A STUDY ON THE INFLUENCE OF THE FENESTRATION COMPONENT ON THE COOLING LOAD CALCULATION USING THE HEAT BALANCE METHOD

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Abstract. *In this work was analyzed the influence of the fenestration heat gain component on the results of cooling load using the heat balance method. The heat balance method was first proposed in the middle 90s and incorporated in the ASHRAE 2001 Handbook of Fundamentals. The method is based on the energy conservation principle, which is applied to all surfaces (inner and outer sides) and to the air in the building. In this work, a brief review of the heat balance method is also presented. Next, the fenestration component is discussed. The fenestration term refers to the glazing apertures (windows) existing in a building to be air-conditioned. Fenestrations are important elements to be considered when doing cooling load calculations. It is shown that the quantifying of the heat gain through fenestrations obeys to different methodologies from the used for opaque surfaces. Such methodologies are different in two aspects: solar radiation and conduction. In quantifying the solar radiation through windows, reflection, absorption and transmission should be considered altogether. The thermal inertia of the glazing materials should be considered for the transient conduction quantifying. This work is concluded with the presentation and discussion of the results of cooling load for an example room with different glazing materials.*

Keywords*: air conditioning, thermal comfort, HVAC, cooling load calculation*

1. INTRODUCTION

The use of air conditioning was proven to be advantageous not only for human comfort, but also for controlling the environment (temperature, humidity and air quality) during processing of several products (e.g., electronics, pharmaceuticals, food, textiles and chemicals).

One of the prior steps during the design of air conditioning systems is related to the calculations of energy requirements for keeping a zone in desirable temperature, humidity and air quality. Such calculations will be referred here as cooling load calculations.

The computers advent in the second half of the last century, make possible the application of different methodologies for cooling load calculations. One of the most applied methodologies is the TETD/TA (total equivalent temperature differential method with time averaging). Originally presented in the 1967 ASHRAE Handbook of Fundamentals, this method does not require iterations and is convenient for spreadsheet implementation. On the other hand, its use is complicated by a number of empirical variables. Paying attention to these difficulties for the TETD/TA and similar methods, Liesen and Pedersen (1997), Pedersen *et al.* (1997), McClellan and Pedersen (1997), Pedersen *et al*. (1998) had proposed a more rigorous method, published in the 2000 ASHRAE Handbook of Fundamentals, based on the energy conservation and termed Heat Balance Method.

2. FUNDAMENTALS OF THE HEAT BALANCE METHOD

The heat balance method involves four distinct processes.
- Outside face heat balance
- Wall conduction
- Inside face heat balance
- Air heat balance

During the quantifying of these processes three fundamentals hypothesis are made:
- The air in the zone is modeled as well mixed, i.e., uniform temperature throughout the zone.
- The surfaces of the zone (walls, windows, floor, ceiling) have uniform temperatures and radiant properties are uniform and diffuse
- The conduction is treated as one-dimensional.

The heat balance method assures the energy conservation for the balances made for the surfaces and for the air zone. The method involves the simultaneous solution of thermal balance equations for the zone air, interior surfaces, floor, ceiling, inner and outer wall sides. Next, is shown a summary of these balance equations for opaque surfaces, where Eqs (1), (9) and (22) represents the balances for external surfaces, internal surfaces and zone air, respectively. . A more detailed discussion of these equations is presented in Costa *et al.* (2006).

The heat balance in W/m^2 for the external surface j in the hour θ:

$$q''_{cond,ext,j,\theta} = q''_{solar,ext,j,\theta} + q''_{conv,ext,j,\theta} + q''_{rad,ext,j,\theta} \tag{1}$$

The absorbed solar radiation, external convective exchange and the external radiation exchange are given by:

$$q''_{solar,ext,j,\theta} = \alpha.G_t \tag{2}$$

$$q''_{conv,ext,j,\theta} = h_{c,ext}\ (t_o - t_{os,j,\theta}) \tag{3}$$

$$q''_{rad,ext,j,\theta} = h_{r,g}(t_g - t_{os,j,\theta}) + h_{r,sky}(t_{sky} - t_{os,j,\theta}) \tag{4}$$

Where

$$h_{r,g} = \varepsilon\,\sigma\left[\frac{F_{s,g}(t_g^4 - t_{os,j,\theta}^4)}{t_g - t_{os,j,\theta}}\right] \quad h_{r,sky} = \varepsilon\,\sigma\left[\frac{F_{s,sky}(t_{sky}^4 - t_{os,j,\theta}^4)}{t_{sky} - t_{os,j,\theta}}\right] \tag{5}$$

From Eq. (2), "G_t" is the total solar irradiation in W/m^2 and "α" is the surface absorptivity. "G_t" is composed by direct, diffuse and reflected terms, which are function of angles measured from wall, solar rays and ground.

"t_o" and "t_i" are the outdoor and indoor air dry bulb temperature (in C).

Typical correlations for the convection coefficients "$h_{c,ext}$" and "$h_{c,int}$" used in Eqs. (3) and (14) are given in Incropera and DeWitt (2002).

In Eq. (4), the radiation exchange with low temperature sources (i.e., long wavelength) is modeled considering opaque, diffuse and gray surfaces. In this way, the radiation exchange to the atmosphere is modeled as radiation to a hypothetical surface with an effective sky temperature "t_{sky}". Other radiation parameters are: "ε", the surface emissivity; "σ", the Stefan-Boltzmann constant (= $5.67.10^{-8}$ W/(m^2.K^4)); "$F_{s,g}$", the view factor surface to ground; "$F_{s,sky}$", the view factor surface to sky; "t_g", the ground temperature.

The external conduction flux is given by Eqs. (6) and (7):

$$q''_{cond,ext,j,\theta} = -Y_0 t_{is,j,\theta} + X_0 t_{os,j,\theta} + H_{ext,j,\theta} \tag{6}$$

$$H_{ext,j,\theta} = -\sum_{n=1}^{N_y} Y_n t_{is,j,\theta\text{-}n\delta} + \sum_{n=1}^{N_x} X_n t_{os,j,\theta\text{-}n\delta} + \sum_{n=1}^{N_q} \varphi_n q''_{cond,ext,j,\theta\text{-}n\delta} \tag{7}$$

From Eq. (7), "Y_n", "X_n", "Z_n", and "ϕ_n" are conduction transfer function (CTF) coefficients based on material type and surface thickness. Typical values for the coefficients are given in McQuiston et al. (2000).

Also from Eq. (7), "$t_{is,\,j,\,\theta\,-\,n\,\delta}$" refers to interior j surface temperature in the time "θ - nδ".

The external surface temperature:

$$t_{os,j,\theta} = \frac{\alpha.G_t + h_{c,ext}\ t_o + h_{r,g} t_g + h_{r,sky} t_{sky} + Y_0 t_{is,j,\theta} - H_{ext,j,\theta}}{X_0 + h_{c,ext} + h_{r,g} + h_{r,sky}} \tag{8}$$

It should be pointed that Eq. (8) is solved iteratively because the heat exchange coefficients "$h_{c,ext}$", "$h_{r,g}$" and."$h_{r,sky}$" depends on its external surface temperature. A similar iterative procedure is required for the internal surface temperatures given by Eq. (21).

The heat balance for the interior surface j in the hour θ:

$$q''_{cond,int,j,\theta} + q''_{solar,int,j,\theta} = q''_{conv,int,j,\theta} + q''_{rad,int,j,\theta} \tag{9}$$

The interior conduction flux is given by:

$$q''_{cond,int,j,\theta} = -Z_0 t_{is,j,\theta} + Y_0 t_{os,j,\theta} + H_{int,j,\theta} \tag{10}$$

$$H_{int,j,\theta} = -\sum_{n=1}^{N_y} Z_n t_{is,j,\theta\text{-}n\delta} + \sum_{n=1}^{N_x} Y_n t_{os,j,\theta\text{-}n\delta} + \sum_{n=1}^{N_q} \varphi_n q''_{cond,int,j,\theta\text{-}n\delta} \tag{11}$$

Transmitted solar radiation for interior sides of all surfaces except floor, and for the floor:

$$q''_{sol,int,j,\theta} = \sum \dot{q}_{dif} \Big/ \sum_{j=1}^{N} A_j \tag{12}$$

$$q''_{sol,int,floor,\theta} = \sum \dot{q}_{dif} \Big/ \sum_{j=1}^{N} A_j + \dot{q}_{dir} / A_{floor} \tag{13}$$

In all equations, “A” is the surface area and “N” is the number of surfaces.
Interior convective exchange

$$q''_{conv,int,j,\theta} = h_{c,int}\ (t_{is,j,\theta} - t_i) \tag{14}$$

Radiation exchange components for interior side of surfaces

$$q''_{rad,int,j,\theta} = -q''_{rad\text{-}int,in,j,\theta} + q''_{rad\text{-}surf,int,j,\theta} \tag{15}$$

The first right hand side term in Eq.(15) is the radiant exchange due to internal heat sources and is given by Eq. (16) divided by the total internal surfaces area.
The radiation exchange component due to M internal sources

$$\dot{q}_{rad\text{-}int,in,j,\theta} = \sum_{k=1}^{M} \dot{q}_{k,\theta} F_{rad,k} \tag{16}$$

From Eqs. (16) and (27), “$F_{conv,j}$” and “$F_{rad.k}$” represents the convective and radiant fractions for internal heat sources (people, equipments and lights). The convective fractions corresponds to the instantaneous heat exchange with the zone air whereas the radiant fraction corresponds to radiation heat exchange with the zone internal surfaces.
The second right hand side term in Eq. (15) is the radiation exchange between the internal surfaces. This exchange is modeled using the mean radiant temperature (MRT) approach given in Eqs. (17) to.(20). In this approach “q''_{bal}” is a correction term for closing the balance. Therefore, the radiation exchange component is given by:

$$q''_{rad\text{-}surf,int,j,\theta} = h_{r,j}\left(t_{is,j,\theta} - t_{f,j}\right) - q''_{bal} \tag{17}$$

$$h_{r,j} = \sigma\, F_{j,f}\, \frac{t_j^4 - t_{f,j}^4}{t_j - t_{f,j}}\ ;\ F_{j,f} = \frac{1}{\dfrac{1-\varepsilon_j}{\varepsilon_j} + 1 + \left(\dfrac{A_j}{A_{f,j}}\right)\dfrac{1-\varepsilon_{f,j}}{\varepsilon_{f,j}}} \tag{18}$$

$$A_{f,j} = \sum_{i=1}^{N} A_i\left(1 - \delta_{ij}\right)\ ;\ \varepsilon_{f,j} = \frac{\sum_{i=1}^{N} \varepsilon_i A_i\left(1 - \delta_{ij}\right)}{\sum_{i=1}^{N} A_i\left(1 - \delta_{ij}\right)}\ ;\ t_{f,j} = \frac{\sum_{i=1}^{N} \varepsilon_i t_{is,i,\theta} A_i\left(1-\delta_{ij}\right)}{\sum_{i=1}^{N} \varepsilon_i A_i\left(1-\delta_{ij}\right)} \tag{19}$$

$$q''_{bal} = \sum_{j=1}^{N} A_j h_{r,j}\left(t_{is,j,\theta} - t_{f,j}\right) \Big/ \sum_{j=1}^{N} A_j \tag{20}$$

Interior surface temperature:

$$t_{is,j,\theta} = \frac{h_{c,int}\ t_i + q''_{rad\text{-}int,in,j,\theta} + h_{r,j} t_{f,j} + q''_{bal} + Y_0 t_{os,j,\theta} + H_{in,j,\theta} + q''_{sol,int,j,\theta}}{Z_0 + h_{c,int} + h_{r,j}} \tag{21}$$

Zone air heat balance:

$$\dot{q}_{conv,air,\theta} + \dot{q}_{inf,\theta} + \dot{q}_{sist,\theta} + \dot{q}_{int,conv,\theta} = 0 \tag{22}$$

Convective exchange for zone air:

$$\dot{q}_{conv,ar,\theta} = \sum_{j=1}^{N} A_j \; q''_{conv,in,j,\theta} \tag{23}$$

Heat exchange due to air infiltration:

$$\dot{q}_{inf,\theta} = \dot{q}_{inf,sens,\theta} + \dot{q}_{inf,latent,\theta} \tag{24}$$

$$\dot{q}_{inf,sens,\theta} = \dot{m}_a c_p \left(t_o - t_i\right) \tag{25}$$

$$\dot{q}_{inf,latent,\theta} = \dot{m}_a \left(w_o \text{-} w_i\right) h_{fg} \tag{26}$$

From Eqs (25) and (26): "$\dot{m}_a$" is the infiltration dry air mass flux, "c_p" , air specific heat at constant pressure, "w_o" and "w_i", absolute humidity for external and internal air , respectively; "h_{fg}", water vaporization enthalpy.

In Eq. (22), "$\dot{q}_{sist,\theta}$" is the system load. It corresponds to the heat that must be extracted by the air conditioning equipment.

The convective exchange due to M internal sources:

$$\dot{q}_{int,conv,\theta} = \sum_{j=1}^{M} \dot{q}_{j,\theta} \; F_{conv,j} \tag{27}$$

It should be pointed that "$\dot{q}_{dif}$" and "$\dot{q}_{dir}$" terms in Eqs. (12) and (13) are related to the fenestration components and will discussed in detail in section 4. In the next section is presented the solution procedure.

3. HEAT BALANCE METHOD CALCULATION SCHEME

The calculation steps for the heat balance method are given next:

1. Areas, properties and initial temperatures for surfaces during a 24 hour period.
2. Total solar irradiation and transmitted solar for all surfaces during the period.
3. Distribute the transmitted solar radiation for all internal surfaces during the period.
4. Specify all the internal heat sources during the period.
5. Distribute the radiation and convective fraction from internal heat sources for all surfaces during the period.
6. Calculate heat exchange due to air infiltration.
7. Iterate and verify convergence for conduction fluxes according to Fig. 1.
8. Show the results

```
For DayIter = 1 to MaxIterDay
     For θ = 1 to 24 (daily hours)
       For SurfIter = 1 to MaxIterSurf
         For j = 1 to N
           Solve Eqs. (8) and (21)
         Next j
       Check surfaces temperature convergence
       Case divergence, next SurfIter
       Solve Eq. (22) for q̇_sist,θ
     Next θ
     Check conduction fluxes convergence.
Case divergence, next DayIter
```

Figure 1. Heat balance cooling load calculation algorithm

4. FENESTRATION

The term fenestration refers to any aperture in a building with allowance for solar radiation passage. Usually the fenestration is associated with windows, their glazing materials, geometric form and shading devices. The radiation that strikes an unshaded window is divided between the reflected, absorbed and transmitted portion. In its turn, absorbed radiation is divided in an inward directed and an outward directed portion. Therefore, the heat flux due to solar radiation is the sum of the transmitted and inward directed portion. Because heat is also conducted through the glass whenever there is an outdoor-indoor temperature difference, the total heat balance for the windows must also include the conduction.

To solve all the complexities involved in fenestration one has to take into account the size of the window, their monochromatic radiation response properties and the sunbeam angle. All these complexities are solved with the spectral method. The spectral method involves the development of the solar heat gain coefficient, the fraction of the incident irradiance (i.e., incident solar energy) that passes through the window and penetrates in the zone.

The results of the spectral method are summarized by the TSHGF and ASHGF factors. The transmitted solar radiation that occurs in a unit area of standard (DSA) glass for a given orientation and time is referred to as the transmitted solar heat gain factor (TSHGF). Its units are given in W/m^2. Similarly, the absorbed solar inward directed heat gain that occurs in a unit area of standard glass (DSA) for a given orientation and time is referred to as the absorbed solar heat gain factor (ASHGF). Equations (28) and (29) gives the expressions for these two factors:

$$\mathrm{TSHGF} = G_D\left(\sum_{j=0}^{5} t_j[\cos\Theta]^j\right) + G_d\left(2\sum_{j=0}^{5}\frac{t_j}{j+2}\right) \tag{28}$$

$$\mathrm{ASHGF} = G_D\left(\sum_{j=0}^{5} a_j[\cos\Theta]^j\right) + G_d\left(2\sum_{j=0}^{5}\frac{a_j}{j+2}\right) \tag{29}$$

In the previous equations: "G_D" is the direct incident solar irradiation, "G_d" is the diffuse incident solar irradiation, "Θ" is the solar incidence angle.

From Eq. (28) the terms enclosed in parentheses represents transmissivity for direct and diffuse solar radiation, respectively. While at the same time, absorptivity for direct and diffuse solar radiation is given by the terms in parentheses from Eq. (29).

The coefficients "t_j" and "a_j" are related to the transmissivity and absorptivity for the standard glass. Their values are presented in the 2001 Fundamentals ASHRAE handbook.

Procedures for estimating solar heat gain assumes that a constant ratio exists between the solar heat gain through any given type of fenestration system and the solar heat gain through the standard (DSA) glass. SC abbreviates this ratio, called the shading coefficient. Thus for any type of glass the transmitted solar radiation is given by:

$$\mathrm{TSHG} = \mathrm{SC}\ \mathrm{TSHGF} \tag{30}$$

$$\mathrm{ASHG} = \mathrm{SC}\ \mathrm{ASHGF}\left(\frac{h_{c,int}}{h_{c,int}+h_{c,ext}}\right) \tag{31}$$

The units of the transmitted and absorbed solar radiation flux, TSHG and ASHG, are given inW/m^2.

At this point should be mentioned that previous section terms "$\dot{q}_{dir}$" and "$\dot{q}_{dif}$" in Eqs. (12) and (13) are obtained by multiplying sunlit and shaded areas, "A_{SL}" and "A_{shade}", of windows, by the first and second terms of the right hand side of Eq.(28).

For the closure of the heat balance cooling load calculation procedure a balance for windows should be made. The heat balance on windows must be treated differently than the heat balances on walls and roofs (Eqs. (1) and (9)). For this case, the following simplifying assumptions will be made:

A window contains very little thermal mass, so we will assume that it behaves in a quasi-steady-state mode.

The conductive resistances will be neglected in comparison with convective and radiant resistances. In this way, it has a uniform temperature.

Figure 1 shows the heat fluxes considered for a single pane window.

Writing the balance, results:

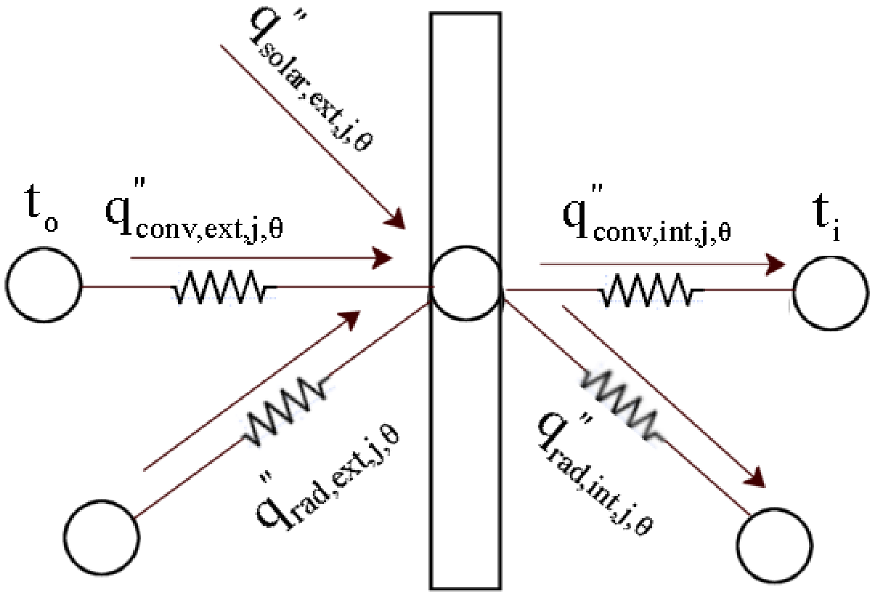

Figure 2. Schematics of heat balance processes in a single pane window

$$q''_{solar,ext,j,\theta} + q''_{conv,ext,j,\theta} + q''_{rad,ext,j,\theta} = q''_{conv,int,j,\theta} + q''_{rad,int,j,\theta} \tag{32}$$

When solved for the temperature "$t_{window,j,\theta}$", the single pane window balance given in Eq. (32) produces:

$$t_{window,j,\theta} = \frac{\alpha.G_t + h_{c,ext}\ t_o + h_{r,g} t_g + h_{r,sky} t_{sky} + h_{c,int}\ t_i + q''_{rad\text{-}int,in,j,\theta} + h_{r,j} t_{f,j} + q''_{bal}}{h_{c,int} + h_{r,j} + h_{c,ext} + h_{r,g} + h_{r,sky}} \tag{33}$$

Equation (33) should be added to the set given by Eqs. (8) and (21) in the algorithm given in section 3.

5. RESULTS

The heat balance calculation procedure for cooling load described in section 3 was implemented, using the Mathematica software, for a example zone with a single room located at -23^o S 25' W. Figure 3 illustrates the room geometry. The area values for the floor, ceiling, north, east, west, south and window surfaces are: 90, 90, 36, 36, 36, 28 and 8 m^2, respectively.

The following single values were used: $\varepsilon = 0.9$, $t_i = 22$ C, $q''_{rad\text{-}int,in,j,\theta} = 0.9\ W/m^2$, $F_{s,g} = 0.5$, $F_{s,sky} = 0.5$, $A_{sl} = 0.8\ A_{window}$, $t_{sky} = t_o - 6$ K,

The following assumptions were made:

- The outside surface and outdoors temperature are already known for all 24 hours period
- Except for "$\dot{q}_{sist,\theta}$" and "$\dot{q}_{conv,air,\theta}$" all the zone air heat balance terms in Eq. (22) were negligible;
- The reflected terms for calculating "G_t" were negligible;
- The ground temperature was assumed equal to the air;
- All the surfaces except south does not receive solar radiation.

Figure 4 (a) shows the outside surface and outdoor temperature distribution for the period. Figure (b) presents the values for the direct and diffuse radiation. The values for Fig 4 (b) are also known in advance.

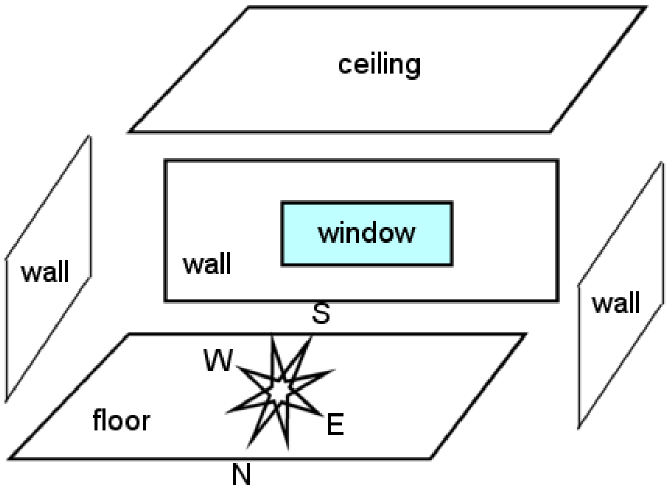

Figure 3. Geometric description for the single room example (north wall not shown)

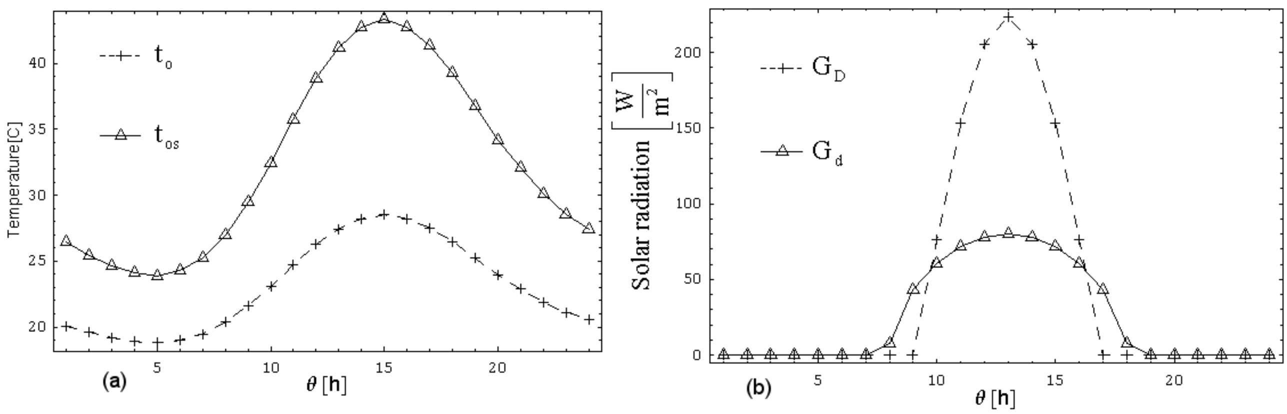

Figure 4. Temperature and radiation distributions: (a) outside surface and outdoor temperature, (b) direct and diffuse solar radiation

Some typical values after the calculations are given in Fig. 5(a) and Fig. 5(b). The calculated conduction flux for the first and last iterations are given in Fig. 5(a). Figure 5(b) presents a typical calculated system load. Comparison with Figs. 4(a) and 4(b) shows that the sinusoidal nature of the known "t_{os}" and "t_o" given in Fig. 4(a) is also observed for the conduction flux and equipment load, although the respective peaks and valleys occurs at different hours.

The effect of neglecting the heat balance equation for the window (Eq. 33) on temperature for a typical wall is shown in Figs. 6(a). As can be seen the effect on the temperature is very small and more noticeable for the nightly hours. Figure 6(b) presents the effect on the system load.

The effect of shading coefficient variation, corresponding to different glazing materials, is shown in Fig 7(a) and 7(b) for wall and window temperatures. Analysis shows very small dependence of the shading coefficients. From Fig. 7(a), a small difference is more noticeable for walls during afternoon hours.

Figure 8(a) presents the effect of using different external to internal convection coefficient ratios on the window temperature. The differences between temperatures are more noticeable than previous cases. As can also be observed the higher ratio produces a small temperature for the early hours. Figure 8(b) shows that the effect on the system load is negligible.

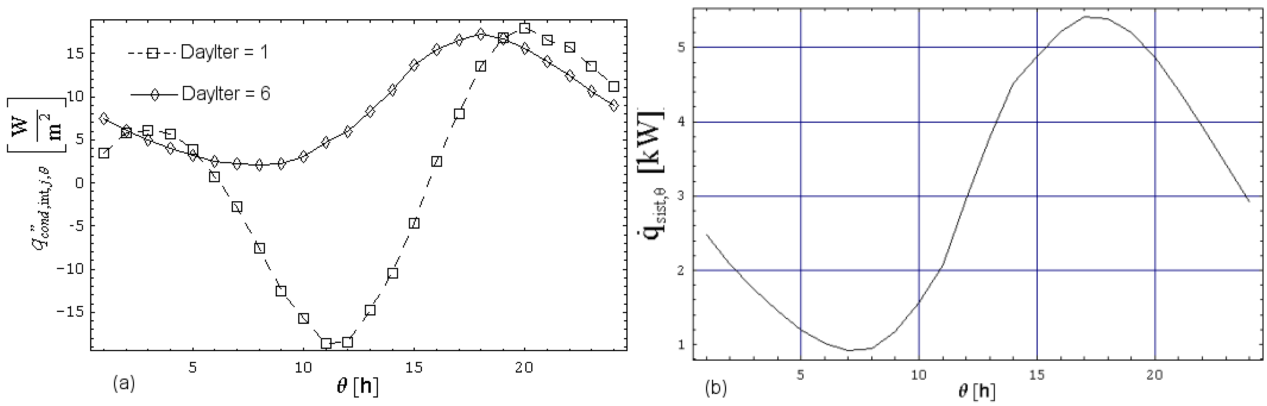

Figure 5. (a) Typical conduction flux for the 1st and 6th iteration, (b) equipment cooling load

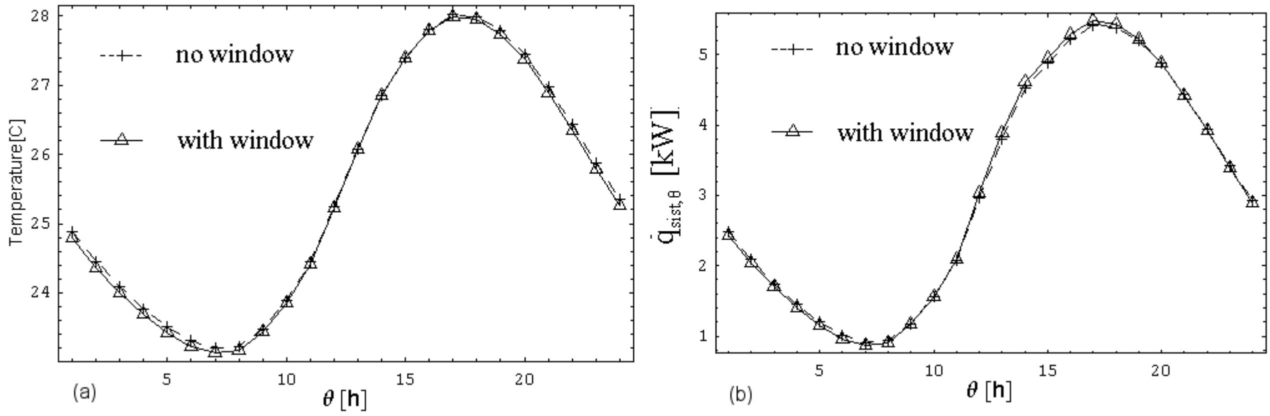

Figure 6. Effect of accounting the window balance equation: (a) temperature distribution; (b) equipment cooling load

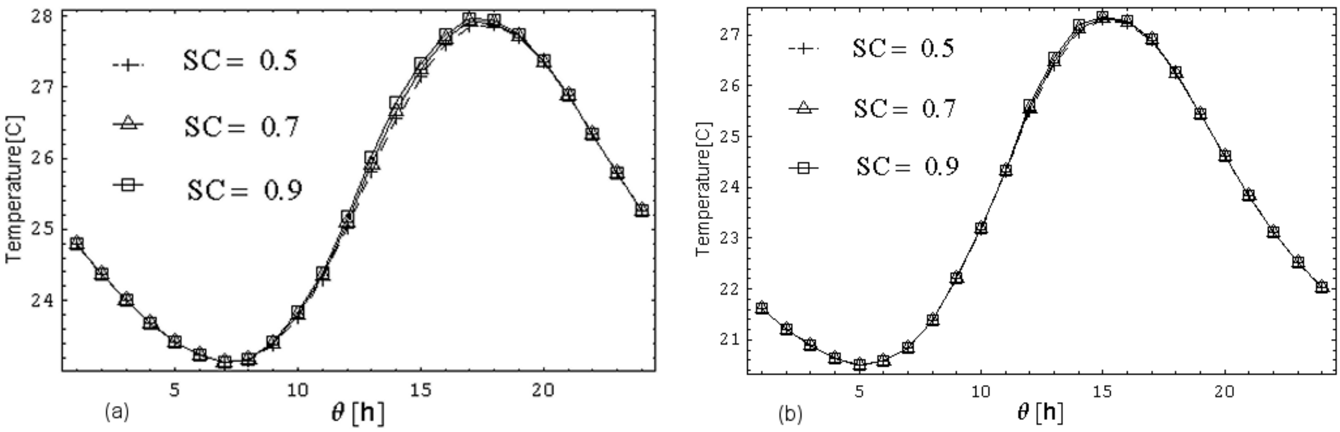

Figure 7.Effect of the shadow coefficient on temperature distribution: (a) wall, (b) window

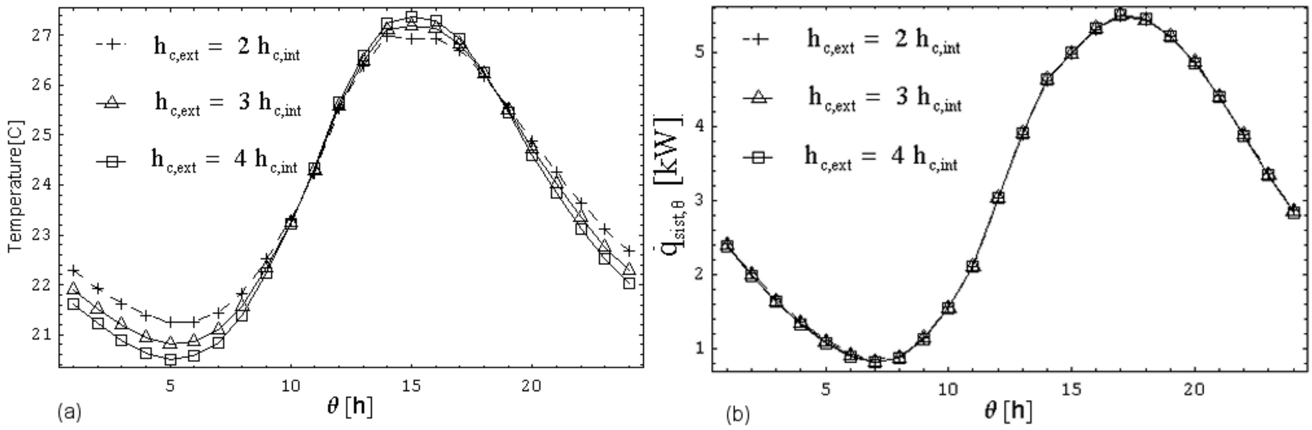

Figure 8. Effect of the relation between convective coefficients: (a) window temperature, (b) equipment cooling load

6.CONCLUSIONS

The heat balance method allows for precisely considering the effect of the fenestration in the total cooling load. Ours results for an example room suggests that including a separate balance equation for the window does not introduce significant deviation. Also, from our results, the type of glazing does not result in significant differences in temperatures. The effect of different convective conditions points to more noticeable temperature differences. Finally, the results for the system load points to no dependence of the fenestration related parameters used in this study..

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