# Qualitative Comparison of North American and U.K. Cooling Load Calculation Methods

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*A qualitative comparison is presented between three current North American and U.K. design cooling load calculation methods. The methods compared are the ASHRAE Heat Balance Method, the Radiant Time Series Method and the Admittance Method, used in the U.K. The methods are compared and contrasted in terms of their overall structure. In order to generate the values of the 24 hourly cooling loads, comparison was also made in terms of the processing of the input data and the solution of the equations required. Specific comparisons are made between the approximations used by the three calculation methods to model some of the principal heat transfer mechanisms. Conclusions are drawn regarding the ability of the simplified methods to correctly predict peak-cooling loads compared to the Heat Balance Method predictions. Comment is also made on the potential for developing similar approaches to cooling load calculation in the U.K. and North America in the future.*

## INTRODUCTION

Calculation of design cooling and heating loads is an essential task in the design of HVAC systems and has long been a subject of strong interest to ASHRAE and to its U.K. sister organization, CIBSE. Both societies publish methods for calculating design cooling and heating load calculations in their handbooks. However, each society has historically taken somewhat different approaches to the cooling load calculation procedure.

The increasing internationalization of the construction industry has resulted in an increasing number of North American companies working in Europe, and *vice versa*. North American and European companies are also competing for work in other parts of the world, such as the Far East. In the longer term, both the efficiency and the reputation of the HVAC industry worldwide would be improved if common methods of performing key design calculations were adopted. An essential step in the process of adopting common methods, and a worthwhile activity in its own right, is the comparison of existing methods and an understanding of the practical consequences of their differences.

Research jointly sponsored by ASHRAE and CIBSE has recently been undertaken to compare cooling load calculation procedures both quantitatively and qualitatively (Spitler et al. 1997). This paper describes work done as part of the project to qualitatively compare the procedures currently used by the societies in North America and the United Kingdom. The two North American methods compared here are the Heat Balance Method (Pedersen et al. 1997) and the Radiant Time Series Method (Spitler et al. 1997). The U.K. method is known as the Admittance

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Method (Loudon 1968, CIBSE 1986). Separate companion papers (Spitler and Rees 1998, Rees et al. 1998) describe a quantitative comparison of these methods.

The background to the development of the methods is discussed with a view to illustrating some of the common origins of the methods, and the later divergence in the development of the methods used by the two societies. The paper compares and contrasts both the structure of the three methods and the data flow in the different calculation processes. Finally, a discussion of the treatment of the principal heat gains and heat transfer mechanisms is given.

## BACKGROUND

The predecessors to ASHRAE and CIBSE, the American Society of Heating and Ventilating Engineers (ASHVE) and the Institution of Heating and Ventilating Engineers (IHVE), respectively, have sponsored research into and published work concerning the effects of transient heat transfer in buildings since the 1930s. The methods currently published by the societies are rather different in structure and in the methods they use. It is interesting to consider some of the historical developments in cooling load calculation procedures and observe some of the commonalties and differences that have arisen. Development of the methods has been strongly influenced by the development and availability of digital computing facilities. At the same time, both societies have had the strong desire to provide methods that are of utility to the average practicing engineer and there has always been a demand for methods that can be used with tabulated data (Stewart 1948).

Air conditioning was first commented on in the 1923 edition of the ASHVE *Guide*. It is apparent from the advice given in the guides of this period that the main applications of air conditioning at this time were industrial and in large public spaces (theatres and department stores) so that equipment capacities were mainly dependent on internal gains and fresh air loads. Calculation of cooling loads arising from “Sun Effects on Buildings” was not specifically discussed in the *Guide* until 1933. Concerns about transmission of solar energy into buildings had prompted a series of experimental projects in which solar fluxes; wall conductances, surface temperatures and absorptivities were measured (Houghten and Gutberlet 1930). The cooling load calculation methods that were introduced in the 1933 *Guide* allowed an engineer to calculate instantaneous gains based on surface temperature and conductance data measured by Houghten. It was also noted that “a customary rule-of-thumb is to add 25°F (14 K) to the outside dry-bulb temperature in calculating the heat transmission through walls, glass, and roof, which may be exposed to the sun for some time.”

Experimental measurements of solar fluxes and material absorptivities continued through the 1930s so that in the 1938 *Guide* tabulated solar flux data were given for the first time. These data were to be used to calculate the transfer of absorbed solar radiation through walls and roofs using the relation $H_R = AF\alpha I$, where $A$ is the area, $\alpha$ is the absorptivity of the surface, and $I$ is the incident flux. The factor $F$ was taken from a graph that correlated this reduction factor directly with the U-factor of the surface (Faust et al. 1935). Solar gains through glazing were obtained at this time by multiplying fluxes by shading coefficients.

The work on time varying heat gains through building fabrics published Mackey and Wright (1944, 1946) marked a change from these semi-empirical methods. They adopted what they noted as being the English practice of using the Sol-air temperature (called Equivalent temperature in the U.K. at the time) as the outside driving temperature. They developed a method of calculating the net flux to the inside of the wall or roof based on the theoretical consideration of a sinusoidal variation of this external temperature. Using Fourier analysis, they were able to define the response to each harmonic of the driving function by a decrement factor and an associated time lag. Using tabulated decrement factors and lags it was possible for the engineer to calculate manually the overall heat gains using only a few harmonics. The original work by

Mackey and Wright (1944) treated homogeneous walls and roofs but was later extended (Mackey and Wright 1946) to include composite constructions. Mackey and Wright's method for homogeneous constructions was included in the 1947 ASHVE *Guide*. The treatment of composite walls and slabs by this method, however, was thought at the time to be too complex for practical application.

Stewart (1948) later used the method to calculate "Equivalent Temperature Differentials" for different materials and hours of the day. In this method, the U-factor was simply multiplied by the equivalent temperature difference to calculate the conduction gain. However, various *ad hoc* corrections needed to be made for conditions that varied from those under which the tabulated data had been calculated. This method was adopted in the 1949 *Guide*. The concept of this method is appealingly (perhaps deceptively) simple and is used in the Total Equivalent Temperature Difference/Time Averaging (TETD/TA) method and the Cooling Load Temperature Difference/Solar Cooling Load/Cooling Load Factor (CLTD/SCL/CLF) method described in the 1997 *ASHRAE Handbook—Fundamentals* and the Cooling and Heating Load Calculation Manual (McQuiston and Spitler 1992).

At this point the models proposed to deal with dynamic conduction and solar gains were not integrated in any way with a model of the zone radiant and convective heat transfer. Little attention was paid to the dynamic effect of internal radiant gains until later (Mackey and Gay 1949, 1952). In the 1950s and 60s, attempts were made to model the whole building zone using electrical network analogies. These zone models had detailed internal radiant and convective models that were solved using either calculating machines or analog computers (Nottage and Parmalee 1954, Buchberg 1958). However, they were unacceptably slow and costly for practicing engineers and never found their way into the *Guide*.

Primarily for climatic reasons, the application of air conditioning to office spaces in the United Kingdom in the post-war period lagged behind that of the U.S. Changes in U.K. architectural practice in the 1950s resulted in buildings with larger amounts of glazing and stimulated the development of what became known as the Admittance Method. It was not the need for the calculation of cooling load, but with the need to calculate maximum temperatures in natural and mechanically ventilated buildings that the method was first developed. Unlike ASHRAE, whose methods were directed toward creating a constant internal temperature, so that internal mass had only a second order effect, CIBSE's primary aim was to demonstrate the role of internal mass in modifying room temperature. The development of the Admittance Method is attributed to Danter (1960) who presented a method for dealing with heat flow transmitted through the structure driven by sinusoidal external excitation. He expressed the flow conducted into the interior per unit variation in external temperature as $fU$, where $U$ is the usual steady state transmittance and $f$ is a decrement factor, dimensionless and less than unity, and having an associated time lag expressed in hours. Loudon (1968) developed the method to treat internal excitation. The method uses a very similar analytical approach to finding the response of the zone fabric to sinusoidal external excitation as that used by Mackey and Wright (1944). The mathematical technique used to find the properties of composite constructions however owes more to the matrix methods given by van Gorcum (1951) and Pipes (1957).

Another difference between the earlier U.S. calculation methods, and that developed by Danter and Loudon is that the dynamic model of the room fabric is integrated with a simplified zone convection and radiant heat transfer model. The room model is known as the environmental temperature model. Two internal nodes were defined, one of which was the air node, the other being an "environmental temperature" node, which is used to calculate the combined radiant and convective heat exchange with the room surfaces. The likely reason for this is that the U.K. methods were originally developed for calculating heating loads, and with a preponderance of hydronic radiant heating systems, a combined radiant and convective temperature was more useful than the zone air temperature. In comparison, the U.S. methods were developed for cooling, where the load was met by an air-based system. Consequently, the load at the air point was of more interest.

The concept of environmental temperature is similar to that of sol-air temperature used to define external surface heat transfer in that a combined radiant and convective conductance is used. Although the environmental temperature model was later shown to have several logical flaws (Davies 1992a, 1996a), it proved relatively popular with engineers, as it was tractable by manual calculation

In order to account for the dynamic effects of temperature changes inside the zone and internal radiant gains two further parameters besides the Decrement factor needed to be introduced. These were the fabric admittance $Y$ and the related surface factor $F$, both of which have time lead/lags associated with them. Values of typical admittance and decrement factor values published by Loudon were found using an electrical analog computer. However the later values in the 1986 *CIBSE Guide* were found by digital computation (Milbank and Harrington-Lynn 1974). The term admittance is borrowed from electrical circuit theory and has the same meaning as control point heat admittance used in the U.S. by Brisken and Reque (1956), except that they defined the internal driving temperature as the air temperature rather than the environmental temperature.

The work of Brisken and Reque (1956) in developing what they called the Thermal Response Method marked a change in the development of cooling load calculation methods in the U.S. As well as being one of the first calculation procedures to make use of a digital computer, it was the first to propose the use of response factors. These response factors were used to define the response of a particular wall or roof construction to a unit temperature pulse. The response to the diurnal excitation could be calculated by decomposing the excitation into unit pulses, and superimposing the response to each pulse at each hour. Wall and roof response factors were determined using a two-lump thermal circuit, a rectangular excitation pulse, and a Laplace transform-based solution to the two ordinary differential equations. Although the method was not adopted in the ASHRAE *Guide*, this approach—defining the response to conduction heat gains using unit pulses—was to be a common theme in the future development of ASHRAE methods.

Through the 1960s, the method recommended to U.S. engineers was based on equivalent temperature differences. (The original method was overhauled and presented as the TETD/TA method in 1967.) During this period, there was also growing interest in developing more accurate transient conduction algorithms that could be used with digital computers for both load and annual energy calculation. This work was largely pioneered in the North America by Mitalas and Stephenson. In 1967 they described a procedure for obtaining both wall and roof response factors and room thermal response factors (Mitalas and Stephenson 1967). In contrast to Brisken and Reque, a triangular pulse was used to obtain the response factors; a more detailed thermal circuit was used to generate the zone response factors; and an exact analytical solution for the transient heat conduction problem was used to generate the wall and roof response factors. A companion paper (Stephenson and Mitalas 1967) describes the use of the response factors in a cooling load calculation procedure.

Later, Stephenson and Mitalas (1971) presented a conduction transfer function (CTF) approach to modeling transient heat conduction in multi-layer slabs. Two methods for determining the transfer function coefficients were presented—one based on using an excitation function with a known Laplace-transform and z-transform. The second was based on matching the frequency response to the frequency response of the s-transfer function at several frequencies. The authors estimated that there was a five-fold reduction in the number of arithmetic operations required when using the CTF formulation when compared to using the response factor formulation. Davies (1996b, 1997) has recently shown that the full set of wall coefficients $a$, $b$, $c$, and $d$ can be evaluated using elementary time-domain solutions, without a Laplace transform.

In 1972, apparently without ever being published in a peer-reviewed archival publication, the Transfer Function Method (TFM) for computing zone thermal response was introduced in the 1972 ASHRAE *Handbook of Fundamentals*. The procedure for obtaining the room transfer function coefficients was not documented in the handbook, but a computer program was cited by Mitalas and Aresenault (1971). The method, as presented in the handbook, relied on a set of tabulated room transfer function coefficients. For each heat gain component, four heat gain coefficients ($v$) and three cooling load coefficients ($w$) were tabulated. The method was intended for use both manually and with a computer.

According to Romine (1992), the TFM method was not well received by practicing engineers due to its complexity. There was a perceived need for a method that could be used manually but that avoided the uncertain time averaging procedure of the TETD/TA method. This resulted in the development of the CLTD/SCL/CLF method (Rudoy and Duran 1975). The temperature differences and Cooling Load Factors used in this method were "backed out" of typical zone calculations made with the TFM.

The conduction transfer function ideas developed by Mitalas and Stephenson have been adopted for the calculation of transient conduction in the principal U.S. energy calculation codes. It is also within the context of the development of energy calculation codes that methods classed as heat balance methods have developed. Such methods explicitly formulate inside and outside surface and zone air heat balances and solve the resulting equations simultaneously. This approach has been successfully employed in the TARP (Walton 1983) and BLAST (1986) programs and is arguably the most fundamental of approaches in that it seeks to model the underlying physical processes most directly.

In 1996, ASHRAE funded a research project titled, Advanced Methods for Calculating Peak Cooling Loads (875-RP). The goal of this project was to replace the existing methods with two new methods: the Heat Balance Method (Pedersen et al. 1997) and the Radiant Time Series (RTS) Method (Spitler et al. 1997). The Heat Balance Method uses the same approach as the TARP and BLAST energy analysis programs. Its introduction marks a crossover of technology from energy analysis load calculation methods to design-day cooling load calculation methods. The method is also the first ASHRAE method to rely completely on computer implementation.

The Radiant Time Series Method can, in principal, be used manually and is intended to be the new ASHRAE simplified cooling calculation method, replacing the TETD/TA and CLTD/SCL/CLF methods. The method shares many heat transfer sub-models with the Heat Balance Method but has most in common with the older Transfer Function Method and can be shown in certain circumstances to be equivalent (Spitler and Fisher 1999).

## THE STRUCTURE OF THE METHODS

Considering their nodal network representation, the nature of the structure of each of the three calculation methods is compared here. A nodal network diagram, showing nodes at which temperatures are calculated and/or heat gains are added and connecting thermal resistances, is given for each of the three methods in Figures 2, 3, and 4. These diagrams are the simplest networks that will serve as an adequate example. They show a single external wall of an imaginary zone. This type of representation could not deal with some features adequately. In particular where the wall resistances are shown in the case of the Heat Balance and Radiant Time Series methods, these are more accurately described as resistive elements with distributed capacitance. This is noted in the diagrams by a modified resistor symbol illustrated in Figure 1. It should also be noted that only cooling loads that are removed by the air stream are considered, indicated in the diagrams as $Q_{Pa}$. Heat gains are defined as the rates at which heat enters or is generated within a space. Cooling loads are defined as the rates at which heat must be removed from the space to maintain a constant air temperature.

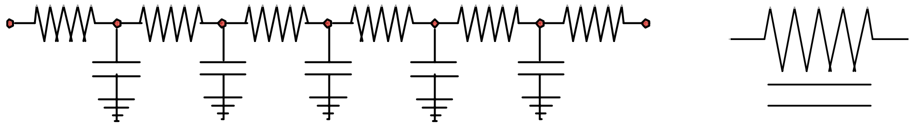

**Figure 1. Approximation of wall with thermal mass as series of lumped capacitances (left) and its representation (right) in nodal network diagrams of Heat Balance and RTS calculation methods**

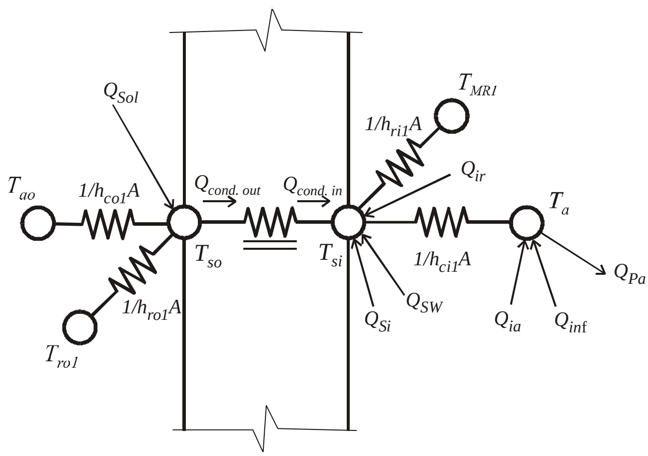


**Figure 2. ASHRAE Heat Balance cooling load calculation method represented as nodal network A single wall is shown with the outside surface on the left**

The network diagram shown in Figure 2 represents the ASHRAE Heat Balance Method (Pedersen et al. 1997), so called because it is based on a set of heat balance equations for the zone air and each of the exterior and interior surfaces. In the diagram, the nodes at which the heat balances are calculated are shown labeled $T_{si}$ and $T_{so}$ on the inner and outer surfaces and $T_a$ representing the zone air. The separate treatment of the outside and inside radiation and convection is indicated by the presence of two resistances connected to the surface nodes.

The heat balance approach can be summarized as follows. The algebraic sum of the convection $Q_{co}$, radiation $Q_{oLW}$, and absorbed solar heat gain $Q_{SOL}$ at the exterior surface must be equal to the conduction into the wall $Q_{cond.out}$. This can be expressed as a series of heat balance equations for each outside surface as follows:

$$Q_{SOL} + Q_{oLW} + Q_{co} - Q_{cond.out} = 0 \tag{1}$$

Similarly, at the interior surface the conduction out of the wall is balanced by convection to the room air $Q_{ci}$, radiant exchange with the other surfaces $Q_{iLW}$, as well as radiant fluxes from internal sources $Q_{ir}$, lights $Q_{SW}$, and the redistributed fluxes transmitted through glazing $Q_{Si}$. Expressed as a series of heat balance equations for each inside surface this gives:

$$Q_{Si} + Q_{ir} + Q_{SW} + Q_{iLW} - Q_{ci} + Q_{cond.in} = 0 \tag{2}$$

The radiant exchange with other surfaces in the zone is calculated using the MRT/Balance algorithm (Walton 1980) in which each surface is connected to a mean radiant temperature node. This accounts for the single connection to the node indicated by $T_{MR1}$ in the diagram. The temperatures $T_{so}$ and $T_{si}$ calculated at the surfaces may also be used to calculate the temperature dependent variables such as convection coefficients. Internal partitions are dealt with by coupling the inside and outside surfaces so that they have the same boundary conditions and hence temperature.

The zone air in the Heat Balance Methods, as in the other methods, is assumed to be well mixed and to have negligible capacity. Convection from the ($n$ in total) zone surfaces $Q_{ci}$, internal convective loads $Q_{ia}$ and the sensible infiltration load $Q_{inf}$ are in balance with the air load on the system $Q_{Pa}$. Thus, the heat balance on the zone air can be expressed as a single equation:

$$\sum_{1}^{n} Q_{ci} + Q_{ia} + Q_{inf} + Q_{Pa} = 0 \tag{3}$$

The set of outside heat balance Equations (1) are coupled to the inside heat balance Equations (2) by the conduction terms and the air heat balance Equation (3) is coupled to all the inside heat balance equations by the surface convection terms. This set of equations therefore requires simultaneous solution for each hour of the day. It is conventional to assume a fixed inside air temperature when making cooling load calculations. However, the Heat Balance approach is equally able to deal with variable inside temperature and system capacities.

The nodal network representation of the Radiant Time Series (Spitler et al. 1997) is shown in Figure 3. Comparison with the diagram for the Heat Balance Method shows a number of simplifications. The zone model is first simplified by the treatment of exterior and interior convection and radiation using combined surface resistances. Conduction heat gains are calculated by response factors driven by the difference between the Sol-air temperature $T_{SA}$ and room air temperature $T_a$, which is assumed constant. Hence, no surface heat balances are required.

The RTS calculation method can be thought of as a two-stage process. The first stage of this process is to calculate all the radiant and convective heat gains to the zone. The second stage is the conversion of these gains into contributions to the load on the zone air. These contributions are finally added up to arrive at the hourly loads. In this method then, no heat balances are actu-

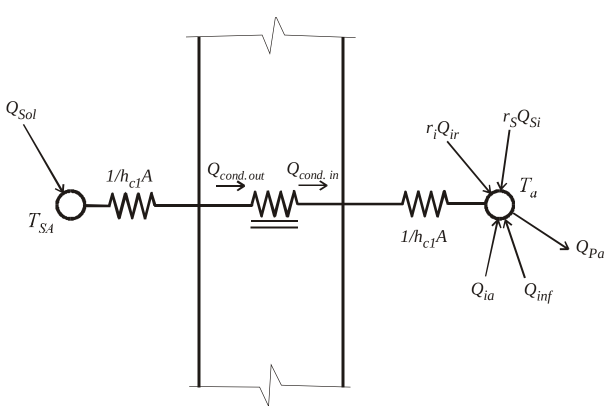


**Figure 3. Radiant Time Series cooling load calculation method represented as a nodal network. A single wall is shown with the outside surface on the left.**

ally calculated. At the room air node, the contributions to the load are simply added up. Being able to do this makes the calculation process very straightforward but requires special treatment of the radiant components of the heat gains. Radiant gains at each hour are modified by a series of twenty-four zone response factors known as the radiant time series. Thus, in the nodal diagram the radiant component of the internal loads $Q_{ir}$ and transmitted solar loads $Q_S$ are shown (somewhat unphysically) appearing at the air node but multiplied by the factors $r_i$ and $r_s$ respectively. It should be emphasized however, that the internal node is not being used as a hybrid air-and-radiant temperature node as in other methods (such as the Admittance Method).

A network representation of the CIBSE Admittance Method (Danter 1960, Louden 1968) is shown in Figure 4. The calculation of cooling loads with the Admittance Method is a two-part process in that the mean component of the heat gains and loads are treated separately from the fluctuating components. Hence, two network diagrams are necessary to illustrate the method. The diagrams are also rather different to those of the U.S. methods in that the conductances involve the properties of the whole zone, not individual surfaces.

The approach taken in the Admittance Method is in one sense similar to that of the ASHRAE Heat Balance Method in that the loads are found by solution of heat balance equations. However, the model used to represent the building zone, from which the heat balance equations are formulated, as well as the models used to treat the heat transfer processes, is rather simplified. A further significant feature of the Admittance Method is that the derivation of the method relies on the assumption that the boundary conditions (solar radiation, etc.) fluctuate sinusoidally with a period of 24 hours, much in the same way as the methods developed for ASHRAE by Mackey and Wright (1944, 1946).

Whereas the U.S. methods generally have used the zone air temperature as the reference point for the indoor condition, U.K. methods have relied on the concept of environmental temperature, which is used to calculate the combined radiant and convective heat exchange with the room surfaces. The environmental temperature lies between the room air and mean surface temperatures and is approximately $1/3T_a + 2/3T_m$, where $T_m$ is the mean surface temperature. A derivation of environmental temperature is given in Appendix A. The conductance between $T_e$ and $T_a$ is noted as $SAh_a$ where $SA$ is the sum of the areas of the room surfaces and $h_a$ (= 4.5 W/m$^2$·K) includes consideration of convective and radiative exchange.

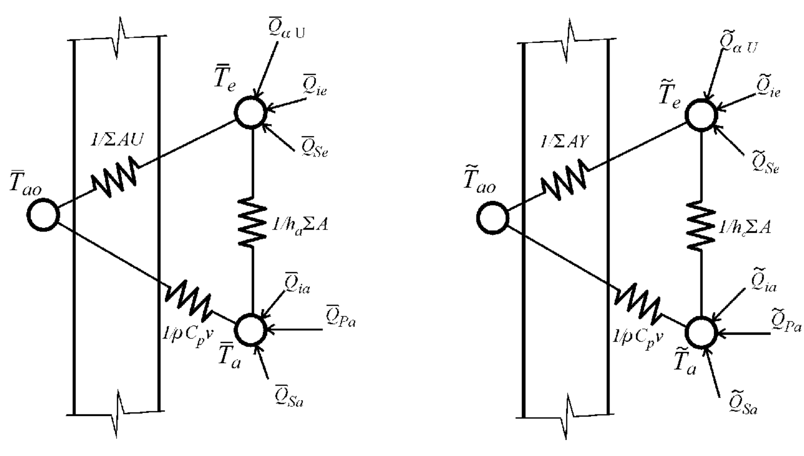


**Figure 4. CIBSE Admittance cooling load calculation method represented as a nodal network. Different networks are used for the steady state (left) and fluctuating components (right) of the heat gains and loads.**

It should be further noted that CIBSE recommended practice has been to use comfort temperature $T_c$ as the internal design temperature. This temperature is defined as being half way between the air and mean surface temperatures. It follows that $T_c = 0.25T_a + 0.75T_e$. This feature is significant when the room is heated or cooled by a panel with relatively large radiant capacity but less so when by an air system. $T_c$ does not appear in the equations or figures below though there is no difficulty in designing a cooling system to a given value of $T_c$ rather than $T_a$.

In the Admittance Method zone model, there are therefore two nodes on the inside of the zone, one being the air temperature node $T_a$ and the other, the environmental temperature node $T_e$. The steady state conductances, labeled as resistances, are shown in the left of Figure 4. $\Sigma AU$ is the sum of the external wall conductances and $rC_pv$ is the infiltration conductance corresponding to airflow from ambient into the room. In an unconditioned building $rC_pv$ is an important term, possibly larger than $\Sigma AU$.

The mean components of the heat gains to the air temperature node can be aggregated into a single term $\bar{Q}_a$ such that

$$\bar{Q}_a = \bar{Q}_{Pa} + \bar{Q}_{ia} + \bar{Q}_{Sa} \tag{4}$$

where $\bar{Q}_{Pa}$ is the mean component of the plant load, $\bar{Q}_{sa}$ the mean glazing solar gain to the air (from any blinds) and the convective internal gains are $\bar{Q}_{ia}$. The heat inputs to the environmental temperature node can be aggregated in a similar way into a single term $\bar{Q}_e$ such that,

$$\bar{Q}_e = \bar{Q}_{\alpha U} + \bar{Q}_{ie} + \bar{Q}_{Se} \tag{5}$$

where $\bar{Q}_{\alpha U}$ is the gain through the zone fabric due to absorbed solar radiation, $\bar{Q}_{ie}$ is the internal gain apportioned to the environmental temperature node and the solar load from glazing are noted as $\bar{Q}_{Se}$. By performing a heat balance at each of the nodes in the steady state nodal model, it can be shown that the net mean heat inputs at the air and environmental points can be related to the mean air temperature difference by the following equation (CIBSE 1986: Eq. A5.23),

$$\bar{Q}_a + F_{au}\bar{Q}_e = \{C_p\rho v + F_{au}\Sigma(AU)\}(\bar{T}_a - \bar{T}_{ao}) \tag{6}$$

where $F_{au}$ is a non-dimensional room factor defined by:

$$F_{au} = \frac{h_a\Sigma(A)}{h_a\Sigma(A) + \Sigma(UA)} \tag{7}$$

In Equation (6,) once a choice of $\bar{T}_a$ has been made, all the quantities except the plant load $\bar{Q}_{Pa}$ are known.

The model used to deal with the fluctuating component of the zone energy flows in the Admittance Method can be drawn as a three-node network similar to the steady state network and is shown in the right of Figure 4. Although this model can be shown as a three-node network with resistive links, there are phase differences between the energy flows and the driving temperatures along the admittance link [see Equation (8)]. The air temperature nodes are again connected through the infiltration conductance $\rho C_p v$ and the environmental temperature node by the admittance conductance $\Sigma AY$. $\tilde{Q}_{vi}$ is the cyclic energy flow due to infiltration and fluctuations in internal air temperature, and is simply $\rho C_p v\, \tilde{T}_{a\theta}$. The cyclic energy flow $\tilde{Q}_y$ is the variation of stored energy in the structure due to fluctuations in the internal environmental temperature. It can be expressed as

$$\tilde{Q}_{y\theta} = \Sigma(AY)\tilde{T}_{e(\theta + \omega)} \tag{8}$$

where $Y$ is the surface admittance, $\Sigma AY$ includes all zone surfaces, and $\omega$ is the time lead associated with the admittance.

In the case of the dynamic model the heat inputs to the air temperature node are the fluctuating component of the plant load $\tilde{Q}_{Pa}$, the glazing solar gain to the air $\tilde{Q}_{Sa}$ and the convective internal gains $\tilde{Q}_{ia}$. The heat inputs to the environmental temperature node are the fluctuating component of the glazing solar gain to the environmental node $\tilde{Q}_{Se}$, the internal gains to the environmental node $\tilde{Q}_{ie}$ and the conduction gain $\tilde{Q}_{\alpha U}$. Again these inputs can be aggregated into net inputs at each node, $\tilde{Q}_{a\theta}$ and $\tilde{Q}_{e\theta}$. Constructing a heat balance results in the following equation (CIBSE 1986: Eq. A5.40),

$$\tilde{Q}_{a\theta} + F_{ay}\tilde{Q}_{e\theta} = \{C_P\rho v + F_{ay}\Sigma(AY)\}\tilde{T}_{a\theta} \tag{9}$$

where $\theta$ indicates the hour and $F_{ay}$ is a non-dimensional room factor defined by

$$F_{ay} = \frac{h_a\Sigma(A)}{h_a\Sigma(A) + \Sigma(AY)} \tag{10}$$

After finding the components to the mean and fluctuating components of the loads it is then possible to find the cooling loads by solving the heat balance Equations (6) and (9) for each hour. Although the zone model is simple, it does allow the calculation of fluctuating internal temperatures (as in natural ventilation) and the treatment of systems that remove or add heat by radiation.

## THE CALCULATION PROCESS

The calculation methods can be further compared and contrasted in terms of the processing of the data and solution of the equations that takes place to arrive at the 24 hourly cooling loads. To this end, flow diagrams for the three methods are given below in Figures 5, 6, and 7.

Figure 5 shows a flow diagram representation of the Heat Balance calculation procedure. As with the other methods, the initial stages of the calculation process are concerned with the calculation of certain heat gains. All gains that are independent of the zone temperatures can be calculated at this stage and stored as hourly values for later use. These include solar gains through glazing, infiltration (assuming fixed internal air temperature), and internal gains which are determined from a schedule.

Consideration of the main part of the flow diagram for the Heat Balance Method immediately shows the iterative nature of the calculation process. At each hourly step the zone heat balance equations need to be assembled and solved. This is indicated by the inner loop of the diagram. (Iteration required to solve the set of simultaneous equations at a given hour is not shown). In this design, the day cooling load calculation, the boundary conditions are steady periodic. As the treatment of conduction-by-conduction transfer functions requires a history of past surface temperatures and fluxes, it is also necessary to iterate over the whole day until these temperatures and fluxes establish a steady periodic pattern.

The calculation flow diagram representation of the Radiant Time Series Method is shown in Figure 6. Again, the first stages of the calculation are concerned with the calculation of certain heat gains. This is done in exactly the same way as for the Heat Balance Method. (In the prototype implementation, the coding is common to the two ASHRAE methods.) One of the aims in

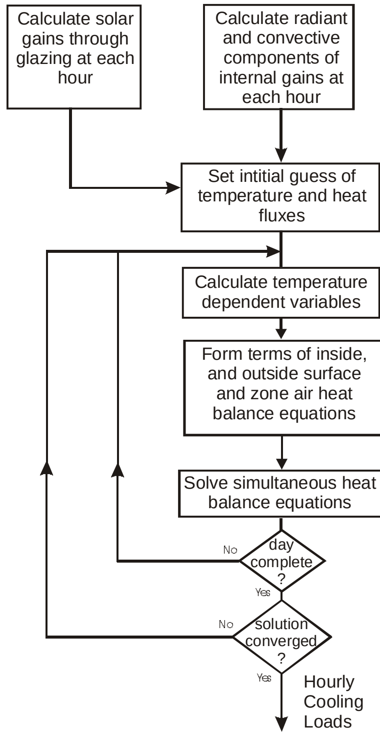


**Figure 5. ASHRAE Heat Balance cooling load calculation method represented as a flow diagram**

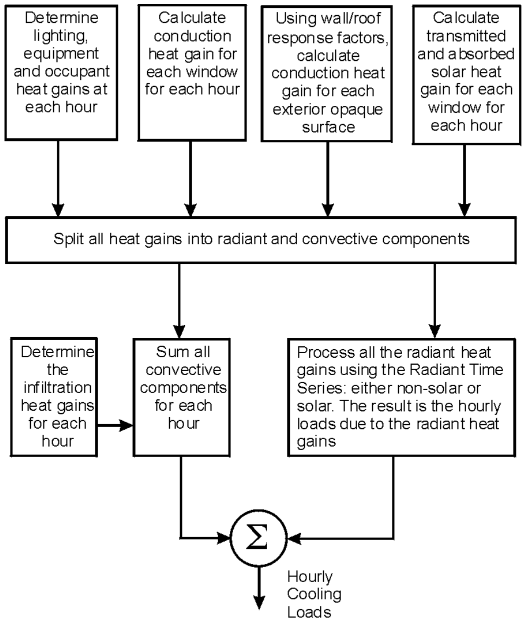


**Figure 6. Radiant Time Series cooling load calculation method represented as flow diagram**

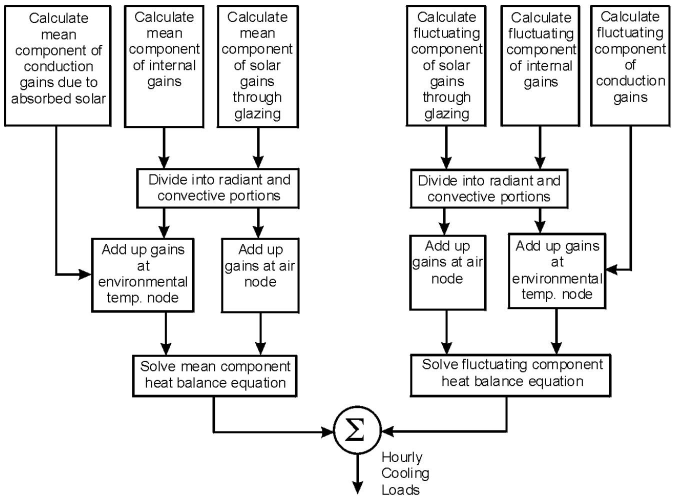


**Figure 7. CIBSE Admittance cooling load calculation method represented as a flow diagram**

the development of the RTS Method has been to produce a method that is suitable for spreadsheet implementation. For this reason, iteration loops have been specifically avoided. This is possible in the first place by the use of response factors to calculate heat conduction. These act only upon the sol-air and internal dry bulb temperatures—the hourly values of which are known at the start of the calculation—and the need for past surface flux values is avoided.

Once individual gains have been calculated, the next important step is to divide all the gains into their radiant and convective components. This is done using fixed ratios for each type of gain [see Spitler et al. (1997) for details].

The second stage of the RTS calculation procedure is to convert all the heat gains into contributions to the load at the air node. Convective components make an instantaneous contribution to the cooling load. The radiant components of the gains are treated rather differently. Each hourly radiant gain is modified by the application of the radiant time series coefficients. The radiant time series is dependent on the overall thermal storage properties of the zone and defines how the radiant gain at a given hour is redistributed in time to become contributions to the cooling load at future hours. Once the radiant gains have been processed in this way, they are simply added to the hourly convective gains to give the hourly cooling loads.

The Admittance Method flow diagram is shown in Figure 7. The separate treatment of the mean and fluctuating components of the heat gains and cooling loads is indicated by the symmetrical arrangement of the diagram. As with the other two methods, the internal and solar gains are calculated for each hour at the start of the calculation process. Both the mean and fluctuating components of these gains need to be divided into convective and radiant portions. With reference to the nodal diagram (Figure 4) this means assigning gains to either the internal air or environmental temperature nodes. Heat gains added to the environmental temperature node have an implicit 2/3 radiant, 1/3 convective split (see Appendix A). In order to add a purely radiant load it is therefore necessary to increase the gain to the environmental node by 50% and then subtract 50% of that gain from the air node.

The flow diagram also shows that conduction gains are always added directly to the environmental temperature node. Although the Admittance Method uses heat balance equations to find the load components, the main component of the conduction gains can be pre-calculated for each hour. This is because these gains are calculated based on an analytical solution to sinusoidally excited conduction heat flow.

After the different components of the gains have been added to the environmental temperature and air nodes, the heat balance equations [Equations (4) and (6)] can be solved at each hour to find the mean $\bar{Q}_{Pa}$ and fluctuating components $\tilde{Q}_{Pa}$ of the cooling loads. This can be done as a single step operation as no iteration is required to find unknown temperatures. These components of the load are finally added to give the 24-hour cooling loads.

## HEAT TRANSFER PHENOMENA

This section compares specific practices used by the three calculation methods to model some of the principal zone heat transfer mechanisms. Some observations are made regarding the ability of the simplified methods to correctly predict peak-cooling loads compared to the Heat Balance Method predictions.

### Exterior Convection and Radiation

The Heat Balance Method treats exterior convection and radiation heat transfer separately. McClellan and Pedersen (1997) review a number of possible models for exterior convection and radiation heat transfer that could be applied in a heat balance based load calculation procedure. The model of exterior convection heat transfer used in the Heat Balance Method has a convection coefficient that is correlated to wind speed and surface-to-air temperature difference (Yazdanian and Klems 1994). The convection resistance is thus a non-linear function of the node temperatures.

Exterior radiation heat transfer in the Heat Balance Method is modeled with a first-principles radiation analysis—surface emissivity, surface temperature, view factor to sky, sky temperature, view factor to ground, and a ground temperature are all required inputs. While it is possible to use a detailed sky model and/or a detailed ground surface temperature model with the method, it is simply assumed that the sky temperature is 6 K lower than the air temperature and that the ground surface temperature is the same as the outdoor air temperature.

Both the RTS Method and the Admittance Method use a fixed exterior surface conductance combined with a sol-air temperature to model exterior convection and radiation.

### Transient Conduction Heat Transfer

Transient conduction heat transfer through the zone fabric is driven by both external driving forces (e.g., solar radiation, air temperature) and internal driving forces (e.g., lighting, equipment, solar radiation transmitted through a window onto a surface). All of the methods discussed here make the conventional assumption that the heat flow is one-dimensional and that the surfaces are isothermal.

The ASHRAE Heat Balance Method models transient conduction due to both internal and external excitation simultaneously, with conduction transfer functions. Conduction transfer functions relate the current surface heat flux to the current and past values of surface temperature on both sides of the wall and the past values of surface flux. This is expressed as:

$$q_{i\theta} = -\sum_{n=0} c_n T_{i\theta-n\delta} + \sum_{n=0} b_n T_{o,\theta-n\delta} - \sum_{n=1} d_n q_{i\theta-n\delta} \qquad (11)$$

for the inside surface and

$$q_{o\theta} = -\sum_{n=0} b_n T_{i\theta-n\delta} + \sum_{n=0} a_n T_{o,\theta-n\delta} - \sum_{n=1} d_n q_{o\theta-n\delta} \tag{12}$$

for the outside surface. Each summation has as many terms as there are nonzero values of the coefficients, depending on the construction of the wall or roof. Although this allows a very generalized formulation, neither the surface temperatures or past heat fluxes can be known in advance and so a simultaneous solution of the equations must be sought. The CTF coefficients ($a,b,c,d$) are determined for any combination of construction layers using the method described by Hittle and Bishop (1983). The interior and exterior surface conductances are not included in the coefficients.

The RTS Method treats external and internal excitation of conduction heat flow separately. In the RTS procedure, transient conduction heat transfer due to external excitation is modeled using a set of 24 periodic response factors. Given the constant zone air temperature $T_a$ and the current and 23 past values of sol-air temperature $T_{SA\theta}$, the current hour's conduction heat gain per unit surface area is given by:

$$q_\theta = \sum_{j=0}^{23} Y_{Pj}(T_{SA,\,\theta-j\delta} - T_a). \tag{13}$$

The periodic response factors $Y_{Pj}$ include both the interior and exterior surface conductances and may be determined from the generalized form of the CTFs with a procedure described by Spitler and Fisher (1999). The sol-air and inside temperatures are known at the beginning of the calculation, therefore the heat gains due to conduction can be calculated straightforwardly without the need for any iteration. These gains subsequently have to be divided into radiant and convective components. Conduction due to internal excitation is treated using the radiant time series coefficients and is discussed below.

In the Admittance Method, transient conduction heat transfer through the wall is modeled in a rather different way to either of the two ASHRAE methods. The conduction of the mean component is treated using the U-factor and the mean air temperature as the external temperature (as in the nodal model on the left of Figure 4). The additional mean component due to absorption of solar radiation $\bar{Q}_{\alpha U}$ is added separately as an input to the environmental temperature node.

The fluctuating components of the energy flows into and out of the zone fabric are dealt with in three ways. The primary component of transient conduction is that due to external excitation by variations in sol-air temperature. Other components are due to internal excitation by either variations in internal environmental temperature or radiant heat flux at internal surfaces.

The response to the fluctuating components of these excitations is determined by the decrement factor $f$ (non-dimensional), admittance $Y$, and the surface factor $F$ (non-dimensional), respectively. Each of these has a time lead/lag associated with it. The values of these quantities are derived from the thermophysical properties of the fabric layers using a frequency domain solution to the unsteady conduction heat transfer equation assuming that the fluctuating temperatures and heat fluxes can be defined by sinusoidal functions with a period of 24 hours. The derivation of these properties is given in Milbank and Harrington-Lynn (1974) and the draft revision of Sections A5, 8, and 9 of the CIBSE Guide (Holmes and Wilson 1996).

These properties can be defined as follows:

- **Decrement Factor ($f$).** The ratio of the cyclic conductance though the fabric to the steady state U-factor. It defines the degree to which the fabric attenuates a cyclic variation in heat flux at the outside as it is conducted to the inside for a fixed internal environmental temperature. Decrement factor decreases with increasing thermal mass of the fabric construction.

- **Admittance (*Y*).** The rate of heat flow between the inside surface of the fabric and the environmental temperature point for each degree of swing in environmental temperature. It has the same units as U-factor and its value is mainly dependent on the thermal properties of the inside layers of the fabric construction. For thin constructions, its value approaches the U-factor, and it approaches a constant value for constructions more than 200 mm (8 in.) thick.
- **Surface Factor (*F*).** The ratio of the fluctuating component of a heat input at an internal surface to the fluctuating component that is readmitted to the environmental temperature point. Such a flow of heat does not correlate to a single heat transfer mechanism but use of the Surface Factor allows the effect of a radiant flux on the heat balance at the environmental temperature node to be calculated without regard for other heat flows in the room. It is used to determine how much fluctuating radiant heat inputs at surfaces (e.g., transmitted solar energy) is admitted to the room and how much of that energy is stored in the fabric. Like admittance, its value mostly depends on the thermal properties of the inside layers of the fabric construction and it approaches a constant value for constructions more than 200 mm (8 in.) thick.

The fluctuating component of the conduction due to external excitation (variations in sol-air temperature) is treated by using a conductance that is the U-factor multiplied by the decrement factor so that $\tilde{Q}_{\alpha U\theta} = \sum(AfU)\tilde{T}_{SA(\theta-\phi)}$ where the summation is for all external surfaces and $\phi$ is the time lag associated with the decrement factor. It is this element of the method that is most similar to that of Mackey and Wright (1946).

The transient conduction that is due to variations in internal environmental temperature is determined using the zone admittance *Y*. This component of the load is effectively added to the environmental temperature node (shifted in time by $\omega$, the time lead associated with the admittance) and is calculated by the application of the energy balance Equation (8) at the environmental temperature node. (Note that even when the air temperature is held constant there might be small fluctuations in the internal environmental temperature.)

Radiant fluxes arriving at internal surfaces—typically from equipment and solar gains—may be stored, conducted out of the zone, or reradiated. Such fluxes can be dealt with by multiplying the surface factor *F* by the fluctuating component of internal radiant gains, and applying an associated time lag in order to find the heat input to the environmental temperature node.

## Interior Convection and Radiation

The Heat Balance Method, in contrast to both simplified methods, treats both interior convection and radiation separately. Interior convection is determined with a set of constant coefficients applied to the difference between the surface temperature and the zone air temperature. The coefficients used depend on the surface orientation, and for horizontal surfaces, the direction of heat flow. The coefficients assume natural convection, and are based on experimental work. One might note that the method does not require fixed coefficients, but could use coefficients that are a function of temperature difference or air movement.

The Heat Balance Method estimates interior surface-to-surface radiation heat transfer with Walton's (1980) MRT/Balance procedure. This method creates a fictional mean radiant temperature (MRT) for each surface that is a weighted average of all of the other surface temperatures in the room. The weighting is based on the surface areas and emissivities. Because the calculation made using the fictional MRTs results in a radiation imbalance, redistributing the imbalance to each surface such that energy is conserved makes the correction. Although the method generally works well (Stefanizzi et al. 1990a, 1990b) it cannot be represented adequately as a network. Other methods, such as Davies' (1988, 1992b) radiant star network method could be used interchangeably in the heat balance method.

The RTS Method and the Admittance Method both combine the interior radiation and convection heat transfer. However, they are fundamentally different in how they apply the combination. The RTS Method, when calculating the conduction gains, uses fixed surface conductances equivalent to the convection coefficients used by the Heat Balance Method, but with a radiation conductance added. The combined convection and radiation coefficient is added (as a resistance) into the wall, but the combination has the effect of having the wall radiating to the zone air temperature. In most cases, this results in a slight over-prediction of the cooling load.

When dealing with both the redistribution of the radiant part of the conducted gains, as well as all other radiant gain components, the RTS Method uses radiant time factors—the coefficients of the radiant time series. Like response factors, radiant time factors are used to determine the cooling load for the current hour based on current and past heat gains. The radiant time series for a particular zone gives the time-dependent response of the zone to a single steady periodic pulse of radiant energy incident at the zone internal surfaces. The series shows the portion of the radiant pulse that is convected from the zone surfaces to the zone air at each hour. The radiant time factors are defined such that $r_0$ represents the proportion of the radiant pulse convected to the zone air in the current hour and $r_1$ in the previous hour, and so on. Thus, the cooling load $Q_\theta$ due to radiant gains in the current hour and past hours ($q_{\theta-n\delta}$) is given by:

$$Q_\theta = r_0 q_\theta + r_1 q_{\theta-\delta} + r_2 q_{\theta-2\delta} + r_3 q_{\theta-3\delta} + \dots + r_{23} q_{\theta-23\delta} \tag{14}$$

The radiant time series is generated by driving a heat balance model of the zone with a periodic unit pulse of radiant energy under adiabatic wall conditions. The radiant factors are therefore different for every combination of zone construction and geometry. In principle, they are also different for every chosen distribution of radiation. In practice, however, for a given zone, only two series are necessary. One is found assuming an equal distribution of radiation on all zone surfaces and is used for all radiant gains except direct solar gains. A second set is found with the unit pulse of radiant energy added at the floor surface and is used to treat the direct (beam) solar gains. An example of both radiant time series for a typical lightweight and heavyweight zone is illustrated in Figure 8.

The treatment of radiant gains by the use of radiant time factors, where there is no requirement for knowledge of past temperatures or cooling loads, allows a sequential calculation pro-

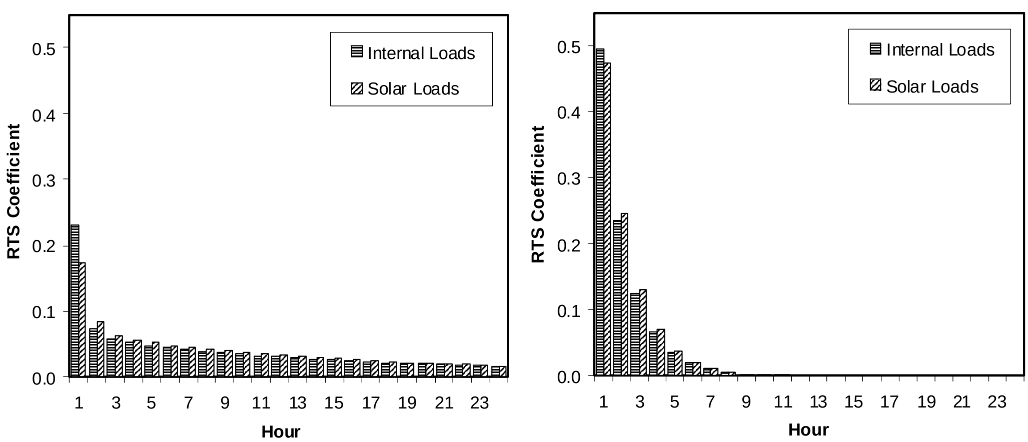


**Figure 8. RTS coefficients for typical heavyweight (left) and lightweight (right) zone**

cess, i.e., iteration is again avoided. One assumption made in calculating the radiant time factors does, however, have important implications. It was noted earlier that these factors are found from a heat balance model of the zone with adiabatic wall conditions. This means that the external surfaces are treated as internal surfaces. The radiant pulse used to calculate the radiant time factors is then only redistributed in time, but its energy is entirely conserved. The assumption is that no radiant gains are conducted out of the zone. In making cooling load calculations, this is generally a conservative assumption and leads to slight over-prediction of the peak-cooling load. However, in zones with high conductance walls and windows, a significant portion of the radiant heat gains can be conducted out of the zone, never to become part of the cooling load. In these cases a much larger over-prediction relative to the Heat Balance Method prediction, can be expected. Another paper (Rees et al. 1998) quantifies this effect for some specific zones and gives further explanation.

The Admittance Method also uses a combined radiation and convection coefficient, but it has the walls convecting and radiating to the environmental temperature. This simplified model makes for a very simple calculation procedure, but involves a number of assumptions (see Appendix A) and, arguably, a number of logical flaws (Davies 1992a, 1996a).

## Transmission and Distribution of Solar Radiation

Transmission of solar radiation through fenestration, and its distribution in the zone, is a very important part of the load calculation for most zones in modern buildings. The response of the zone is dependent not only on the value of the transmitted and absorbed solar energy but also on its distribution in the zone and its division between radiant and convective components.

In both the Heat Balance procedure and the RTS procedure, the transmission of solar radiation is estimated with the fairly simple model in Chapter 29 of the 1997 *ASHRAE Handbook—Fundamentals*. The angle dependent properties for a single sheet of standard reference glass are used in conjunction with a constant solar heat gain coefficient (SHGC). The SHGC is defined so that simply multiplying the coefficient by the incident irradiance gives the total solar gain per unit area of window [ASHRAE 1997: Ch. 29 Eq. (23)]. The solar heat gain coefficient includes both the transmitted portion of the solar energy as well as the absorbed and re-emitted portion. This therefore precludes the separate treatment of the absorbed and re-emitted solar energy, which becomes quasi-instantaneous convective gains. For a review of more sophisticated models that might be used in conjunction with the heat balance procedure, see Chorpening (1995).

The heat balance procedure assumes that all of the direct (beam) solar radiation is incident on the floor. (This assumption is not a requirement of the method, and a more sophisticated model might be used.) Likewise, the RTS procedure makes a similar assumption, using the solar radiant time series to convert the direct solar gains to cooling loads. The diffuse solar gains are treated in a similar way to other internal radiant gains.

As noted previously in the discussion on internal convection and radiation heat transfer, some of the solar radiation that is re-radiated can be conducted back out of the zone. The RTS procedure cannot account for this, and so for some zones over-predicts the cooling loads.

The Admittance Method, as set out in the CIBSE Guide (1986), suggests that solar gains through glazing be dealt with in two ways, depending on whether the cooling load or overheating is being calculated:

1. If an overheating calculation is required, the total incident radiation is first divided into its mean and fluctuating components. The mean and fluctuating components are then multiplied by a *Solar Gain Factor* and an *Alternating Solar Gain Factor*. These factors are constant and are defined for energy transfer to both the air and environmental temperature points. The solar gain is then obtained by multiplying the glazing area by the incident irradiation by the appropriate solar gain factor. The alternating component is shifted in time by a lag associated

with the alternating solar gain factor. These components are then added at the environmental temperature point. (If there is an internal blind there may also be a component to be added at the air node). These solar gain factors are tabulated in the guide for various window/blind types in heavy and lightweight buildings located in London.

2. If a peak-cooling load is required, then tabulated loads due to solar gains in typical heavyweight or lightweight zones are given in the Section A9 of the CIBSE Guide. These tabulated loads have been calculated using a detailed glazing model and what is otherwise the admittance model. In this model, due account is taken of the variation of transmittance with incidence angle and re-emittance of absorbed radiation for various latitudes and window/shading combinations. Holmes and Wilson (1996) describe the exact calculation method.

In either case, the solar gains are assumed to be evenly distributed over all the internal surfaces. Both of these approaches have historically been developed with manual calculation in mind, but by relying on tabulated values, lack generality. The solar gain factor approach is a very simplified model of the transmission of solar irradiation through glazing. Of the two recommended methods, the Solar Gain Factor approach is more practical for computer implementation. There is no reason, in principle, why a more detailed model could not be used in a computer implementation of the method (as in the calculations described by Holmes and Wilson, 1996).

## Internal Heat Gains

In the ASHRAE Heat Balance Method, hourly schedules for all internal heat gains (e.g. people, lights, and equipment) are specified by the user, together with the respective radiative/convective fractions. The radiative/convective fractions are assumed to be fixed. While this assumption is somewhat artificial, as the split between radiative and convection actually depends on the zone air temperature and surface temperatures, it has been commonly used, even in detailed building energy analysis programs. The convective portion of the heat gain is assumed to contribute instantaneously to the cooling load and appears directly in the air heat balance. The radiative portion is distributed uniformly over all the interior surfaces of the zone, and appears as a surface heat flux in the interior surface heat balance.

The RTS Method uses the same approximations as the heat balance procedure, but effect of the radiative portion of the heat gain on the cooling load is estimated with the radiant time series, rather than the surface heat balances. As discussed above, the RTS procedure has no way to account for the portion of the radiant heat gain that is conducted out of the zone and so for some zones over-predicts the cooling loads.

The Admittance Method allows, in principle, the convective and radiant portions of the internal gains to be treated separately by adding appropriate proportions at the air and environmental temperature nodes. In practice, internal gains are often introduced directly at the environmental temperature node. Loads introduced at the environmental temperature node in this way have an implicit radiant/convective split of 2/3 radiant, 1/3 convective. This split is assumed to be characteristic of most internal loads. The result of adding some of the internal gains at the environmental temperature node is to reduce the peak load at the air node and increase the mean load, i.e. some of the load becomes evenly distributed in time. This is a simplification of the effect of radiant internal gains, which have the effect of heat being stored in the fabric when first applied and gradual release at a later time, and can lead to the under-prediction of the peak load. Addition of all the internal gains at the air point is a conservative approach and will always result in over-prediction of peak load. However, for most realistic loads (with a significant radiant component) this will result in a significant over-prediction of the peak load.

## CONCLUSIONS AND FUTURE DEVELOPMENTS

Consideration of the historic development of cooling load calculation procedures in North America and the U.K. has showed that the greatest commonality in the theoretical basis of the methods occurred during the 1960s. It was at this time that methods based on the work of Mackey and Wright were being used in the ASHRAE community and the CIBSE Admittance Method was being developed. CIBSE has retained this method as its primary recommendation for dynamic load analysis, while equivalent temperature difference methods and response factor methods have been developed by ASHRAE. The structure of the simplified methods used by the two societies has therefore become rather different.

The structure of the three methods can be summarized as follows.

- The Heat Balance Method takes an approach that is least abstracted from the physical zone heat transfer processes. It does this by modeling the interior and exterior surface and zone air heat balances explicitly. This requires the simultaneous solution of the heat balance equations at each hour. The method is therefore the most fundamental and general of those examined but requires computer implementation.
- The Radiant Time Series Method uses a two-step calculation procedure, which assumes that all heat gains must eventually become cooling loads. The calculation procedure is sequential in nature and given the applicable response and radiant factors could be calculated manually.
- The Admittance Method uses a two-step calculation procedure in which the steady-state and fluctuating components of the load are calculated in turn. The steady-state component is calculated using a three-node model incorporating an environmental temperature node to which all zone surfaces are connected by a combined radiant and convective conductance. The fluctuating component of the load is calculated using an adaptation of the steady-state model, in which the zone thermal properties characterizing the response to fluctuating loads are derived from an analytical model in which the gains are assumed to vary sinusoidally with a 24-hour period.

Consideration of the treatment of the main heat transfer phenomena by the two simplified methods has shown a number of additional reasons why predicted loads may differ from those given by the Heat Balance Method.

- The RTS Method’s treatment of interior radiation and convection heat transfer in a combined manner should result in an over-prediction of cooling loads, compared to the Heat Balance Method
- Heat gains that are conducted back out of the zone cannot be accounted for in the RTS Method. This may result in significant over-prediction of peak cooling loads, particularly where there are high gains and a zone construction of low thermal resistance.
- The environmental temperature model used in the Admittance Method is very simplified and involves a number of assumptions. However, in contrast to the RTS Method, the Admittance Method does take into consideration losses through the fabric. The method can be expected to predict a different response compared to the Heat Balance Method in cases where the dominant gains differ from a sinusoidal pattern. It does not always give conservative results relative to the Heat Balance Method.
- Solar gains in the Admittance Method are treated by one of two methods, both of which rely on tabulated data, and have been historically developed for manual application. These methods are very simplified by current standards and cannot be expected to give accurate results except in a limited range of circumstances. In a computer implementation of the Admittance Method, a more sophisticated solar gain model could be used.

The turn of the millennium sees both ASHRAE and CIBSE working on the revision of their recommended cooling load calculation procedures. In addition, the Comité Européen de Normalisation (CEN), the standards-making organization that includes all the major countries of Western Europe, including the U.K., is in the process of developing a standard approach to load calculations. The draft CEN standard (CEN 1997) takes the form of a specification consisting of a set of heat balance equations and a set of qualification tests against which particular computer codes can be evaluated. It is also likely that this draft standard may also be proposed as a new ISO standard for cooling load calculations.

In the draft revision to the sections of the CIBSE Guide that relates to load calculation procedures (Holmes and Wilson 1996), a number of models of differing complexities are proposed. Two dynamic methods are proposed, one based on a detailed reference model and another based on a simplified model (which is, in fact, the Admittance Method). The reference model consists of a performance specification, along with a list of features that must be included. Particular model equations or calculation methods are not specified.

It appears, then, that in the near term, the simplified cooling load calculation methods recommended by CIBSE (and CEN) would be rather different to those recommended by ASHRAE. However, consideration of the specifications for the detailed models to be recommended by these institutions shows that the requirements could probably only be met by methods based on explicit heat balances. In the longer term, therefore, there could be considerable convergence in the detailed cooling load calculation methods used in North America and the United Kingdom.

## ACKNOWLEDGEMENTS

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## NOMENCLATURE

| | |
|---|---|
| $a,b,c$ | conduction transfer coefficients multiplying temperatures, W/(m$^2$·K) (Btu/h·ft$^2$·°F) |
| $d$ | conduction transfer coefficient multiplying fluxes |
| $A$ | surface area, m$^2$ (ft$^2$) |
| $C$ | conductance, W/K (Btu/h·°F) |
| $C_p$ | specific heat capacity (kJ/(kg·K) (Btu/lb·°F) |
| $f$ | decrement factor |
| $F$ | surface factor |
| $F_{au}$ | nondimensional room factor |
| $F_{ay}$ | nondimensional room factor |
| $h$ | convective heat transfer coefficient, W/(m$^2$·K) (Btu/h·ft$^2$·°F) |
| $K$ | conductance, W/K (Btu/h·°F) |
| $q$ | heat flux, W/m$^2$ (Btu/h·ft$^2$) |
| $Q$ | heat flux or gain, W (Btu/h) |
| $Q$ | mean component of heat flow, W (Btu/h) |
| $Q$ | fluctuating component of heat flow, W (Btu/h) |
| $r$ | radiant time factor |
| $R$ | surface resistance, K·m$^2$/W (h·ft$^2$·°F/Btu) |
| $S$ | conductance, W/K (Btu/h·°F) |
| $T$ | temperature, °C (°F) |
| $U$ | overall conductivity or transmittance, W/(m$^2$·K) (Btu/h·ft$^2$·°F) |
| $v$ | ventilation flow rate, m$^3$/s (cfm) |
| $Y$ | response factor or admittance, W/(m$^2$·K) (Btu/h·ft$^2$·°F) |
| $\varepsilon$ | emissivity |
| $\phi$ | time lag associated with decrement factor, h |
| $\rho$ | density, kg/m$^3$ (lb/ft$^3$) |
| $\omega$ | time lead associated with admittance, h |

**Subscripts**

| | |
|---|---|
| $a$ | air |
| $c$ | convective |
| $e$ | environmental index |
| $i$ | inside |
| $m$ | mean |
| $MR$ | mean radiant |
| $o$ | outside |
| $r$ | radiant |
| $s$ | surface |
| $SA$ | sol-air |
| $z$ | zone |
| $\theta$ | time index |

## REFERENCES


ASHRAE 1997. *ASHRAE Handbook—Fundamentals*. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Brisken, W.R., and S. G. Reque, 1958. Heat Load Calculations by Thermal Response. *Transactions American Society of Heating & Air-Conditioning Engineers* 62:391-424.

BLAST Support Office. 1986. *BLAST (Building Loads and System Thermodynamics)*. Urbana-Champaign: University of Illinois.

CEN. 1997. Thermal Performance of Buildings: Room Sensible Cooling Loads Calculation—General Criteria and Validation Procedures: Appendix A, CEN/TC/89/WG6 N.225 pp. 52.

CIBSE. 1986. *Guide Book A –Design Data*. London: Chartered Institution of Building Services Engineers.

Danter, E. 1960. Periodic Heat Flow Characteristics of Simple Walls and Roof. *J. Inst. Heat. Vent. Engrs*. 28: 136-146.

Davies, M.G. 1988. Design Models to Handle Radiative and Convective Exchange in a Room. *ASHRAE Transactions* 94(2): 173-195.

Davies, M.G. 1992a. Flaws in the Environmental Temperature Model. *Building Services Engineering Research and Technology* 13(4): 209-215.

Davies, M.G. 1992b. The Basis for a Room Global Temperature. *Phil. Trans. Royal Society* 339: 153-191.

Davies, M.G. 1996a. Comfort Temperature: Flawed Status in the CIBSE Guide. *Building Services Engineering Research and Technology* 17(3): 161-165.

Davies, M.G. 1996b. A Time-Domain Estimation of Wall Conduction Transfer Function Coefficients. *ASHRAE Transactions* 102(1): 328-343.

Davies, M.G. 1997. Wall Transient Heat Flow Using Time-Domain Analysis. *Building and Environment* 32(5): 427-446.

Faust, F.H., L. Levine, and F.O. Urban. 1935. A Rational Heat Gain Method for the Determination of Air Conditioning Cooling Loads. *Transactions American Society of Heating & Ventilating Engineers* 41: 327.

Gorcum, A.H. van. 1951. Theoretical Considerations on the Conduction of Fluctuating Heat Flow. *Applied Scientific Research* A2: 272-280. The Hague: Martinus Nijhoff.

Hittle, D.C., and R. Bishop. 1983. An Improved Root-Finding Procedure for use in Calculating Transient Heat Flow through Multi-Layered Slabs. *International Journal of Heat and Mass Transfer*. 26(11): 1686-1693.

Holmes, M.J., and A. Wilson. 1996. Thermal Response and Plant Sizing (5th draft of Section 5, Environmental Design, of Book A of the CIBSE Guide). London: Ove Arup & Partners.

Houghten, F.C., and C. Gutberlet. 1930. Absorption of Solar Radiation in its Relation to the Temperature, Color, Angle and other Characteristics of the Absorbing Surface. *Transactions American Society of Heating & Ventilating Engineers* 36:137-152.

Loudon, A.G. 1968. Summertime Temperatures in Buildings Without Air Conditioning. *Building Research Station Current Paper* 47/68. (also *Journal of Heating and Ventilating Engineers* 32: 280-292, 1970)

Mackey, C.O., and, L.T. Wright. 1943. Periodic Heat Flow—Homogeneous Walls or Roofs. *Transactions American Society of Heating & Ventilating Engineers* 50: 293-312.

Mackey, C.O., and, L.T. Wright. 1946. Periodic Heat Flow—Composite Walls or Roofs, *Heating, Piping and Air Conditioning* 18(6): 107-10.

McClellan, T.M., and C.O. Pedersen. 1997. Investigation of Outside Heat Balance Models for Use in a Heat Balance Cooling Load Calculation Procedure. *ASHRAE Transactions* 103(2).

McQuiston, F.C., and J.D. Spitler. 1992. *Cooling and Heating Load Calculation Manual*, 2nd Edition. Atlanta: ASHRAE Inc.

Milbank, N.O., and J. Harrington-Lynn. 1974. The Admittance Procedure: Variable Ventilation. *Building Services Engineering* (42): 38-51.

Mitalas, G.P., and J.G. Arsenault. 1971. Fortran IV Program to Calculate Z-Transfer Functions for the Calculation of Transient Heat Transfer through Walls and Roofs. Proceedings of "Use of Computers for Environmental Engineering Related to Buildings." *NBS Building Science Series* 39. Gaithersburg, MD.

Mitalas, G.P., and D.G. Stephenson. 1967. Room Thermal Response Factors. *ASHRAE Transactions* 73(1): 2.1-2.10.

Nottage, H.B., and, G.V. Parmelee. 1954. Circuit Analysis Applied to Load Estimating. *Transactions American Society of Heating & Ventilating Engineers* 60: 59-102.

Pedersen, C.O., D.E. Fisher, and R.J. Liesen. 1997. Development of a Heat Balance Procedure for Calculating Cooling Loads. *ASHRAE Transactions* 103(2): 459-468.

Pipes, L.A. 1957. Matrix Analysis of Heat Transfer Problems. *Journal of the Franklin Institute* 625: 195-206.

Rees, S.J., J.D.Spitler, and P. Haves. 1998. Quantitative Comparison of North American and U.K. Cooling Load Calculation Procedures—Results. *ASHRAE Transactions* 104(2): 47-61.

Romine, T.B. 1992. Cooling Load Calculation: Art or Science? *ASHRAE Journal* 34(1): 14-24.

Rudoy, W., and Duran, F. 1975. Development of an Improved Cooling Load Calculation Method. *ASHRAE Transactions* 81(2): 19-69.

Spitler, J.D., and S.J. Rees. 1998. Quantitative Comparison of North American and U.K. Cooling Load Calculation Procedures—Methodology. *ASHRAE Transactions* 104(2): 36-46.

Spitler, J.D., S.J. Rees, P. Haves, M.G. Davies, and M.J. Holmes. 1998. Final Report: Comparison of Load Calculation Procedures ASHRAE/CIBSE 942-RP. Atlanta: ASHRAE.

Spitler, J.D., D.E. Fisher, and C.O. Pedersen. 1997. The Radiant Time Series Cooling Load Calculation Procedure. *ASHRAE Transactions* 103(2): 503-515.

Spitler, J.D., and D.E. Fisher. 1999. On The Relationship between the Radiant Time Series and Transfer Function Methods for Design Cooling Load Calculations. *International Journal of Heating, Ventilating, Air-Conditioning and Refrigerating Research* 5(2): 125-138.

Stefanizzi, P., A. Wilson, and A. Pinney. 1990. Internal Long-Wave Radiation Exchange in Buildings: Comparison of Calculation Methods: I Review of Algorithms. *Building Services Engineering Research and Technology* 11(3): 81-85.

Stefanizzi, P., A. Wilson, and A. Pinney. 1990. Internal Long-Wave Radiation Exchange in Buildings: Comparison of Calculation Methods: II Testing of Algorithms. *Building Services Engineering Research and Technology* 11(3): 87-96.

Stephenson, D.G., and, G.P. Mitalas. 1967. Cooling Load Calculation by Thermal Response Factors. *ASHRAE Transactions* 73 (1): III 1.1-1.7.

Stephenson, D.G., and G.P. Mitalas. 1971. Calculation of Heat Transfer Functions for Multi-Layer Slabs. *ASHRAE Transactions* 77(2): 117-126.

Stewart, J.P. 1948. Solar Heat Gain through Walls and Roofs for Cooling Load Calculations. *Transactions American Society of Heating & Ventilating Engineers* 54: 361-388.

Walton, G.N. 1980. A New Algorithm for Radiant Interchange in Room Loads Calculations. *ASHRAE Transactions* 86(2): 190-208.

Walton, G. 1983. *Thermal Analysis Research Program Reference Manual*. National Bureau of Standards.

Yazdanian, M., and J.H. Klems. 1994. Measurement of the Exterior Convective Film Coefficient for Windows in Low-Rise Buildings. *ASHRAE Transactions* 100(1): 1087.

## APPENDIX A: DERIVATION OF ENVIRONMENTAL TEMPERATURE

The environmental temperature model for room internal heat exchange is based on a cubic enclosure (Figure A1). One surface (area $A$) is at a temperature $T_1$ and the remaining surfaces (area $5A$) are at a mean temperature $T_2$. The mean air temperature is $T_a$. Nodes $T_1$ and $T_2$ are linked by a radiant conductance $AEh_r$, $T_2$ and $T_a$ are linked by a convective conductance $5Ah_c$ and $T_a$ and $T_1$ are linked by a convective conductance $Ah_c$. An expression for the environmental temperature $T_{ei}$ is derived in Appendix A5.1 of the CIBSE Guide. This expression can be derived more succinctly using a delta-to-star transformation.

Let $C_1 = 5Ah_c$, $C_2 = Ah_c$ and $C_3 = AEh_r$. $C_1$, $C_2$, and $C_3$ form a delta network that can be transformed exactly to a star network with conductances $K_1$, $K_2$, and $K_3$, where

$$C_1K_1 = C_2K_2 = C_3K_3 = C_1C_2 + C_2C_3 + C_3C_1 \tag{A1}$$

and the star temperature is a weighted mean

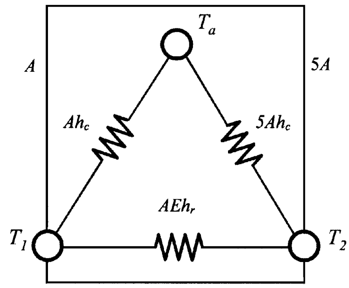


**Figure A1. CIBSE Guide enclosure model (links are conductances with units of W/K)**

$$T_S = \frac{K_1 T_1 + K_2 T_2 + K_3 T_a}{K_1 + K_2 + K_3} \tag{A2}$$

It is found after some manipulation that

$$K_1 = A\left(\frac{6}{5}Eh_r + h_c\right) \qquad \text{and} \qquad K_2 = 5A\left(\frac{6}{5}Eh_r + h_c\right) \tag{A3}$$

These are proportional to the respective areas; thus there is a transmittance of $\frac{6}{5}Eh_r + h_c$ to each surface. Further, the conductance linking $T_a$ is

$$K_3 = 6A\left(\frac{6}{5}Eh_r + h_c\right)h_c/\left(\frac{6}{5}Eh_r\right) \tag{4}$$

With $E = 0.9$, $h_r = 5.7$ W/(m$^2$·K) and $h_c = 3$ W/(m$^2$·K); and, as in the CIBSE *Guide*, writing $\Sigma A$ in place of $6A$, $K_3 = 4.46\Sigma A$, or $h_a \Sigma A$ where $h_a$ is about 4.5 W/(m$^2$·K) as given in the *Guide* (see Eq. A5.118).

Also, noting that the mean surface temperature $T_m = \frac{1}{6}T_1 + \frac{5}{6}T_2$, the star temperature, i.e., the environmental temperature is

$$T_{ei} = \frac{\frac{6}{5}Eh_r T_m + h_c T_a}{\frac{6}{5}Eh_r + h_c} = 0.672T_m + 0.328T_a \tag{A5}$$

or approximately, $\frac{2}{3}T_m + \frac{1}{3}T_a$ (Eq. 5.101 in the CIBSE *Guide*).

The star network is shown in Figure A2. The ambient temperature is linked to the two surface nodes by conduction and to $T_{ai}$ by infiltration. These additional heat loss paths are shown in Figure A3, which is similar in form to Figure A5.6 in the CIBSE *Guide*.

Although it is mathematically rigorous, two assumptions have been made in the above analysis that limit its applicability:

1. The model does not take account of the possibly different emissivities $\varepsilon_1$ and $\varepsilon_2$ for surfaces 1 and 2, nor their different convective coefficients $h_{c1}$ and $h_{c2}$. If this is included, the transmittance between say $T_1$ and $T_{ei}$, which should depend on $\varepsilon_1$ and $h_{c1}$ only, depends additionally on $\varepsilon_2$ and $h_{c2}$ values.

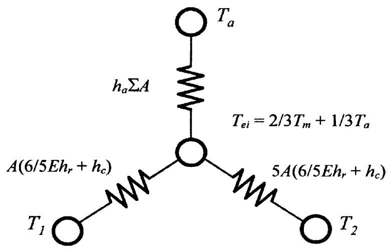


**Figure A2. Merging of radiant and convective exchanges and environmental temperature**

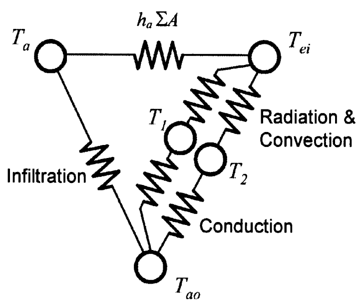


**Figure A3. Inclusion of loss conductances: Infiltration and conduction**

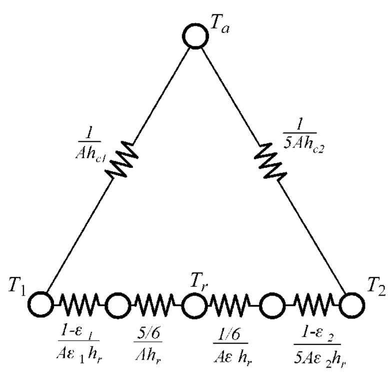


**Figure A4. Figure 1 modified so as to derive a valid global temperature (Links are shown as resistances with units of K/W)**

2. The analysis is based on a cube from which the factor 6/5 comes. If Surface 1 is small compared with the total surface area,

$$T_{ei} \Rightarrow \frac{\varepsilon_1 h_r T_2 + h_{c1} T_a}{\varepsilon_1 h_r + h_{c1}} \tag{A6}$$

and it has this form when Surface 1 decreases to zero. But $T_{ei}$ is now based on the thermal parameters $\varepsilon_1$ and $h_{c1}$ of a non-existent surface. Thus $T_{ei}$ is a dubious (if not valid) construct.

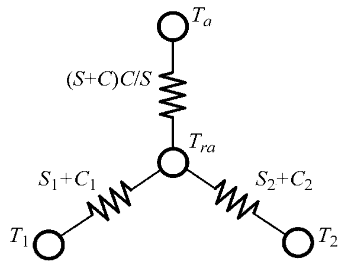


**Figure A5. Corresponding star-based model**

In order to find a valid global temperature with links to surfaces which validly merge radiant and convective exchange, the radiant resistance ($1/AEh_r$ above) between $T_1$ and $T_2$ must first be separated into its emissivity-based and geometrically based resistances as shown in Fig A4. A radiant star node $T_r$ is located on the geometrical resistance $1/Ah_r$ dividing it in the ratio 5:1. A conductance $S_1$ is then formed between $T_1$ and $T_r$, given by:

$$\frac{1}{S_1} = \frac{1-\varepsilon_1}{A\varepsilon_1 h_r} + \frac{\frac{5}{6}}{Ah_r} = \frac{1}{Ah_r}\left(\frac{1-\frac{1}{6}\varepsilon_1}{\varepsilon_1}\right) \tag{A7}$$

$S_2$ is formed similarly, with $5A$ replacing $A$ and $\varepsilon_2$ in place of $\varepsilon_1$.

Now the air temperature $T_a$ is linked to $T_1$ and $T_2$ through conductances $C_1 = Ah_{c1}$ and $C_2 = 5Ah_{c2}$ respectively. A global node $T_{ra}$ can now be formed as a weighted mean of the radiant and air nodes:

$$T_{ra} = \frac{ST_r + CT_a}{S + C} \tag{A8}$$

where $S = S_1 + S_2$ and $C = C_1 + C_2$. $T_{ra}$ is linked to $T_1$ by the combined radiant and convective conductances $S_1 + C_1$ and to $T_2$ similarly. This transformation is exact only if $S_1/C_1 = S_2/C_2$. Figure A5 shows this model.

In this formulation, the link to $T_1$ only involves the thermal parameters of Surface 1 and $T_{ra}$ is independent of them if the area of Surface 1 becomes zero. The factor of 1/6 in the expression for $S_1$ again derives from the choice a cube for the enclosure. Strictly speaking, a two-surface enclosure is too simple to derive a radiant temperature $T_r$ because $T_r$ was located arbitrarily on $1/Ah_r$. For a three-surface enclosure, $T_r$ is defined exactly; and for four or more, it becomes an approximate, though convenient, index. The previous argument, however, shows how a valid global enclosure temperature can be constructed using the enclosure used in the CIBSE Guide.