac{(\frac{T'}{100})^4 - (\frac{T''}{100})^4}{t' - t''}}$$

Then radiation heat transfer between two parallel surfaces can be defined as,
$\mathbf{Q_r = h_r \times A \times (t' - t'')}$

Where typically,
$\mathbf{h_r = 5.7 \times \boldsymbol{\varepsilon}}$ at 20°C and, $\mathbf{h_r = 4.6 \times \boldsymbol{\varepsilon}}$ at 0°C.

ε=0.9 for ordinary building surface
ε=0.2 for dull aluminium
ε=0.05 for polished aluminium
T= Temperature in Kelvin
t= is temperature in Celsius

The solar heat that the surfaces receive, does not immediately released inside the building. Usually there is some hours of delay before the received heat is emitted back into the building. This emission of heat is usually convective and thus it can be added into cooling load calculations. In order to take delay effect into consideration, radiation time factors must be used in calculations. RTS method (Radiation Time Series) effectively takes radiation factors into consideration.

Ventilation is used for three different processes and it is used for three different purposes:
1. Supply fresh air to remove smells. Co2 and other contaminants
2. Remove some internal heat when $T_o < T_i$
3. To promote heat dissipation from the skin, physiological cooling effect.

If the ventilation rate Vr is known, then ventilation heat flow rate can be found out as

$$\mathbf{qv = 1200\ (\frac{J}{m^3K}) \times vr(\frac{m^3}{s})}$$

Where, 1200 $\frac{J}{m^3K}$ is the volumetric heat capacity of moist air. Often the air changes per hour is known, and from that the ventilation rate can be found out as

$$\mathbf{vr = N \times \frac{V}{3600}\ (\frac{m^3}{s})}$$

Therefore, $\mathbf{qv = 0.33 \times N \times V}$ and heat flow rate due to ventilation will be

$$\mathbf{Q_v = qv \times \Delta T}$$

Evaporative is cooling one of the passive strategies. However, when evaporation happens, it may lower the dry bulb temperature, but it increases the humidity, therefore the latent heat content. In effect it converts the sensible heat into latent heat. And thus, the total heat content of the system doesn't change.

This moisture then can be removed by ventilation and that will be called the mass transfer effect. So if the evaporation rate is known, the corresponding heat loss will be

$$\mathbf{Q_e = \left(\frac{2400}{3000}\right) \times er}$$

Where 2400 is latent heat of evaporation of water $\frac{\mathrm{Kj}}{\mathrm{Kg}}$
And er is evaporation rate $\frac{\mathrm{Kg}}{\mathrm{h}}$

Internal Gain is usually calculated as the sum of heat input from equipment, lights, and occupants.
Therefore,

$$\mathbf{Q_i = W_o + W_l + W_e}$$

Where, W_o is the heat input form occupants, and that depends on the activity of occupants. As a result of metabolic activity, the humans emit heat to the environment. This heat gain is significant for spaces such as; auditorium, theatres, and meeting rooms, where for a short period of time there is high occupancy. The heat emitted by occupants needs to be fractioned into sensible and latent heat, where the sensible heat gain is typically radiant.

Activity	**at 20˚C**		**at 26˚C**	
	Sensible	**Latent**	**Sensible**	**Latent**
Seated at Rest	90	25	65	50
Sedentary Work	100	40	70	70
Seated & Eating	85	65	70	80
Slow Walking	110	50	75	85
Light Bench work	130	105	80	55
Medium Work	140	125	90	175
Heavy Work	190	250	105	335
Very Heavy Work	205	350	175	420

Table 1 [ASHRAE standard 90.1-2007]

$\mathbf{W_e}$ **&** $\mathbf{W_i}$ are heat input from light sources and equipment, and that can be calculated as follow:
Consumption rate of equipment/light source = P (power)
Time of use = t (time)
Energy use (Wh) = Power X Time
Calculate the Energy use in Wh for all the equipment and take average for whole day

Therefore,

$$\mathbf{W_l + W_e = \frac{Total\ Energy\ Use\ (Wh)}{24\ (h)}}$$

<u>Heat gain from Lighting</u>: The primary source of heat from lighting comes from light-emitting elements, or lamps, although significant additional heat may be generated from ballasts and other appurtenances in the luminaires. Generally, the instantaneous rate of sensible heat gain from electric lighting may be calculated from

$$\mathbf{Q_{el} = W \times F_{ul} \times F_{sa}}$$

Where,
qel = heat gain, W
W = total light wattage, W
Ful = lighting use factor
Fsa = lighting special allowance factor

The total light wattage is obtained from the ratings of all lamps installed, both for general illumination and for display use. Ballasts are not included, but are addressed by a separate factor. Wattages of magnetic ballasts are significant; the energy consumption of high efficiency electronic ballasts might be insignificant compared to that of the lamps.

$$\text{The lighting Use Factor} = \frac{\text{Wattage in Use, Under present condition}}{\text{Total Wattage Installed}}$$

For commercial applications such as stores, the use factor is generally 1.0.

$$\text{The Special Allowance Factor} = \frac{\text{Fixture's Power Consumption (Lamp + Ballast)}}{\text{Nominal Power Consumption}}$$

For incandescent lights, this factor is 1. For fluorescent lights, it accounts for power consumed by the ballast as well as the ballast's effect on lamp power consumption. The special allowance factor can be less than 1 for electronic ballasts that lower electricity consumption below the lamp's rated power consumption. For high-intensity-discharge lamps, such as; metal halide, mercury lamp, and for sodium vapor lamps, the actual power consumption should be available form manufacturer for lamps and ballast. In case of high-intensity-discharge lamps, the ballast would have allowance factor different than the typical ballasts, and that should also be available from the manufacturer. In situations, when the lighting powers use for each luminaire is not available, the lighting power density shall be taken into consideration for the particular space use. In this method, the lighting heat gain is calculated on square foot or square meter basis. The lighting power density for different types of space use shall be obtained from ASHRAE standard 90.1-2007.

In addition to calculating the heat gain from lighting, to include this heat gain in cooling load calculation, the heat gain from lighting needs to be factored in components. Where one component will enter the conditioned space and will affect the cooling load of a space, whereas, the other will enter the plenum space and affect the return or supply air temperature. Also, the component of light heat gain entering the conditioned space shall further be divided in radiative and convective components before considering them in cooling load calculations. The information on space vs plenum component, and radiative vs convective component, can be obtained for particular type of luminaire. ASHRAE Handbook-Fundamentals provides information on recessed fluorescent luminaire without lens.

Heat gain from Equipment: Equipment are present in all types of spaces, and can contribute a good amount of internal heal gain. Therefore, the loads from them must be taken into consideration while calculating the cooling load for a space. The calculation of heat gain from equipment of cooking use, clinical use, and office use will be discussed in this paper.

Cooking equipment are mainly electric, gas fired, or steam fired. For the calculation method mentioned in the paper, the cooking equipment are assumed to be placed in a conditioned kitchen, and hooded with a good quality hood and sufficient exhaust supply. The sensible heat gain from any cooking equipment can be divided into radiative and convective components. The gas fired equipment typically contribute more towards the cooling load of a space due to a higher radiative component. It is observed that when the equipment are installed with a hood, only radiant gains are added to the cooling load of that space, convective and latent heat from the cooking and combustion products are exhausted and do not enter the space. The heat gain for cooking equipment can be calculated as,

$$\mathbf{q_s = q_{input} \times F_U \times F_R}$$

Where,

$$F_R \text{ (Radiation Factor)} = \frac{\text{Radiaiton heat gain}}{\text{Total Energy Consumed}}$$

$$F_U \text{ (Use Factor)} = \frac{\text{Energy in use}}{\text{Total Energy Consumed}}$$

q_{input} = Total energy used

Heat gain from meals can also be taken into consideration for cooling load calculation. Considering 15 W of heat input from each meal served, 75% of total heat gain will be considered sensible and 25% of total heat gain will be taken as latent.

Heat gain from Laboratory equipment: commonly, heat gain from office equipment in a laboratory setting range from 50 to 220 $\frac{W}{m^2}$

Heat gain from office equipment: the heat gain from office equipment can be taken as 25% to 50% of total wattage installed for equipment with nameplate power consumption less than 1000 W. Generally, the nameplate value is only known and no actual heat gain data are available for similar equipment, in such cases, it is conservative to use 50% of nameplate as heat gain and more accurately 25% of nameplate value. The heat load calculation for equipment primarily depends on the nameplate values, which are mostly available for such type of equipment due to their consumer use. However, in case when the number and type of equipment to be used in a space is not decided, equipment power density value can be used for that particular space type.

Infiltration is another source of heat gain to a building, which traditionally is not part of heat balance calculation for a building. When the building is in cooling mode, the outside air which has higher temperature has a tendency to enter the building where a low temperature is maintained. Infiltration will be prominent in the zones which are located on the wind ward side of prevailing wind. However, downward stack effect, when the indoor air is denser than the outdoor air can eliminate some portion of air infiltration. Nevertheless, the upper floors may still suffer some infiltration when the building is in cooling mode. Designers usually predict overall rates of infiltration using the number of air changes per hour. Commonly, for a typical building located in Central America, the air change rate per hour is derived for heating period, and then half of that is considered for cooling period.

Moisture Diffusion is a natural phenomenon and is always present in building. Although it is insignificant in relation with other heat gains, it can reduce the insulation value of an envelope if capacitive insulation is used. Therefore, it is usually not desired to allow the moisture to penetrate the building envelope, and thus vapor retarders are often integrated into envelope construction. The heat gain form moisture diffusion can be calculated as,

$$\mathbf{q_m = M \times A \times \Delta P_v \times (h_g - h_f)}$$

Where,

q_m= latent heat gain form moisture migration, W

M= permeance of wall or roof assembly $\frac{ng}{(s \times m^2 \times P_{a)}}$

A= area of wall or roof surface, m^2

ΔP_v= is vapor pressure difference

h_g= is enthalpy at room condition, $\frac{Kj}{Kg}$

h_f= enthalpy of water condensed at cooling coil, $\frac{Kj}{Kg}$

$h_g - h_f$ = 2500$\frac{Kj}{Kg}$, when room temperature is 24℃ and condensation coil is 10℃

Observations

Due to prevailing building specification uncertainty and simulation uncertainly, lot of simplification assumptions are made in order to perform calculations with this method. This simplification makes the result of the calculation procedure approximate.

Some examples of approximation in this calculation method are as follow:

- The temperature of air is assumed to be constant throughout the space
- For conduction processes, the heat flow is assumed to be always normal to the plane of surface
- All the particles of a surface are assumed to have similar temperature
- The method does not account for the thermal bridges that are common in all the structures due to complexity of geometry and various materials used.

Various external variables are involved in the calculation procedure. For example, variables associated with weather data, location, wind, etc. the architect needs to depend on external sources for information on such variables, which does not allow one to use this method independently. Also, the calculation for conduction, convection, and radiation components is performed for particular time only. For better understanding of thermal; behavior of a space, at least a 24- hour cycle calculation needs to be performed.

6B. RADIANT TIME SERIES — METHOD 02

The radiant time series method is further development of heat balance method discussed in prior section of this paper. The fundamental improvement in RTS method over heat balance method is inclusion of time delay effect of conduction and radiation heat transfer in heat balance calculation. This method is suitable for peak design load calculation, but shall not be used for annual energy simulation because of intensive repetitive nature of calculations and dependence on external data sources.

In any building, external wall surfaces, windows, and roof receive solar heat gain. Walls and roofs pass on this heat indoors via conduction, whereas, windows radiate the heat indoors. The conductive heat transfer does not directly translate into cooling load of space. Due to the mass and thermal capacity of wall and roof materials, there is substantial delay before the heat received by walls and roof is released into indoor air space. Similarly, the heat received by solar radiation must first be absorbed by the walls, floors, and roof surfaces of the building, and then that heat is released back into the air space with a sufficient time delay.

In RTS method, this delay effect of conduction and radiation heat transfer is addressed by introducing Conduction Time Factor and Radiation Time Factor. These factors reflect the percentage of an earlier conduction and radiant heat gain that becomes cooling load during the current hour. Each radiant and conduction time series must total 100% of heat gain.

Conducting successful energy simulation for a building requires extensive data entry, however, during the design stage, all information is not available. In such a situation, an architect is required to make calculated assumptions. By using RTS method, an architect can compare various envelope options for their influence on building heat balance. This characteristic of RTS method makes it useful for design stage calculations and evaluations.

The RTS method will be discussed by using it to calculate loads for a building example located in Atlanta, USA.

Single Room Example: The room is on the second floor of a two-story building and has two vertical exterior exposures, with a flat roof above.

Area: 25.47 m2
Floor: Carpeted 127 mm concrete slab on metal deck above a conditioned space.
Roof: Flat metal deck topped with rigid mineral fiber insulation and perlite board (R = 2.2), felt, and light-colored membrane roofing. Space above 2.75 m suspended acoustical tile ceiling is used as a return air plenum. Assume 30% of the cooling load from the roof is directly absorbed in the return airstream without becoming room load. Use roof U = 0.40 W/(m2·K).
Spandrel wall: Spandrel bronze-tinted glass, opaque, backed with air space, rigid mineral fiber insulation (R = 0.88), mineral fiber batt insulation (R = 0.88), and 16 mm gypsum wall board. Use spandrel wall U = 0.45 W/(m2/K).
Brick wall: Light-brown-colored face brick (102 mm), mineral fiber batt insulation (R = 1.76), lightweight concrete block (152 mm) and gypsum wall board (16 mm). Use brick wall U = 0.45 W/(m2·K).

Windows: Double glazed, 6 mm bronze-tinted outside pane, 13 mm air space and 6 mm clear inside pane with light-colored interior mini blinds. Window normal solar heat gain coefficient (SHGC) = 0.49. Windows are non-operable and mounted in aluminum frames with thermal breaks having overall combined U = 3.18 W/(m^2·K).Inside attenuation coefficients (IACs) for inside mini blinds are based on light venetian blinds

(assumed louver reflectance = 0.8 and louvers positioned at 45° angle) with heat-absorbing double glazing, IAC(0) = 0.74, IAC(60) = 0.65, IAD(diff) = 0.79, and radiant fraction = 0.54. Each window is 1.91 m wide by 1.95 m tall for an area per window = 3.72 m2.
South exposure: Orientation = 30° east of true south
Window area = 3.72 m^2
Spandrel wall area = 5.57 m^2
Brick wall area = 5.57 m^2
West exposure: Orientation = 60° west of south
Window area = 7.43 m^2
Spandrel wall area = 11.15 m^2
Brick wall area = 6.97 m^2
Occupancy: 12 people from 8:00 AM to 5:00 PM.
Lighting: Four 3-lamp recessed fluorescent 600 by 1200 mm parabolic reflector (without lens) type with side slot return-air-type fixtures. Each fixture has three 32 W T-8 lamps plus electronic ballasts, for a total of 110 W per fixture or 440 W total for the room. Operation is from 7:00 AM to 7:00 PM. Assume 26% of the cooling load from lighting is directly absorbed in the return air stream without becoming room load.
Equipment: Several computers and a video projector may be used, for which an allowance of 10.76 W/m2 is to be accommodated by the cooling system, for a total of 274 W for the room. Operation is from 8:00 AM to 5:00 PM.
Infiltration: For purposes of this example, assume the building is maintained under positive pressure during peak cooling conditions and therefore has no infiltration. Assume that infiltration during peak heating conditions is equivalent to one air change per hour.
Weather data: for Atlanta, Georgia, latitude = 33.64, longitude = 84.43, elevation = 313 m above sea level, 99.6% heating design dry-bulb temperature = –6.3°C. For cooling load calculations, use 5% dry-bulb/coincident wet-bulb monthly design day profile.
Inside design conditions: 22.2°C for heating; 23.9°C with 50% rh for cooling.

Heat Gain from Lighting: The heat output from lighting is split into radiative and convective components. Appropriate radiant time series applied to radiant component to convert it into convective component. Later on, the total convective component for lighting is derived.

$$\mathbf{Q_{el} = W \times F_{ul} \times F_{sa}}$$

Where,
qel = heat gain, W
W = total light wattage, W
Ful = lighting use factor
Fsa = lighting special allowance factor

The lighting heat gain profile, based on the occupancy schedule indicated is:

q1 = (440 W)(0%) = 0	q13 = (440 W)(100%) = 440
q2 = (440 W)(0%) = 0	q14 = (440 W)(100%) = 440
q3 = (440 W)(0%) = 0	q15 = (440 W)(100%) = 440
q4 = (440 W)(0%) = 0	q16 = (440 W)(100%) = 440
q5 = (440 W)(0%) = 0	q17 = (440 W)(100%) = 440
q6 = (440 W)(0%) = 0	q18 = (440 W)(100%) = 440

q7 = (440 W)(100%) = 440 q19 = (440 W)(0%) = 0
q8 = (440 W)(100%) = 440 q20 = (440 W)(0%) = 0
q9 = (440 W)(100%) = 440 q21 = (440 W)(0%) = 0
q10 = (440 W)(100%) = 440 q22 = (440 W)(0%) = 0
q11 = (440 W)(100%) = 440 q23 = (440 W)(0%) = 0
q12 = (440 W)(100%) = 440 q24 = (440 W)(0%) = 0

The convective portion is simply the lighting heat gain for the hour being calculated times the convective fraction for recessed fluorescent lighting fixtures without lens and with side slot return air.

Q_c15 = (440)(52%) = 229 W

The radiant portion of the cooling load is calculated using lighting heat gains for the current hour and past 23 h, the radiant fraction (48%), and radiant time series, in accordance with Equation.

$$\mathbf{Q_r\theta = r_0 q_r\theta + r_1 q_r\theta - 1 + r_2 q_r\theta - 2 + \ldots + r_{23} q_r\theta - 23}$$

Where,
$Q_r\theta$ = radiant cooling load Q_r for current hourθ, W
$q_r\theta$ = radiant heat gain for current hour, W
$q_r\theta - n$ = radiant heat gain n hours ago, W
r_0, r_1 etc. = radiant time factors

Radiant time series is selected for medium weight construction, assuming 50% glass and carpeted floor as representative of the described construction. Thus, the radiant cooling load for lighting is

Qr15 = r0(0.48)q15 + r1(0.48)q14 + r2(0.48)q13 + r3(0.48)q12+ … + r23(0.48)q16
= (0.49)(0.48)(440) + (0.17)(0.48)(440) + (0.09)(0.48)(440) + (0.05)(0.48)(440) + (0.03)(0.48)(440)
+ (0.02)(0.48)(440) + (0.02)(0.48)(440) + (0.01)(0.48)(440)
+ (0.01)(0.48)(440) + (0.01)(0.48)(0) + (0.01)(0.48)(0)
+ (0.01)(0.48)(0) + (0.01)(0.48)(0) + (0.01)(0.48)(0)
+ (0.01)(0.48)(0) + (0.01)(0.48)(0) + (0.01)(0.48)(0)
+ (0.01)(0.48)(0) + (0.01)(0.48)(0) + (0.01)(0.48)(0)
+ (0.00)(0.48)(0) + (0.00)(0.48)(440) + (0.00)(0.48)(440)
+ (0.00)(0.48)(440)
= 188 W

The total lighting cooling load at the designated hour is thus;

Qlight = Qc15 + Qr15 = 229 + 188 = 417 W

As noted in the example definition, if it is assumed that 26% of the total lighting load is absorbed by the return air stream, the net lighting cooling load to the room is

Qlight-room15 = Qlight15 (74%) = 417(0.74) = 309 W

Heat Gain form External Wall: the calculation will be performed for spandrel wall section facing 60° west of south at 3:00 PM local standard time, in July for previously described conference room. In order to calculate the heat gain from an external wall, the solar air temperature at the surface will be calculated. Based on that temperature, the heat input will be derived. This heat will be released indoors after certain time delay. Therefore conduction time series and radiation time series will be used to calculate final convective heat transfer to indoor air space.

Total clear-sky irradiance: Et reaching the receiving surface is the sum of three components: the beam component Etb originating from the solar disc; the diffuse component Etd, originating from the sky dome; and the ground-reflected component Etr originating from the ground in front of the receiving surface. Thus,

Et = Etb + Etd + Etr

Beam Component**:** The beam component is obtained from a straight forward geometric relationship:

Etb = Eb Cos θ

Where, θ is the angle of incidence. This relationship is valid when Cos θ > 0; otherwise, Etb = 0.

Diffuse Component**:** The diffuse component is more difficult to estimate because of the no isotropic nature of diffuse radiation; some parts of the sky, such as the circumsolar disc or the horizon, tend to be brighter than the rest of the sky, which makes the development of a simplified model challenging. For vertical surfaces, the ratio Y of clear-sky diffuse irradiance on a vertical surface to clear-sky diffuse irradiance on the horizontal is a simple function of the angle of incidence θ:

Etd = EdY with **Y = max (0.45, 0.55 + 0.437 Cos θ + 0.313** $\mathbf{Cos^2}$ **θ)**

For a vertical surface with slope Σ, the following simplified relationships are sufficient for most applications described in this volume:

Etd = Ed(Y Sin Σ + Cos Σ) if Σ ≤ 90°
Etd = EdY Sin Σ if Σ > 90°

Where, Y is calculated for a vertical surface having the same azimuth as the receiving surface considered.

Ground-Reflected Component**:** Ground-reflected irradiance for surfaces of all orientations is given by

$$\mathbf{E_{tr} = (E_b Sin\beta + E_d)\rho_g(\frac{1 - Cos\,\Sigma}{2})}$$

Where, ϱg is ground reflectance. This is often taken to be 0.2 for a typical mixture of ground surfaces. Ground reflectance for other surfaces can be obtained from ASHRAE-Handbook of Fundamentals
Solar-air temperature at 3:00 PM local standard time (LST) (4:00 PM daylight saving time) on July 21 for a vertical, dark-colored wall surface, facing 60° west of south, located in Atlanta, Georgia.
Latitude = 33.64,
Longitude = 84.43,
Clear Sky Optical depth for beam irradiance (taub) = 0.556,

Clear Sky Optical depth for diffuse irradiance (taud) = 1.779 from monthly Atlanta weather data for July. The calculated outdoor design temperature for that month and time is 33.3°C.
The ground reflectivity is assumed ϱg = 0.2.
For the dark colored wall, α/ho = 0.053,
Vertical surfaces, $\varepsilon\Delta R/ho$ = 0.

Sol-air temperature is calculated using following equations:

$$\mathbf{t_e = t_o + \frac{\alpha E_t}{h_o} - \frac{\varepsilon \Delta R}{h_o}}$$

Where,
α = absorptance of surface for solar radiation
Et = total solar radiation incident on surface, W/m2
ho = coefficient of heat transfer by long-wave radiation and convection at outer surface, W/(m2·K)
to = outdoor air temperature, °C
ts = surface temperature, °C
ε = hemispherical emittance of surface
ΔR = difference between long-wave radiation incident on surface from sky and surroundings and radiation emitted by blackbody at outdoor air temperature, W/m2

Solar Angles:
ψ = southwest orientation = +60°
Σ = surface tilt from horizontal (where horizontal = 0°) = 90°
For vertical wall surface 3:00 PM, Local Standard Time (LST) = hour 15
Solar altitude, solar azimuth, surface solar azimuth, and incident angle are as follow:
Equation Time (ET) = –6.4 min
Declination Angle (δ) = 20.4°
Solar Radiation (Eo) = 1324 W/m2

Local standard meridian (LSM) for Eastern Time Zone = 75°.
Apparent solar time AST = LST + ET/60 + (LSM – LON)/15
= 15 + (–6.4/60) + [(75 – 84.43)/15]
= 14.2647

Hour angle H = 15(AST – 12)
= 15(14.2647 – 12)
= 33.97°

Solar altitude β
Sin β = Cos L Cos δ Cos H + Sin L Sin δ
= Cos (33.64) Cos (20.4) Cos (33.97) + Sin (33.64) Sin (20.4)
= 0.841
β = Sin–1(0.841) = 57.2°

Solar azimuth φ
Cos φ = (Sin β Sin L – Sin δ)/(Cos β Cos L)

= [(Sin (57.2) Sin (33.64) – Sin (20.4)]/[Cos (57.2) Cos (33.64)]
= 0.258
φ = Cos–1(0.253) = 75.05°

Surface-solar azimuth γ
$\gamma = \varphi - \psi$
= 75.05 – 60
= 15.05°

Incident angle θ
Cos θ = Cos β Cos g Sin Σ + Sin β Cos Σ
= Cos (57.2) Cos (15.05) Sin (90) + Sin (57.2) Cos (90)
= 0.523
θ = Cos–1(0.523) = 58.5°

Beam normal irradiance
Eb = Eo exp(–τbmab)
m = relative air mass
= 1/[Sinβ +0.50572(6.07995 + β)–1.6364], β expressed in degrees
= 1.18905

ab = beam air mass exponent
= 1.219 – 0.043τb – 0.151τd – 0.204τbτd
= 0.72468

Eb = 1324 exp[–0.556(1.89050.72468)]
= 705 W/m2

Surface beam irradiance Etb = Eb Cos θ
= (705) Cos (58.5)
= 368 W/m2

Ratio Y of sky diffuse radiation on vertical surface to sky diffuse radiation on horizontal surface

Y = 0.55 + 0.437 Cos θ + 0.313 $\text{Cos}^2\theta$
= 0.55 + 0.437 Cos (58.5) + 0.313 Cos^2(58.5)
= 0.864

Diffuse irradiance Ed – Horizontal surfaces
Ed = Eo exp(–τdmad)

ad = diffuse air mass exponent
= 0.202 + 0.852τb – 0.007τd – 0.357τbτd
= 0.3101417

Ed = Eo exp(–τdmad)
= 1324 exp(–1.779(1.89050.3101)]

= 203 W/m2

Diffuse irradiance Ed – Vertical surfaces

Etd = EdY

= (203)(0.864)

= 175 W/m2

Ground reflected irradiance Etr = (Eb Sin β + Ed)ϱg(l – Cos Σ)/2

= [705 Sin (57.2) + 203](0.2)[1 – Cos (90)]/2

= 80 W/m2

Total surface irradiance Et

Et = Etb + Etd + Etr

= 368 + 175 + 80

= 623 W/m2

Sol–air temperature

Te = to + αEt /ho – εΔR/ho

= 33.3 + (0.053)(623) – 0

= 66.3°C

This procedure is used to calculate the sol-air temperatures for each hour on each surface. Because of the tedious solar angle and intensity calculations, using a simple computer spreadsheet or other computer software can reduce the effort involved.

Conductive heat gain is calculated using following equations:

$$\mathbf{Q_{i,\theta-n} = UA(t_{e,\theta-n} - t_{rc})}$$

Where,

$Q_{i,\theta-n}$ = conductive heat input for the surface n hours ago, W

U = overall heat transfer coefficient for the surface, W/(m2·K)

A = surface area, m2

$t_{e,\theta-n}$ = sol-air temperature n hours ago, °C

t_{rc} = presumed constant room air temperature, °C

Conductive heat gain through walls or roofs can be calculated using conductive heat inputs for the current hours and past 23 h using conduction time series:

Qθ = C0Qi,θ + C1Qi,θ-1 + C2Qi,θ-2 + C3Qi,θ-3 + … + C23Qi,θ-23

Where,

Qθ = hourly conductive heat gain for the surface, W

Qi,θ = heat input for the current hour

Qi,θ-n = heat input n hours ago

C0, C1, etc. = conduction time factors

Therefore,

Qi,1 = (0.45)(11.15)(23.2 – 23.9) = 0–3 W

Qi,13 = (0.45)(11.15)(51.9 – 23.9) = 142

$Q_{i,2} = (0.45)(11.15)(22.8 - 23.9) = 0–6$
$Q_{i,3} = (0.45)(11.15)(22.4 - 23.9) = 0–8$
$Q_{i,4} = (0.45)(11.15)(22.1 - 23.9) = 0–9$
$Q_{i,5} = (0.45)(11.15)(21.8 - 23.9) = -10$
$Q_{i,6} = (0.45)(11.15)(22.6 - 23.9) = 0–7$
$Q_{i,7} = (0.45)(11.15)(25.8 - 23.9) = 010$
$Q_{i,8} = (0.45)(11.15)(29.9 - 23.9) = 031$
$Q_{i,9} = (0.45)(11.15)(34.0 - 23.9) = 051$
$Q_{i,10} = (0.45)(11.15)(37.4 - 23.9) = 068$
$Q_{i,11} = (0.45)(11.15)(40.3 - 23.9) = 083$
$Q_{i,12} = (0.45)(11.15)(42.9 - 23.9) = 096$
$Q_{i,14} = (0.45)(11.15)(60.8 - 23.9) = 187$
$Q_{i,15} = (0.45)(11.15)(66.3 - 23.9) = 215$
$Q_{i,16} = (0.45)(11.15)(67.1 - 23.9) = 219$
$Q_{i,17} = (0.45)(11.15)(62.7 - 23.9) = 196$
$Q_{i,18} = (0.45)(11.15)(52.6 - 23.9) = 145$
$Q_{i,19} = (0.45)(11.15)(36.7 - 23.9) = 065$
$Q_{i,20} = (0.45)(11.15)(27.6 - 23.9) = 019$
$Q_{i,21} = (0.45)(11.15)(26.6 - 23.9) = 014$
$Q_{i,22} = (0.45)(11.15)(25.5 - 23.9) = 008$
$Q_{i,23} = (0.45)(11.15)(24.6 - 23.9) = 004$
$Q_{i,24} = (0.45)(11.15)(23.9 - 23.9) = 000$

the conduction time factors for wall 1 can be used in conjunction with the 24 h heat input profile to determine the wall heat gain at 3:00 PM LST:

$$
\begin{aligned}
Q_{15} &= C_0Q_{i,15} + C_1Q_{i,14} + C_2Q_{i,13} + C_3Q_{i,12} + \ldots + C_{23}Q_{i,14} \\
&= (0.18)(215) + (0.58)(187) + (0.20)(142) + (0.04)(96) \\
&+ (0.00)(83) + (0.00)(68) + (0.00)(51) + (0.00)(31) \\
&+ (0.00)(10) + (0.00)(-7) + (0.00)(-10) + (0.00)(-9) \\
&+ (0.00)(-8) + (0.00)(-6) + (0.00)(-3) + (0.00)(0) \\
&+ (0.00)(4) + (0.00)(8) + (0.00)(14) + (0.00)(19) \\
&+ (0.00)(65) + (0.00)(145) + (0.00)(196) + (0.00)(219) \\
&= 179 \text{ W}
\end{aligned}
$$

Finally, calculate wall cooling load using radiant time series. Total cooling load for the wall is calculated by summing the convective and radiant portions. The convective portion is simply the wall heat gain for the hour being calculated times the convective fraction for walls from (54%):

$Q_{c15} = (179)(0.54) = 97$ W

The radiant portion of the cooling load is calculated using conductive heat gains for the current and past 23 h, the radiant fraction for walls (46%), and radiant time series, in accordance with Equation mentioned below:

$$\mathbf{Q_{r,\theta} = r_0q_{r,\theta} + r_1q_{r,\theta-1} + r_2q_{r,\theta-2} + r_3q_{r,\theta-3} + \ldots + r_{23}q_{r,\theta-23}}$$

Where,
$Q_{r,\theta}$ =radiant cooling load Q_r for current hour θ, W
$q_{r,\theta}$ =radiant heat gain for current hour, W
$q_{r,\theta-n}$ =radiant heat gain n hours ago, W
r_0, r_1, etc. =radiant time factors

RTS for medium-weight construction, assuming 50% glass and carpeted floors as representative for the described construction. The wall heat gains for 24 h design conditions in July. Thus, the radiant cooling load for the wall at 3:00 PM is

$$
\begin{aligned}
Q_{r,15} &= r_0(0.46)q_{i,15} + r_1(0.46)\, q_{i,14} + r_2(0.46)\, q_{i,13} + r_3(0.46)\, q_{i,12} + \ldots + r_{23}(0.46)\, q_{i,16} \\
&= (0.49)(0.46)(179) + (0.17)(0.46)(138) + (0.09)(0.46)(101)
\end{aligned}
$$

$+ (0.05)(0.46)(81) + (0.03)(0.46)(66) + (0.02)(0.46)(48)$
$+ (0.02)(0.46)(29) + (0.01)(0.46)(9) + (0.01)(0.46)(–5)$
$+ (0.01)(0.46)(–9) + (0.01)(0.46)(–9) + (0.01)(0.46)(–7)$
$+ (0.01)(0.46)(–5) + (0.01)(0.46)(–3) + (0.01)(0.46)(0)$
$+ (0.01)(0.46)(4) + (0.01)(0.46)(9) + (0.01)(0.46)(16)$
$+ (0.01)(0.46)(32) + (0.01)(0.46)(78) + (0.00)(0.46)(144)$
$+ (0.00)(0.46)(192) + (0.00)(0.46)(213) + (0.00)(0.46)(207)$
$= 59$ W

The total wall cooling load at the designated hour is thus

$Q_{wall} = Q_{c15} + Q_{r15} = 97 + 59 = 156$ W

Heat Gain from Window: the windows receive solar radiation, and the heat is passed on to indoor air space by means of convection and radiation. While convective component can directly be used in calculations, the radiative component needs to be converted into convective component by multiplying the radiative component with radiation time series.

For fenestration, the solar heat gain will be calculated with following equations:

Direct beam solar heat gain $\mathbf{q_b = AE_{t,b}\, SHGC(\theta) IAC(\theta,\Omega)}$
Diffuse solar heat gain $\mathbf{q_d = A(E_{t,d} + E_{t,r})\langle SHGC\rangle_D\, IAC_D}$
Conductive heat gain $\mathbf{q_c = UA(T_{out} - T_{in})}$ **(15)**
Total fenestration heat gain $\mathbf{Q = q_b + q_d + q_c}$

Where,
A = window area, m2

E_{tb}, E_{td}, and E_{tr} = beam, sky diffuse, and ground-reflected diffuse irradiance,

SHGC (θ) = beam solar heat gain coefficient as a function of Incident angle θ

$\langle SHGC\rangle_D$ = diffuse solar heat gain coefficient (also referred to as hemispherical SHGC)

T_{in} = inside temperature, °C

T_{out} = outside temperature, °C

U = overall U-factor, including frame and mounting Orientation /(m2·K)

IAC ($\theta.\Omega$) = indoor solar attenuation coefficient for beam solar heat gain coefficient; = 1.0 if no inside shading device. IAC ($\theta.\Omega$) is a function of shade type and, depending on type, may also be a function of beam solar angle of incidence θ and shade geometry

IAC_D = indoor solar attenuation coefficient for diffuse solar heat gain coefficient; = 1.0 if not inside shading device. IAC_D is a function of shade type and, depending on type, may also be a function of shade geometry

E_{tb} = 368 W/m2
E_{td} = 175 W/m2
E_{tr} = 80 W/m2
$\theta = 58.5°$

SHGC (θ) = SHGC (58.5) = 0.3978 (interpolated)
$\langle SHGC \rangle_D = 0.41$
For light-colored blinds (assumed louver reflectance = 0.8 and louvers positioned at 45° angle) on double glazed, heat-absorbing windows, IAC (0) = 0.74, IAC (60) = 0.65, IAC (diff) = 0.79, and radiant fraction = 0.54. Without blinds, IAC = 1.0. Therefore, window heat gain components for hour 15, without blinds, are as follow:

$q_{b15} = AE_{t,b}\, SHGC(\theta)(IAC) = (7.43)(368)(0.3978)(1.00) = 1088$ W

$q_{d15} = A(E_{t,d} + E_r)\langle SHGC \rangle_D(IAC) = (7.43)(175 + 80)(0.41)(1.00)$
$= 777$ W

$q_{c15} = UA\,(t_{out} – t_{in}) = (3.18)(7.43)(33.3 – 23.9) = 222$ W
This procedure is repeated to determine these values for a 24 h heat gain profile.

For windows with inside shading (blinds, drapes, etc.), the direct beam, diffuse, and conductive heat gains may be summed and treated together in calculating cooling loads. However, in this example, the window does not have inside shading, and the direct beam solar heat gain should be treated separately from the diffuse and conductive heat gains. The direct beam heat gain, without inside shading, is treated as 100% radiant, and solar RTS factors from are used to convert the beam heat gains to cooling loads. The diffuse and conductive heat gains can be totaled and split into radiant and convective portions.

The solar beam cooling load is calculated using heat gains for the current hour and past 23 h and radiant time series. Select the solar RTS for medium-weight construction, assuming 50% glass and carpeted floors. For this example, values for direct solar heat gain, the radiant cooling load for the window direct beam solar component is

$$
\begin{aligned}
Q_{b,15} &= r_0 q_{b,15} + r_1 q_{b,14} + r_2 q_{b,13} + r_3 q_{b,12} + \ldots + r_{23} q_{b,14} \\
&= (0.54)(1091) + (0.16)(640) + (0.08)(157) + (0.04)(0) \\
&\quad + (0.03)(0) + (0.02)(0) + (0.01)(0) + (0.01)(0) + (0.01)(0) \\
&\quad + (0.01)(0) + (0.01)(0) + (0.01)(0) + (0.01)(0) + (0.01)(0) \\
&\quad + (0.01)(0) + (0.01)(0) + (0.01)(0) + (0.01)(0) + (0.01)(0) \\
&\quad + (0.00)(0) + (0.00)(298) + (0.00)(931) + (0.00)(1287) \\
&\quad + (0.00)(1343) \\
&= 704 \text{ W}
\end{aligned}
$$

This process is repeated for other hours. For diffuse and conductive heat gains, the radiant fraction is 46%. The radiant portion is processed using non solar RTS coefficients. For 3:00 PM, the diffuse and conductive cooling load is 922 W.

The total window cooling load at the designated hour is thus

$Q_{window} = Q_b + Q_{diff} + Q_{cond} = 704 + 922 = 1626$ W

Again, a computer spreadsheet or other software is commonly used to reduce the effort involved to include splitting the heat gain into convective and radiant portions, applying RTS to the radiant portion, and totaling the convective and radiant loads to determine a 24 h cooling load profile for a window without inside shading. If the window has an inside shading device, it is accounted for with the inside attenuation coefficients (IAC), the radiant fraction, and the RTS type used. If a window has no inside shading, 100% of the direct beam energy is assumed to be radiant and solar RTS factors are used. However, if an inside shading device is present, the direct beam is assumed to be interrupted by the shading device, and a portion immediately becomes cooling load by convection. Also, the energy is assumed to be radiated to all surfaces of the room, therefore non solar RTS values are used to convert the radiant load into cooling load.

IAC values depend on several factors: (1) type of shading device, (2) position of shading device relative to window, (3) reflectivity of shading device, (4) angular adjustment of shading device, as well as (5) solar position relative to the shading device.

For this example with venetian blinds, the IAC for beam radiation is treated separately from the diffuse solar gain. The direct beam IAC must be adjusted based on the profile angle of the sun. At 3:00 PM in July, the profile angle of the sun relative to the window surface is 58°.

The beam IAC = 0.653. The diffuse IAC is 0.79. Thus, the window heat gains, with light-colored blinds, at 3:00 PM are
qb15 = AED SHGC (θ)(IACb) = (7.43)(368)(0.3978)(0.653) = 710 W
qd15 = A(Ed + Er) ⟨SHGC⟩D (IACd) = (7.43)(175 + 80)(0.41)(0.79)= 614 W
qc15 = UA (tout – tin) = (3.18) (7.43) (33.3 – 23.9) = 222 W

Because the same radiant fraction and non-solar RTS are applied to all parts of the window heat gain when inside shading is present, those loads can be totaled and the cooling load calculated after splitting the radiant portion for processing with non-solar RTS. The total window cooling load with venetian blinds at 3:00 PM = 1319 W.

Heat Gain from a Window with Overhang: For 3:00 PM LST IN July, the solar position is
Solar altitude $\beta = 57.2°$
Solar azimuth $\varphi = 75.1°$
Surface-solar azimuth $\gamma = 15.1°$
$\Omega = 58.1°$
SH = PH tan Ω = 3.05(1.6087) = 4.9 m

Because the window is 1.95 m tall, at 3:00 PM the window is completely shaded by the 3 m deep overhang. Thus, the shaded window heat gain includes only diffuse solar and conduction gains. This is converted to cooling load by separating the radiant portion, applying RTS, and adding the resulting radiant cooling load to the convective portion to determine total cooling load. Total Window cooling load = 771 W

Total Heat Gain for the conference room: The steps in the previous example parts are repeated for each of the internal and external loads components, including the southeast facing window, spandrel and brick walls, the southwest facing brick wall, the roof, people, and equipment loads. The total room sensible cooling load for the conference room is 2937 W at 3:00 PM in July. When this calculation process is repeated for a 24 h design day for each month, it is found that the peak room sensible cooling load actually occurs in August at hour 15 (3:00 PM solar time) at 2968 W.

Observations

In RTS Method, the design cooling loads are based on the assumption of steady periodic conditions (i.e., the design day's weather, occupancy, and heat gain conditions are identical to those for preceding days such that the loads repeat on an identical 24 h cyclical basis). Thus, the heat gain for a particular component at a particular hour is the same as 24 h prior, which is the same as 48 h prior, etc. This assumption is the basis for the RTS derivation from the HB method.

Unlike heat balance method, RTS method, takes into consideration the delay effect of heat received by radiation and conduction. Also, the 24 hour cycle is used in calculation of delay effect, which can provide a better idea of thermal behavior of a space. Also, for heat transfer from conduction, lights, and equipment, the total heat entering into the space is divided in radiative and convective components, which makes it possible to consider the heat output from all the sources in cooling load calculation. Spreadsheet software or a computer program is necessary to perform all the calculation mentioned in RTS section due to repetitive nature of the calculations.

When windows of any building receive solar radiation, some portion of that heat is emitted inside the building by means of conduction through glass, and radiation. However, some portion of the heat received by window is emitted outside as well. Heat balance method recognizes this phenomenon in calculation; however, RTS method considers all the heat received by a window to be transmitted inside. This limitation of RTS method can give relatively higher peak loads.

Traditionally, in heat balance calculation methods, the total cooling load at a particular design condition is calculated independently and then sum of the load from each component (walls, windows, people, lights, etc.) is done. Although the actual heat transfer processes for each component do affect each other, this simplification is appropriate for design load calculations and useful to the designer in understanding the relative contribution of each component to the total cooling load. This characteristic of the calculation method puts a huge impediment in its ability to closely reflect thermal behavior of a building, which typically is of extremely dynamic nature. All in all, RTS method is relatively more inclusive as a calculation method, however, due to repetitive nature of calculations and the amount of input information required for calculation, it is more suited to be performed with the help of a computer program.

7. REFERENCE

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