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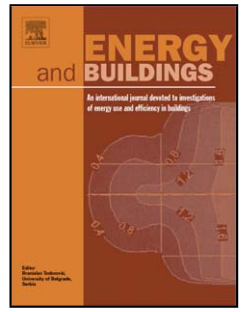

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Solar radiation and cooling load calculation for radiant systems: definition and evaluation of the Direct Solar Load

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Abstract

The study of the influence of solar radiation on the built environment is a basic issue in building physics and currently it is extremely important because glazed envelopes are widely used in contemporary architecture.

In the present study, the removal of solar heat gains by radiant cooling systems is investigated. Particular attention is given to the portion of solar radiation converted to cooling load, without taking part in thermal absorption phenomena due to the thermal mass of the room. This specific component of the cooling load is defined as the Direct Solar Load.

A simplified procedure to correctly calculate the magnitude of the Direct Solar Load in cooling load calculations is proposed and it is implemented with the the Heat Balance method and the Radiant Time Series method.

The F ratio of the solar heat gains directly converted to cooling load, in the case of a low thermal mass radiant ceiling, is calculated for different kinds of office rooms with a large glazed external surface. An example of cooling load calculation developed with the proposed procedure is given.

Keywords: solar radiation, solar heat gains, cooling load, radiant panels, heat balance, radiant time series.

1. Introduction

The use of large glazed envelopes has become common in contemporary architecture, especially in some kinds of buildings: shopping malls, atrium buildings, offices, airports and often even in museums [1, 2]. Large surfaces of glazed envelope may cause, especially in warm climates, a considerable increase in the cooling load of a building, due to the increased solar heat gain. Solar gains should be reduced as much as possible by using external screens (brise soleil) and high performance glass [3]. The remaining heat gain should then be removed by an HVAC system, because of the high energy content of the solar radiation. Solar radiation is nevertheless a particular kind of thermal radiation, with a high energy concentration in the visible spectral region (350 – 750 nm), and is thus substantially different from the thermal radiation emitted by other thermal sources (lights, persons, heated surfaces…) which is mainly concentrated in the infrared spectral region, and especially in the far infrared (wavelength longer than 2500 nm). The solar spectrum beyond the atmosphere is very similar to the spectrum of a black body at a temperature of 5800 K [4, 5], but not all of the extra-atmospheric radiation reaches the surface of the earth because of absorption by carbon dioxide and water vapour, and because of the screening due to dust and pollutants in the atmosphere. Absorption and screening are particularly concentrated in the infrared spectral region (750 – 4000 nm).

The analysis of solar radiation effects on the built environment should thus be separated from the study of other kinds of thermal radiation effects, because of their different spectral composition and because materials exhibit different absorptance and transmittance properties for solar radiation and for infrared thermal radiation.

In this paper, a study of solar radiation effects on radiant cooling systems is presented and a procedure to correctly evaluate cooling loads due to solar heat gains is proposed.

In the literature [6, 7], it is reported that solar radiation has the effect of improving the cooling capacity of radiant systems, but it is not reported how to correctly evaluate this improvement. The cooling capacity of the radiant system and the cooling load in the room are directly correlated. In this paper a procedure to improve cooling load calculations in rooms equipped with radiant systems is proposed and a definition of the different components of the cooling load is given: Room Load and Direct Solar Load.

Particular attention is given to the definition and evaluation of the Direct Solar Load, which represents the proportion of the solar heat gain that is removed by a radiant system before it can contribute to the thermal absorption processes that are due to the room thermal mass. This parameter can also be considered as an improvement of the radiant system cooling capacity due to the solar radiation entering the room.

The F ratio of the solar heat gains converted to Direct Solar Load is calculated for a low thermal mass radiant ceiling, simulating different office rooms with a fully glazed façade.

The proposed procedure can be easily implemented by the usual calculation methods, as shown by a calculation example.

2. Cooling load calculations for all-air systems

2.1Heat gains and cooling load

In cooling load calculations it is essential to use a multi-step analysis tool to be able to simulate the effect of load distribution during the whole day. Heat gains and cooling loads have different values at different times of day, because the building stores thermal energy (especially due to solar radiation) through its thermal mass and after a time lag re-emits it into the room. A specific radiant heat gain at time T_{θ}, may therefore have its actual effect on the cooling system only at time $T_{\theta+n}$.

The most common calculation methods, such as the Heat Balance method (HB) or the Radiant Time Series method (RTS), use the following definitions of *heat gain* and *cooling load* [8] in order to deal with this phenomenon:

- the *heat gain* (HG) is the amount of heat generated or introduced into the room at a specific time T_{θ}
- the *cooling load* (CL) is the amount of heat that must be removed from the room at the same time T_{θ}, in order to maintain the design conditions of temperature and humidity
- the *heat extraction rate* (HER) is the actual rate at which heat is removed from the room by the HVAC system at the time T_{θ} (it depends on the room temperature fluctuations with respect to the

design temperature, which depend in turn on the control system and on the characteristics of the HVAC system).

The thermal mass of the room thus correlates heat gains and cooling loads. Heat gains are nevertheless composed of convective gains (CHG) and radiant gains (RHG): the former, acting directly on air, are not involved in thermal mass absorption phenomena, while the radiant gains are partly absorbed by the elements in the room that have a non-negligible thermal mass. Only a reduced proportion of the absorbed radiation is re-emitted into the room, after a time lag, by convection and far infrared radiation. The thermal mass thus operates on radiant heat gains, delaying and dimming their effect, while convective heat gains directly become cooling loads.

The heat extraction rate is a parameter that is mainly useful in calculating energy use over time; it is not needed for calculating design peak cooling load. It will therefore not be considered in the following analysis.

2.2 Solar heat gains in the Heat Balance and Radiant Time Series methods

The Heat Balance method (HB) is identified by ASHRAE as the fundamental methodology of peak cooling load calculation [8]. The Radiant Time Series method (RTS), is a simplified cooling load calculation method, based on the HB, that replaces all the other simplified (non-heat-balance) methods [8, 9].

The hourly radiant gains are calculated separately from the hourly convective gains in both of these calculation methods (Figure 1).

In the HB method, Solar Heat Gains (SHG) are treated separately from the other radiant heat gains, and the fraction of the solar radiation transmitted by the glazing is divided from the inward-flowing fraction of the solar radiation absorbed by the glass, which is added to the flux caused by conduction through the glazing.

The transmitted radiation is incident directly onto surfaces in the zone and is accounted for in the internal surface heat balance. It is possible to calculate the actual position of a beam of solar radiation, but each internal surface is by assumption considered to be uniformly irradiated, although the irradiation values among surfaces are different.

The hourly cooling loads are derived from the heat balance of each internal surface in the zone, using an iterative process [8-10].

In the RTS, hourly radiant heat gains are converted to hourly cooling loads using radiant time factors, the coefficients of radiant time series. Two different series of radiant time factors are used: one for the directly transmitted solar heat gains, which are assumed to be distributed over the floor only, and one for all the other types of heat gains (including the diffuse solar heat gains), which are assumed to be uniformly distributed over all of the internal surfaces [11].

[place Figure 1 approximately here]

3. Cooling load calculation for radiant systems

3.1 Radiant heat gains and radiant cooling systems

Radiant cooling systems differ from all-air systems because they exchange heat with the surroundings both by convection and by radiation, while all-air system only exchange heat by convection.

When radiant heat gains act on the cooled radiant surface, their effect is direct, delayed only by the action of the thermal mass of the radiant system. If a radiant cooling system is used, the radiant heat gains should thus be divided into the fraction acting on the cooled surface and directly removed by it and the fraction absorbed and re-emitted with a time lag by the action of the room thermal mass [12,13].

Since radiant heat gains produced by internal sources are very low compared to the solar heat gains (SHG), the separation mentioned above can be applied to the solar heat gains only, to simplify calculations.

3.2 Solar heat gains and Direct Solar Load

When solar radiation enters a room, it can be directly incident on the floor surface and on part of the walls. It is then reflected toward the upper part of the walls and toward the ceiling. The cooled radiant surface can be positioned on the floor, on the ceiling or on the walls.

In order to obtain a clear characterization of each component of the cooling load, the *F* ratio of the solar heat gains transmitted through the glazing and directly converted to cooling load by the action of the cooling surface is defined as the *Direct Solar Load* (DSL):

$$\mathrm{DSL} = \mathrm{F} \cdot \mathrm{SHG} \tag{1}$$

If low thermal mass panels such as plasterboard, gypsum board or metallic panels are used, the effect of their thermal mass can be considered negligible and it can be omitted in calculations. The DSL can thus be considered as an instantaneous load (equation 2). Low thermal mass panels are usually mounted on ceilings or walls and rarely on floors.

$$\mathrm{DSL}_{\theta} = \mathrm{F} \cdot \mathrm{SHG}_{\theta} \tag{2}$$

If a high thermal mass radiant system is used, such as a floor cooling system with a concrete or gypsum layer between the floor covering and the water pipes, the effect of the SHG is delayed in time by the effect of the thermal mass of the cooling system. The DSL can then no longer be considered as an instantaneous load (equation 3).

$$\mathrm{DSL}_{\theta} = \mathrm{F} \cdot \mathrm{SHG}_{\theta-\mathrm{n}} \tag{3}$$

The value of the DSL is not reduced by the effect of the thermal mass of the room, neither in the case of low thermal mass radiant systems nor in the case of high thermal mass systems. The DSL can thus be considered as a further load that is removed by the radiant system, to be added to the load calculated by using the following equation:

$$\frac{\dot{Q}_s}{A} = h_{tot}\left(T_s - T_{op}\right) \tag{4}$$

The DSL thus represents an improvement of the radiant system's cooling capacity, one that is due to the fact that solar radiation arrives directly on the radiant surface, as reported in the literature [6, 7]. The solar radiation increases the radiant surface temperature of the cooling system and through conduction and convection it increases the water return temperature. Since the cooling capacity of the system is a linear function of the water temperature difference between supply and return (equation (5)), the impact of the solar radiation on the radiant surface engenders an increase of the actual cooling capacity of the system.

$$\dot{Q}_{cl} = \dot{m} \cdot c_p \cdot \left(T_{w,in} - T_{w,out}\right) \tag{5}$$

If the radiant surface temperature increases in this way, the control system reacts by increasing the water flow rate or by reducing the supply water temperature, in order to reinstate the set point of the room operative temperature. Hence, in such systems, the cooling capacity increases because of the increased water flow or because of the increase in the water temperature difference, and the surface temperature of the radiant system is maintained constant.

3.3 Room Load and Cooling Load

The other component of the cooling load, in a room equipped with a radiant system, is defined as *Room Load* (RL). It is generated by the convective heat gains at time θ, and the radiant heat gains at time θ –n (the radiant heat gains considered here do not include the solar gains already accounted for in the DSL calculation). The effect of the thermal mass of the room is thus concentrated in this component:

$$RL_{\theta} = CHG_{\theta} + RHG_{\theta-n} \tag{6}$$

If the radiant system is the only cooling system in the room, the cooling load is calculated as the sum of the RL and the DLS:

$$\mathrm{CL}_\theta = \mathrm{RL}_\theta + \mathrm{DSL}_\theta \tag{7}$$

If a primary air system is combined with a radiant system, the RL must be divided into two components: Panels Room Load (PRL) and Air Room Load (ARL). The PRL is the part of the RL removed by the radiant system and the ARL is the part of the RL removed by the primary air. They can be expressed by the following equations:

$$\mathrm{PRL}_\theta = \beta \cdot \mathrm{CHG}_\theta + \gamma \cdot \mathrm{RHG}_{\theta-n} \tag{8}$$

$$\mathrm{ARL}_\theta = (1-\beta) \cdot \mathrm{CHG}_\theta + (1-\gamma) \cdot \mathrm{RHG}_{\theta-n} \tag{9}$$

$$\mathrm{RL}_\theta = \mathrm{PRL}_\theta + \mathrm{ARL}_\theta \tag{10}$$

3.4 Cooling load calculation procedure

The following procedure is proposed for cooling load calculations in rooms equipped with radiant systems, in which the DSL is considerable:

- Separation between convective heat gains and radiant heat gains
- Separation between solar heat gains and other radiant heat gains
- Evaluation of the *F* ratio of solar heat gains involved in Direct Solar Load generation
- Calculation of the Direct Solar Load
- Calculation of the Room Load
- Calculation of the Cooling Load

The procedure can be implemented by the HB method, multiplying the transmitted part of the solar radiation (which is the actual SHG) by the proper F value. This part of the SHG will generate the DSL. The remaining part of the transmitted solar radiation (1-F) can be normally accounted for in the heat balance of the internal surfaces. At the end of the iterative procedure the HB will provide the hourly RL, which when added to the hourly DSL, will provide the hourly CL.

When the procedure is implemented by using the RTS method, the F values are applied to the SHG, providing the hourly DSL. The radiant time factors are instead applied to the remaining part of the SHG (1-F), providing, together with the other "non-solar" heat gains, the hourly RL, as explained in section 2.2.

The procedure can be expressed by modifying the scheme reported in Figure 1, specifying the division of solar gains between the *F* ratio, generating the DSL, and the (1-*F*) ratio, generating the RL, as shown in Figure2.

[place Figure 2 approximately here]

4. Cooling load and cooling capacity

The duty of the radiant system is to remove the calculated cooling load by means of radiant heat transfer between internal room surfaces and solar radiation and by means of convective heat transfer with air. Even if these phenomena happen at the same time, it is interesting, in terms of analysis, to consider seprately the component due to the removal of the RL and the component due to the removal of the DSL . The cooling capacity of a radiant system is usually calculated by means of equation (4), which expresses the surface heat transfer of the radiant surface. If the radiant system is the only cooling system in the room, the cooling capacity calculated by using equation (4) must be equal to the RL.

The cooling capacity is substantially limited by radiant surface temperature limits, due to the risk of condensation and the requirements for comfort [14, 15, 16]. Under the conditions that are typical of office buildings or dwellings, the floor surface temperature should not be lower than 19°C for comfort reasons, and the surface temperatures of ceilings, floors and walls should not be lower than the dew-point temperature, due to the risk of condensation. Since the surface heat transfer coefficient is a defined factor for these systems [17, 18], the cooling capacity limits of radiant systems can be easily calculated (Table 1).

[place Table 1 approximately here]

If RL exceeds the maximum cooling capacity of the radiant system (i.e., Q_{max}), the radiant system is not able to remove the whole RL and thus a supplementary air system is required.

The DSL is instead removed by the radiant system without any limitation. As previously reported, the DSL can be considered as an improvement of the radiant system cooling capacity due to the fact that the solar radiation arrives directly at the radiant surface.

This explains why, in the literature, it is reported that a normal radiant floor has a cooling capacity of up to 50 W m^{-2}, but if it is positioned under solar radiation its cooling capacity can exceed 100 W m^{-2} [6, 7]. The performance improvement is due to the DSL.

The DSL cannot exceed the cooling capacity of the radiant system, but it can oexceed the cooling unit capacity. Under these conditions part of the DSL is not removed by the system. The cooling unit must therefore be accurately sized to accord with the DLS.

The total cooling capacity of the radiant system corresponds to the enthalpic flux calculated by using equation (5). If the radiant system is the only cooling system, the total cooling capacity corresponds to the CL, and thus to the sum of the RL and the DSL.

If a primary air system is used to supplement the radiant system, the cooling load is equal to the sum of the total cooling capacity of the radiant system and the total cooling capacity of the air system.

In order to be able to insert the total enhanced cooling capacity of the radiant system into equation (4), it is possible to assume that it is equal to the difference between operative temperature and radiant surface temperature, multiplied by a fictitious total heat transfer coefficient h^*:

$$\frac{\dot{Q}_{cl}}{A} = \frac{\dot{Q}_{s}}{A} + DSL = h_{tot} \cdot (T_s - T_{op}) + DSL = h^* \cdot (T_s - T_{op}) \tag{11}$$

The h^* coefficient is thus equal to the total heat transfer coefficient of the radiant system plus the ratio between the DSL and the difference between the operative temperature and the radiant surface temperature. The second component of the algorithm represents the improvement of the heat transfer coefficient due to the solar radiation:

$$h^* = h_{tot} + (DSL/(T_s - T_{op})) \quad (12)$$

It is furthermore necessary to underline that when the HB and the RTS method are used in buildings equipped with radiant panels, the operative temperature must be used as the set point temperature. This is the reference temperature for thermal comfort analyses and for thermal load calculations [14, 16, 18, 19]. An iterative procedure must then be followed at the end of the calculations, in order to obtain cooling loads related to the expected operative temperature, and not to the air temperature, as is usually the case for all-air systems. In rooms with air velocities lower than 0.2 m s^{-1}, and differences between mean radiant temperature and air temperature less than 4 K, the operative temperature can be calculated as the adjusted air temperature [14, 15, 16]:

$$T_{op} \approx T_{ad} = \frac{T_a + T_{mr}}{2} \quad (13)$$

5. F values evaluation

As mentioned in section 3, the F value represents the ratio of solar heat gains directly removed by the radiant surface, without taking part in thermal absorption phenomena due to the thermal mass of the room . In order to evaluate its value, it is necessary to simulate the distribution of solar radiation inside the room.

Several approaches, with different levels of accuracy and precision, can be used to calculate the amount of solar radiation entering a room and its distribution over the internal surfaces.

A number of studies of the importance of the geometrical description of the room and the definition of the physical properties of the glass envelope and the internal surfaces have been reported in the literature [20, 21]. Inaccurate descriptions may significantly influence the output of any given simulation tool, leading to uncertain results.

The simplest models are based on an area-weighted distribution of the solar radiation, while the most accurate simulation tools exactly calculate the movement of "sun patches" across the interior surfaces of the room.

Solar radiation is composed of the diffuse component and the direct component: the most accurate approaches simulate separately the two components, while the simplest approaches generally assume that, after passing through a window, the direct radiation loses most of its directional character and may thus be simulated together with the diffuse radiation, as being uniformly distributed on all of the internal surfaces [21]. This simplification correctly models the effects of tinted and diffusing glass, but it may generate errors for all other kinds of glazing.

Simulations of solar radiation entering a room and calculations of the *F* value are mainly important in buildings with large surfaces of glazed envelopes (atria, galleries, skyscrapers, greenhouses...). In this kind of building, solar radiation transmitted through the envelope may reach very high levels, but not all of it remains inside the building. A part of the short wave radiation is in fact re-transmitted to the exterior through the glazed envelope, after some reflection back into the room. It has been underlined in the literature [20, 22] that simplified procedures do not take into account this part of the re-transmitted solar radiation. This is not a secondary issue, since it has been calculated that in a glazed space the amount of the transmitted solar radiation which is retained in the room may vary between 30% and 85% [23], and thus the amount of the radiation re-transmitted outside may vary between 25% and 70%.

The calculation of the *F* value depends therefore on several parameters: latitude, time of the year, time of the day, transmittance, absorptance and reflectance of the glass envelope, absorptance and reflectance of room surfaces, and room geometry [12].

In order to take into account all of these parameters, a commercial lighting analysis tool (Lightscape) was used to calculate the solar radiation distribution inside the room. It is an accurate tool that is able to simulate both direct and diffuse radiation through radiosity and ray-tracing algorithms, accounting even for that part of solar radiation that is re-transmitted to the outside of the building through the glazed envelope.

The tool has nevertheless some limitations: only visible radiation is considered by the software because it is not able to simulate infrared and ultraviolet radiation. No simulation tools were found in the literature to be able to simulate ultraviolet, infrared and visible radiation, reproducing the different effects of diffuse and direct radiation inside the room, without forcing the rendering engine or without substantial simplifications in the simulation approach (not even Radiance).
It is important to emphasise that, even without considering the ultraviolet and the infrared fractions of the solar spectrum, the lighting tool produces few computational errors, and results in only minor underestimations of the *F* value and the DSL.

The solar spectrum may be divided into four main parts (Table 2), which at sea level may be assumed to have the following values: ultraviolet between 250 and 350 nm (6%), visible between 350 and 750 nm (55%), near infrared between 750 and 2500 nm (39%), far infrared between 2500 and 4000 nm (~ 0%) [24]. The far infrared radiation at sea level is very close to 0%, because of absorption and screening in the atmosphere. Glazed envelopes used in the construction field are furthermore opaque to far infrared radiation, so it could not thus enter the room in any case. The far infrared radiation is not therefore considered in cooling load calculations.
Ultraviolet radiation represents 6% of solar radiation at sea level, but after the screening due to the glazed envelope it is reduced to 4% of the total transmitted radiation (Table 2) (special glazing used in museums may have a still greater screening effect). Ultraviolet radiation has moreover a limited effect on cooling load generation, as its energy does in fact mainly have its effect on the aging and degradation of surface layers of materials, and only a small fraction of it is liberated as heat [25]. The omission of ultraviolet radiation from cooling load calculations is thus justified.

A single-pane of clear glass was considered in the simulations and analyses in order to obtain the most widely applicable results. Considering the glass spectral transmission factors, it is possible to calculate the spectral distribution of the solar radiation entering the room and the fraction of it involved in DSL generation.

[place Table 2 approximately here]

If a radiant ceiling is considered, only the reflected solar radiation reaches the cooling surface. Since room surfaces are assumed to behave like gray bodies, with an emissivity and absorption factor to infrared radiation of 90% ($\varepsilon=\alpha=0{,}9$) [26], only 10% of the near infrared solar radiation is reflected toward the ceiling and is thus involved in the DSL generation. 90% of the incident infrared radiation is absorbed by the floor and the walls and re-emitted, after a time lag, into the room. It is therefore involved in the RL generation.

10% of the transmitted infrared radiation represents 3.6% of the total solar radiation inside the room. When a radiant ceiling is considered, this is the error made by the lighting analysis tool, because it does not simulate the infrared radiation. *F* values obtained by means of simulations refer to visible radiation only, hence the DSL is partly underestimated. On the other hand no errors due to simplification of the geometry of the room or to limitation of the directional character of the solar radiation are made.

It is thus interesting to notice that the DSL for a radiant ceiling is mainly produced by visible solar radiation, since the infrared radiation contribution is less then 4%. This observation is valid for all the kinds of glass used in the construction field, because the ratio between the transmission factor to infrared radiation and the transmission factor to visible radiation (τ_{IRn}/τ_v) reaches its maximum value for single-pane clear glass.

In a room equipped with a cooled radiant ceiling, therefore, light reflection factors of surfaces play a fundamental role in the DLS and in the RL generation. A highly reflective room will have a high DSL and a low RL. If room surfaces had higher reflection factors to infrared radiation (low-e), a bigger part of it would be involved in the DSL generation and less in the RL generation.

If a radiant floor is considered, a much larger amount of direct solar radiation could usually reach the cooling surface, depending on the furniture in the room. Under these conditions, the use of a lighting analysis tool produces higher underestimations of the DSL, because the infrared radiation is not simulated. In order to reduce the error, the *F* values obtained by means of the lighting analysis tool can be applied both to the visible and to the infrared solar radiation. The absorption factor for infrared radiation of a floor is 0.9, while its absorption factor for visible radiation is usually between 0.4 and 0.7. Using the

F values obtained through the light analysis tool, a fraction of the infrared radiation (between 20% and the 50%) is thus not accounted for in the DSL generation but in the RL generation.

In the case of a single-pane of clear glass the underestimation of the DSL is somewhere in the range between 7% and 18%, depending on the floor absorption factor for visible radiation.

6. Simulation results for a radiant ceiling

6.1 F values in empty rooms

When building with high SHG are considered, floor cooling systems are expected to show better performance than cooled ceiling systems, because they are subjected to high values of direct solar radiation. Ceiling cooling, on the contrary, are reached only by diffuse and reflected direct solar radiation. On the other hand, the calculations of the F values by means of the lighting analysis tool produce higher errors in the case of floor cooling than in the case of ceiling cooling, as described in Section 5. Moreover, cooled ceiling systems are currently much more commonly installed than floor cooling systems.

Due to the last two reasons, the lighting analysis tool was used to calculate the *F* ratio of the solar heat gains converted to DSL in rooms equipped with cooled radiant ceilings. The values obtained can be considered reliable enough for making cooling load calculations.

In previous studies [12, 27], *F* values were calculated with a simplified “reflection method” in which, through the use of angle factors between source and wall, the radiation was spread over the floor, the walls and the ceiling and then reflected between them again. Following all the reflections it was possible to evaluate the amount of the radiant flux removed by the radiant ceiling. Values obtained with the lighting analysis tool present the same trend obtained with the “reflection method”, but are more accurate [27].

F values were calculated for different room dimensions and for two different kinds of floor finishing: light (ρ_{fl}=0.6) and dark (ρ_{fl}=0.3). The height of the room remained constant in these simulations, but the area of the floor was changed.

All the room surfaces, with the exception of the floor, were considered to have a constant light reflection factor of 0.7, corresponding to light colour finishing.

In the following graphs, the ratio *L/h*, is used as a dimensionless index of the room geometry: it increases with each increase in the floor area. *L* is the side of the square floor while *h* is the height of the room. Nine different rooms of a fully glazed building were simulated, with an external wall that was completely glazed and without considering external obstructions. Simulations were developed for all the four cardinal orientations, changing the latitude at which the building was supposed to be located: Athens (38° N), Milan (45° N) and Copenhagen (56° N).

F values were calculated at 10 a.m., 12 a.m. and 4 p.m. for each latitude, orientation and floor surface. In order to have a useful design instrument, the data obtained were then summarised in two curves of average *F* values for the summer design day (21 July) and the winter design day (21 January) for each latitude. Buildings with a fully glazed envelope, usually office buildings, may in fact need cooling even during winter time, because the solar heat gains and the internal gains can be very high.

[place Figure 3 approximately here]

[place Figure 4 approximately here]

Both the orientation of the glazed façade and the reflectance of the floor have significant effects on the *F* values, as is evident in the simulation results (Figure 3 and Figure 4).

Passing from a dark floor (ρ_{fl}=0.3) to a light floor (ρ_{fl}=0.6), *F* values increased in all of the examples. When floor reflectance was increased, a higher amount of solar radiation was reflected toward the radiant ceiling, and therefore much more of it was converted to DSL.

The north exposure shows higher *F* values compared to the other orientations (Fig. 5, 6). In the case of a north orientation, only diffuse radiation enters the room, uniformly distributed on all the internal surfaces. Due to this homogeneous distribution, a larger amount of solar radiation reaches the ceiling surface.

In the case of south, west and east orientations, solar radiation entering the room has both diffuse and direct components. The direct radiation has slightly higher values than the diffuse radiation and it is mainly directed toward the floor and walls. Only a reduced fraction of the direct radiation reaches the ceiling by reflection. Under these conditions the *F* values related to the ceiling assume lower values compared to those for a northern orientation, because solar radiation is mainly concentrated on the floor and walls.

However, it is worth noting that, in the case of a northern orientation, solar radiation is at a very low level, due to the absence of the direct component. Rooms oriented towards the north will therefore never have high DSL values.

Another evident aspect is that *F* values increase with the *L/h* index. The ceiling surface area does in fact increase faster than the glazed surface area with L/h, in a square room and when, the height of the room is kept constant. For larger radiant surfaces, a larger proportion of solar radiation is converted to DSL. This analysis is correct for office rooms where height is constant and independent of the floor area. However, If room height were increased in proportion to the floor surface area, *F* values would probably remain constant.

A parameter which apparently has very little influence on the *F* values is the latitude. It influences mainly direct radiation, because sun height is different at different latitudes even on the same day and at the same hour of the year. Sun height in Copenhagen (lat. 56° N), is generally lower than in Milan (lat. 45° N) or Athens (lat. 38° N), both in winter and in summer. This means that a larger proportion of incident solar energy will enter a room, because solar beams are closer to being normal to the vertical glazed surface. Solar energy reaches its maximum level when it is perpendicular to the irradiated surface and the solar transmission factor of glass is a function of the incident angle between solar beams and glass surface.

For a northern orientation, *F* values are completely independent of the latitude, because the diffuse radiation distribution inside the room is independent of the solar angle. For other orientations the effect of latitude is more evident, but not so influential. Differences between the curves may be neglected. The most noticeable difference is for the south façade, because, in wintertime in Copenhagen, the solar angle is particularly low and solar beams are close to being normal to the glass surface.

Completely different conclusions would apply if the analysis had been for a radiant floor. Under these conditions, *F* values strongly depend on the orientation of the glazed façade and on the latitude. In the case of a cooled floor the generation of the DSL would not be so easily calculated as in the case of a cooled ceiling, because several more parameters influence it.

6.2 F values in rooms with furniture

The results summarised in the previous paragraph were calculated for empty rooms, while in real buildings rooms are not empty, but equipped with furniture items which modify their geometry and whose absorptance and reflectance factors for solar radiation may differ from those of the walls and floor. A further series of simulations was therefore carried out, in order to understand the influence of the furniture on the evaluation of the *F* values. Furniture items which have large horizontal areas (desks, couches, etc.), could change floor reflection toward ceiling. They will thus be the items with the greatest influence on effective *F* values.

As in the previous analyses, nine rooms with different floor area were simulated, inserting in them a number of desks related to the number of possible occupants (0.06 person per m^2). The desks were positioned at 0.8 m above the floor level, and, in order to obtain homogeneous results, the ratio between the total area of desks and the floor area was maintained constant (0.11-0.13).

The desks were assumed to be finished with a plastic layer with a reflection factor for light of 0.5, a value within the range of the reflection factors used for the floor (0.3 and 0.6). This choice made it possible to analyse both furniture with higher reflectance than the floor (when the floor reflectance was 0.3) and furniture with lower reflectance than the floor (when the floor reflectance was 0.6).

Since in previous analyses it has been demonstrated that the latitude has a negligible effect on *F* values, simulations of rooms equipped with furniture were carried out at the latitude of Milan only (45° N). These simplifications were necessary to control simulation times, because the inclusion of furniture increased the complexity of the room geometries, which greatly increased computation time.

When the floor had a reflection factor of 0.3, the presence of furniture did not change the *F* values: the total desk surface was probably not enough to make any noticeable difference.

The desks had higher reflectance compared than the the floor and thus they reflected more light towards the ceiling, but they also blocked a large amount of the light reflected by the floor (diffuse reflection), creating several reflections between their downward side and the floor. These two phenomena had probably the same values and therefore they compensated for each other.

When the floor had a reflection factor of 0.6 (Figure 5), the presence of furniture decreased the *F* values. In this case, the desks had a reflection factor for light that was lower than that of the floor, so they reflected less light compared to the floor. In addition, the desks blocked part of the light reflected by the floor, creating several reflections between their downward facing side and the floor. The two phenomena, when added together reduced the *F* values, though this effect was negligible in the context of cooling load calculations.

The presence of furniture, in conclusion, did not affect the *F* values. The results obtained for empty rooms can therefore be considered a very good approximation of what would have been obtained if the effects of furniture had been included.

However, this would not have been the case for a radiant floor, because desks substantially reduce the proportion of solar radiation directly incident on the floor surface, and thus the *F* values.

[place Figure 5 approximately here]

7. Calculation example

A calculation example will now be reported, in order to render the procedure proposed in this paper and its effect on cooling load calculations more comprehensible.

Cooling load calculations were performed for the summer design day (21 July), for an office room at the latitude of 45° N, with a net floor area of 56 m^2 and a net floor to ceiling height of 3 m. The room has only one external wall, which was completely glazed and oriented towards the south. The building was assumed to have a heavy structure with concrete walls and slabs, and it thus had a high thermal capacity. The set point temperature for the room was 26°C, while the minimum surface temperature of the radiant ceiling was set to 18.5°C in order to avoid condensation. The maximum ceiling cooling capacity (Q_s/A) was estimated to be 98 Watt per square meter.

The calculations were first carried out using the usual cooling load calculation procedure (Figure 6), and then using the new procedure proposed in section 3 (Figure 7). The room was considered to be equipped only with a radiant ceiling with an effective radiant surface equal to 85% of the net ceiling surface. It was therefore assumed that the whole cooling load was removed by the radiant system alone and that there was no load on the air system.

[place Figure 6 approximately here]

[place Figure 7 approximately here]

The graph reported in Figure 6 shows the cooling load profile only, because this is the usual output of the cooling load calculation methods that are currently in use. The graph reported in Figure 7 shows the two components of cooling load: the RL and the DSL, as required by the new procedure.

In Figure 6 the cooling load is compared to the radiant ceiling cooling capacity (Q_s/A). In order to make the two values comparable, the cooling load is expressed in Watts per square meter of actual radiant surface,. The calculations developed with the improved procedure show higher values of cooling loads because a sensible proportion of the solar heat gains is converted to DSL (dashed area in Figure 7), and thus it is not reduced by the effect of the thermal mass of the room. In the typical cooling load calculation procedure, all the solar heat gains are reduced and delayed by the effect of the thermal mass of the room. If typical cooling load calculations methods are used to evaluate cooling loads, a larger radiant surface would be required to remove the calculated peak load.

In Figure 7 the room load, expressed in Watts per square meter of actual radiant surface, is compared to the ceiling cooling capacity (Q_s/A). In this case the cooling capacity of the system always exceeds the RL. The RL in Figure 7 is lower than the cooling load in Figure 6, because some of the solar heat gains are converted to DSL by the action of the radiant surface. The dashed area between the CL curve and the RL curve represents the DSL, which is reported as an independent curve at the bottom of the graph.

The calculations carried out using the new proposed procedure, demonstrate that the useful radiant surface can be reduced, because the cooling capacity of the radiant ceiling is considerably higher than the actual RL. On the other hand, the whole cooling load on the cooling unit is higher than the one that is calculated with the traditional procedure, and this aspect has also to be taken into account for when selecting an appropriate cooling unit.

8. Conclusions

In this study a procedure for improving cooling load calculations in rooms equipped with radiant systems and exposed to large solar gains was proposed.

The most important element of the procedure is the Direct Solar Load, defined as the proportion of the solar heat gain that is directly converted to cooling load by the action of the cooling system. The *F* ratio of the solar heat gains involved in the generation of the DSL was calculated for different types of office room equipped with a radiant ceiling. The simulations were carried out with different glazed façade orientation, floor area, floor reflectance and latitude. The values obtained were summarised in graphs and can be used for cooling load calculations.

The proposed procedure provides a clear definition of the two components of the cooling load in rooms equipped with radiant systems: the Room Load and the Direct Solar Load, as demonstrated in a calculation example.

The procedure, when applied in cooling load calculations for rooms equipped with radiant ceilings, can be extended to rooms with radiant floors, with some corrections due to the differences between the two systems. In the case of a radiant ceiling, the DSL is considered to be removed by the transfer medium without any time lag. In the case of a radiant floor, on the other hand, the solar heat gains are assumed to be absorbed by the concrete or gypsum layer positioned between the floor covering and the water pipes. The thermal mass of this layer delays the conversion of the solar heat gain to DSL. It is therefore necessary to develop a model of the radiant floor that considers the thermal capacity of the concrete or gypsum layer before calculating the DSL.

The procedure proposed in this paper can be considered as a general approach to the analysis of solar radiation effects in the built environment. In this work the procedure was implemented with the most relevant calculation methods, namely the Heat Balance and Radiant Time Series methods.

Another possible way to take account of the effect of solar radiation on the built environment would be to evaluate the improvement of the radiant heat transfer coefficient on surfaces irradiated by solar radiation. This can be done in an environmental chamber, but the reproduction of the different boundary conditions due to the latitude and the time of the year and time of day would require a huge amount of time.

A fictitious total heat transfer coefficient (h^*), based on the evaluation of the DSL, was also proposed, in order to express the results obtained as a conventional coefficient.

The main advantage of the proposed procedure is nevertheless the possibility to analyse separately the different components of the cooling load, the RL and the DSL, and to evaluate their relative importance for the total cooling load. By means of this separation, the designer can easily understand the origin of the cooling load and choose the best solution to control it.

9. Nomenclature

A	Radiant surface	m^2
ARL	Air Room Load	W m^{-2}
c_p	Specific heat	J kg^{-1} K^{-1}
CHG	Convective Heat Gain	W m^{-2}
CL	Cooling Load	W m^{-2}
DSL	Direct solar load	W m^{-2}
F	F value (fraction of solar heat gain converted to DSL)	–
h_{tot}	Total heat transfer coefficient	W m^{-2} K^{-1}
h^*	Fictitious total heat transfer coefficient	W m^{-2} K^{-1}
HB	Heat balance	–
HER	Heat extraction rate	W m^{-2}
HG	Heat gain	W m^{-2}
$\dot{m}$	Mass flow rate	Kg s^{-1}
PRL	Panel Room Load	W m^{-2}
Q_{cl}	Total cooling capacity	W
Q_s	Surface cooling capacity	W
Q_{max}	Maximum cooling capacity	W
RHG	Radiant Heat Gain	W m^{-2}
RL	Room Load	W m^{-2}
RTS	Radiant Time Series	–
SHG	Solar Heat Gain	W m^{-2}
T_a	Air temperature	°C
T_{ad}	Adjusted air temperature	°C

T_{mr}	Mean radiant temperature	°C
T_{op}	Operative temperature	°C
T_s	Mean temperature of the radiant surface	°C
$T_{w,in}$	Supply water temperature	°C
$T_{w,out}$	Return water temperature	°C
α	Absorption factor	–
ε	Emissivity	–
ρ	Reflection factor	–
τ	Transmission factor	–

Subscript

θ	Time
fl	Floor

References

[1] M. Wigginton, Glass in architecture, Phaidon Press Ltd, London, 1996

[2] H.W. Krewinkel, Glass buildings, Birkhäuser, Basel, 1998

[3] W. Kessling, S. Holst, M. Schuler, Innovative design concept for the new Bangkok international airport, NBIA, in: Proceedings of Symposium on Improving Building Systems in Hot and Humid Climates, Richardson, 17-19 May 2004

[4] J.P. Holman, Heat transfer, McGrow-Hill, London, 1992, pp. 383–504

[5] F.P. Incropera, D.P DeWitt, T.L. Bergman, A.S. Lavine, Fundamentals of heat transfer, Wiley, New York, 2007, pp. 723–810

[6] J. Babiak, B.W. Olesen, D. Petras, Low temperature heating and high temperature cooling, Rehva, Brussels, 2007

[7] B.W. Olesen, Possibilities and limitations of radiant floor cooling, ASHRAE Transactions 103 (1) (1997) 42–48

[8] Fundamentals volume of ASHRAE Handbook, Chapter 18: Nonresidential cooling and heating load calculations, American Society of Heating, Refrigerating and Air-Conditioning Engineers, USA, 2009

[9] Fundamentals volume of ASHRAE Handbook, Chapter 28: Nonresidential cooling and heating load calculations, American Society of Heating, Refrigerating and Air-Conditioning Engineers, USA, 1997

[10] J.D. Spitler, Load Calculation Applications Manual, American Society of Heating, Refrigerating and Air-Conditioning Engineers, USA, 2009

[11] J.D. Splitter, D.E. Fisher, C.O. Pedersen, The radiant time series cooling load calculation procedure, ASHRAE Transactions: Symposia BN-97-4-4, 103 (2) (1997) 503–515

[12] S.P. Corgnati, Heat flows and air distribution in rooms cooled by radiant panels, Ph.D Thesis, Department of Energetics, Politecnico di Torino, 2002

[13] S.P. Corgnati, G.V. Fracastoro, M. Perino, Un metodo di calcolo per il dimensionamento dei soffitti radianti per il raffrescamento estivo, CDA 7 (2000) (Italian text)

[14] ISO 7730: 2005 - Ergonomics of the thermal environment - Analytical determination and interpretation of thermal comfort using calculation of the PMV and PPD indices and local thermal comfort criteria

[15] ISO 7726: 1998 - Ergonomics of the thermal environment - Instruments for measuring physical quantities

[16] ASHRAE Standard 55: 2004 Thermal Environmental Conditions for Human Occupancy

[17] EN 15377-1: 2008 - Heating systems in buildings - Design of embedded water based surface heating and cooling systems - Part 1: Determination of the design heating and cooling capacity

[18] F. Causone, S.P. Corgnati, M. Filippi, B.W. Olesen, Experimental evaluation of heat transfer coefficients between radiant ceiling and room, Energy and Buildings 41 (6) (2009) 622–628

[19] EN 12831: 2003 - Heating systems in buildings - Method for calculation of the design heat load

[20] M. Wall, Distribution of solar radiation in glazed spaces and adjacent buildings. A comparison of simulation programs, Energy and Buildings 26 (1997) 129–135

[21] K. Chatziangelidis, D. Bouris, Calculation of the distribution of incoming solar radiation in enclosures, Applied Thermal Engineering 29 (2009) 1096–1105

[22] J.M. Mottard, A. Fissore, Thermal simulation of an attached sunspace and its experimental validation, Solar Energy 81 (2007) 305–315

[23] M. Wall, A design tool for glazed space. Part I: description, ASHRAE Transactions 101 (1995) 1261–1271

[24] ASTM, ASTM G173-03 Tables: Extraterrestrial Spectrum, Terrestrial Global 37 deg South Facing Tilt & Direct Normal + Circumsolar

[25] http://hyperphysics.phy-astr.gsu.edu/hbase/hframe.html

[26] M.F. Modest, Thermal Radiation, in: A. Bejan (Ed.), A.D. Kraus (Ed.), Heat Transfer Handbook, Wiley, Hoboken, 2003, pp. 573–633

[27] F. Causone, S.P. Corgnati, M. Filippi, Calculation method for summer cooling with radiant panels, in: Proceedings of CLIMA 2007 WellBeing Indoors, Helsinki, 10-14 June 2007

Figure 1. Cooling load generation scheme redrawn from: Fundamentals volume of ASHRAE Handbook, Chapter 18: Nonresidential cooling and heating load calculations, © American Society of Heating, Refrigerating and Air-Conditioning Engineers

Figure2. The cooling load generation scheme for radiant systems, taking into account the DSL

Figure3. F values for north, south, east and west façade orientation in rooms equipped with a radiant ceiling and a dark floor. Simulations were run in Athens (lat. 38° N), Milan (lat. 45° N) and Copenhagen (lat. 56° N)

Figure4. F values for north, south, east and west façade orientation in rooms equipped with a radiant ceiling and a light floor. Simulations were run in Athens (lat. 38° N), Milan (lat. 45° N) and Copenhagen (lat. 56° N)

Figure5. F values for north, south, east and west façade orientation in rooms equipped with a radiant ceiling and furniture. Simulations were run in Milan (lat. 45° N)

Figure6. Comparison between the cooling load (or room load) and the ceiling cooling capacity, with usual cooling load procedures

Figure7. Comparison between then room load and the ceiling cooling capacity, with the improved procedure

Table 1. Heat transfer coefficients, temperature limits and maximum cooling capacity of radiant systems (the minimum surface temperature of ceiling and wall was calculated as the dew-point temperature in the case of room air temperature equal to 26°C and relative humidity equal to 58%, which are typical conditions in the cooling season when the RH is controlled)

Table 2. Solar radiation distribution outside the room, used glass transmission factors and transmitted solar radiation distribution

Table 1

	h_{tot} [W m^{-2} K^{-1}]	$T_{s\,min}$ [°C]	$T_{op\,max}$ [°C]	Q_{max} [W m^{-2}]
Ceiling	13	17	26	117
Wall	8	17	26	72
Floor	7	19	26	49

Table 2

SOLAR RADIATION	Total	UV	Visible	Near IR	Far IR
Outside radiation distribution	100%	6%	55%	39%	0%
Glass transmission factors (τ)	0.81	0.56	0.89	0.74	0.00
Transmitted radiation distribution	81%	3%	49%	29%	0%
Radiation distribution inside the room	100%	4%	60%	36%	0%

Figure 1

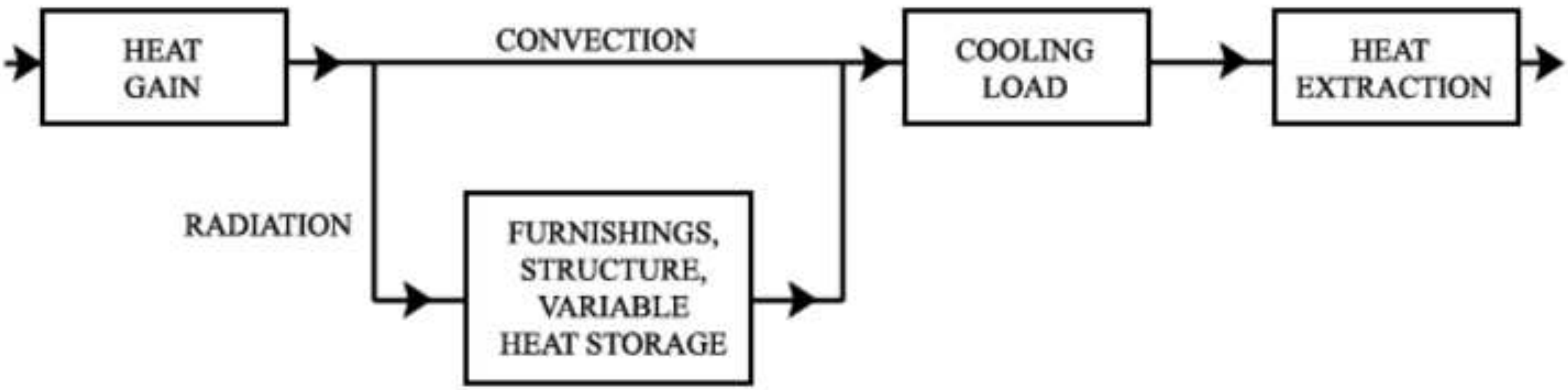

HEAT
GAIN
CONVECTION
COOLING
LOAD
HEAT
EXTRACTION
RADIATION
FURNISHINGS,
STRUCTURE,
VARIABLE
HEAT STORAGE

Figure 2

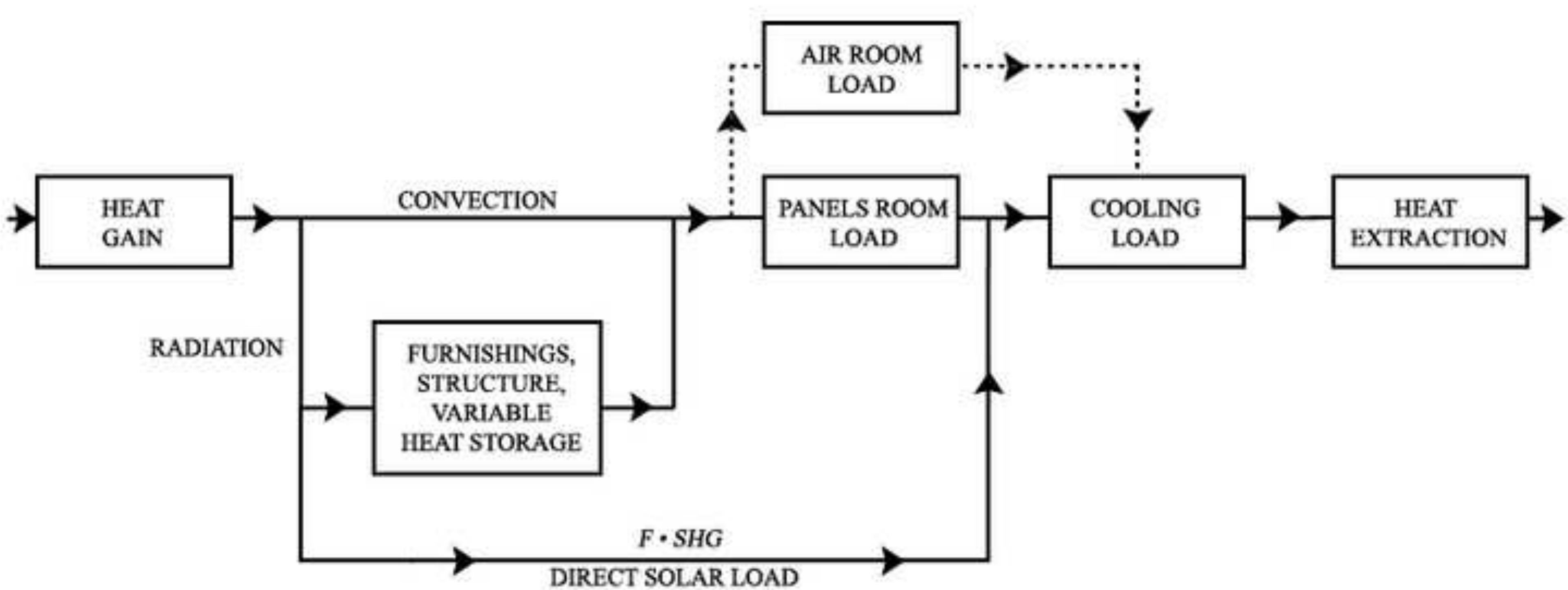
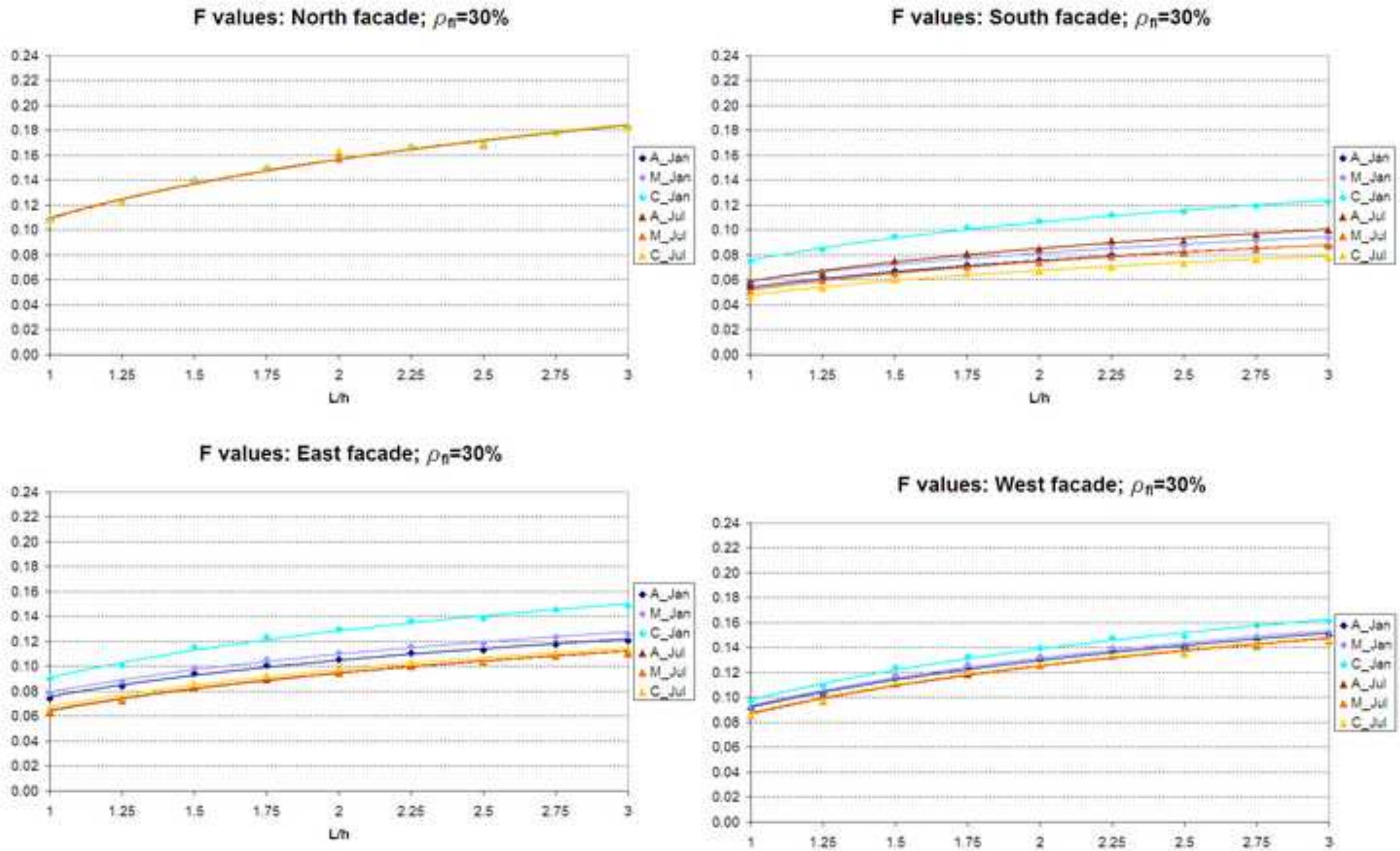

HEAT GAIN
CONVECTION
RADIATION
FURNISHINGS, STRUCTURE, VARIABLE HEAT STORAGE
AIR ROOM LOAD
PANELS ROOM LOAD
COOLING LOAD
HEAT EXTRACTION
F • SHG
DIRECT SOLAR LOAD

Figure 3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F values: South facade; ρfl=30%
F values: East facade; ρfl=30%
F values: West facade; ρfl=30%
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0.20
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1.5
1.75
2
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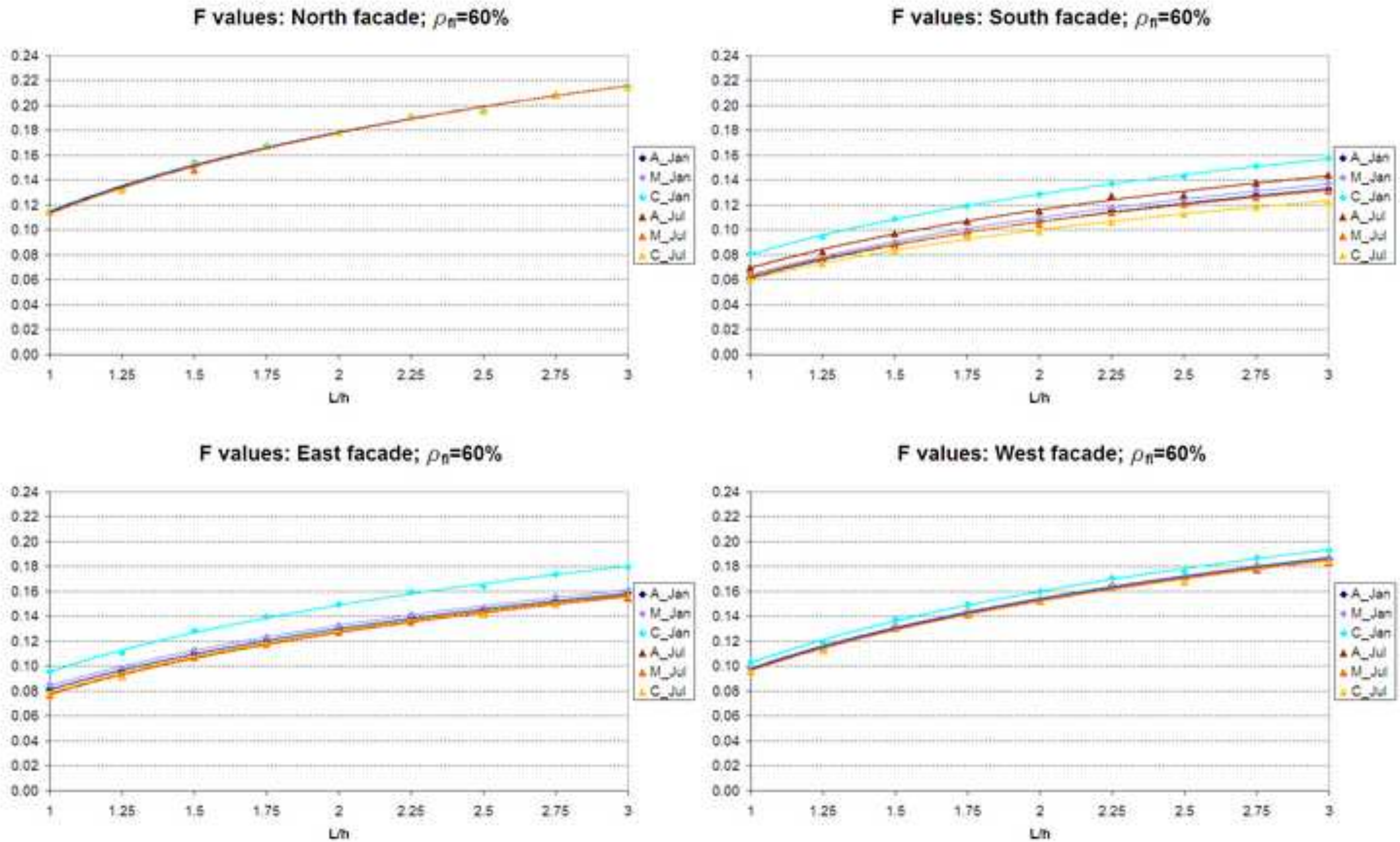
M_Jan
C_Jan
A_Jul
M_Jul
C_Jul

Figure 4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F values: South facade; ρ_{fl}=60%
F values: East facade; ρ_{fl}=60%
F values: West facade; ρ_{fl}=60%
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Figure 5

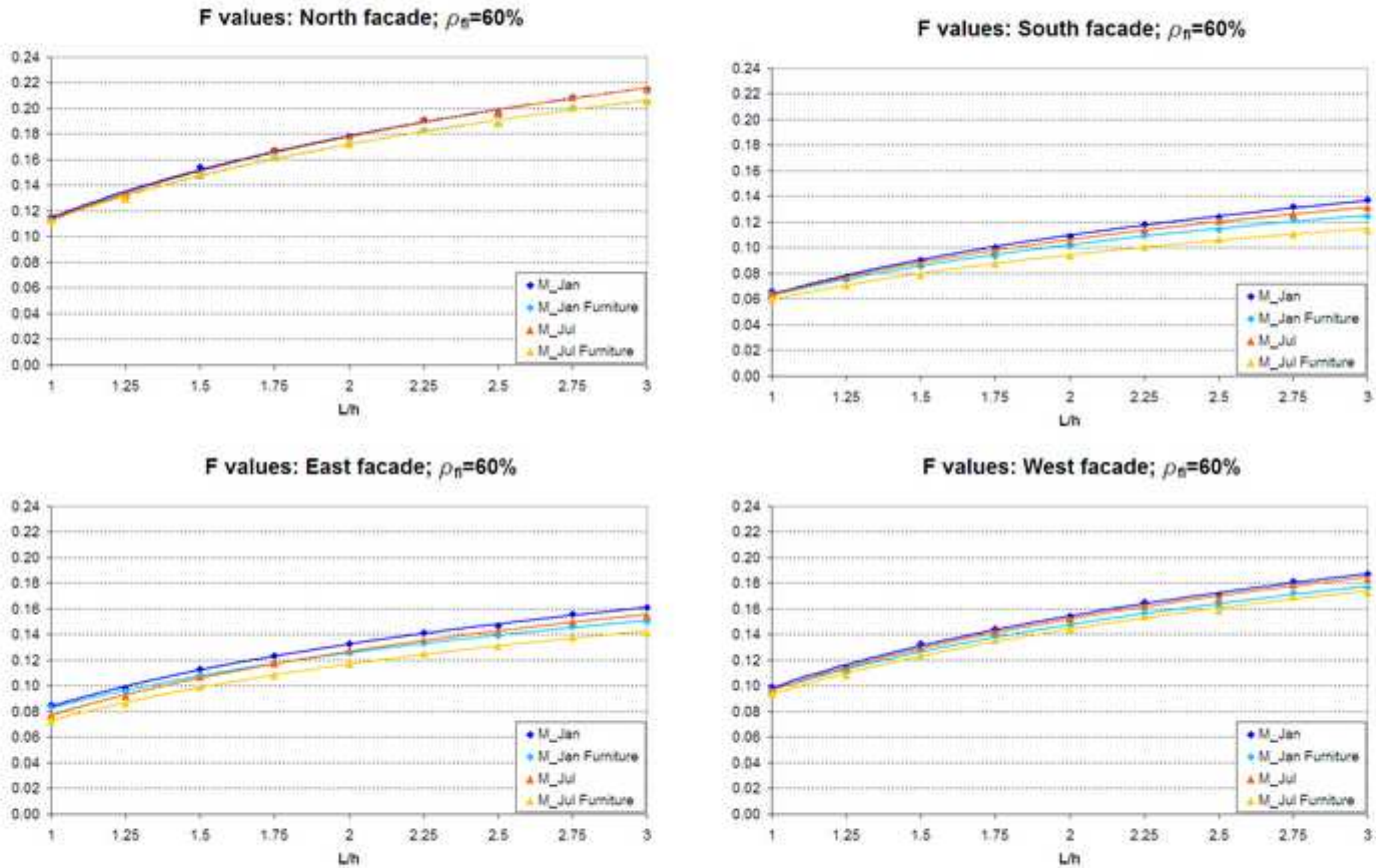
F values: North facade; ρ_{fl}=60%
F values: South facade; ρ_{fl}=60%
F values: East facade; ρ_{fl}=60%
F values: West facade; ρ_{fl}=60%
L/h
M_Jan
M_Jan Furniture
M_Jul
M_Jul Furniture
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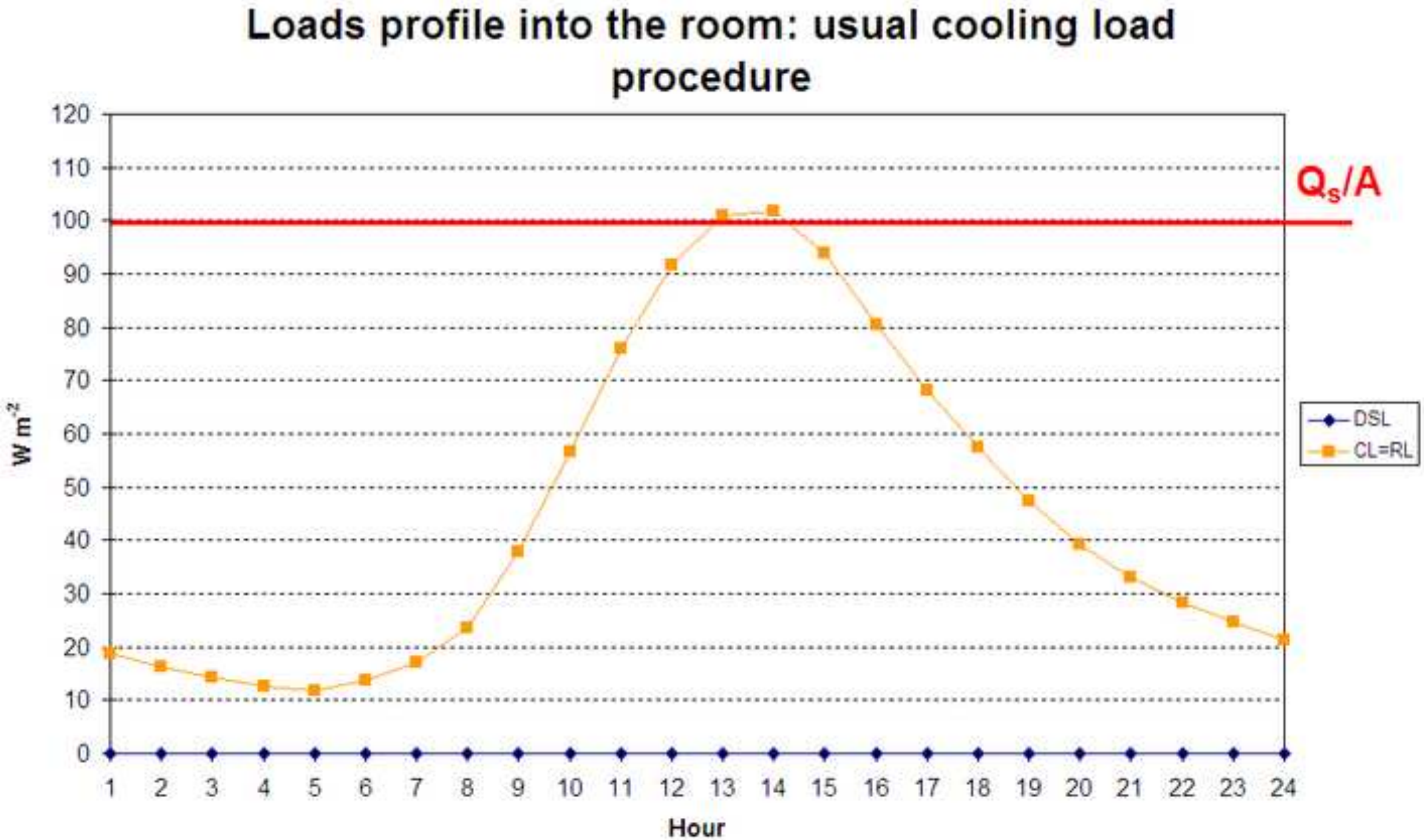
Figure 6


Loads profile into the room: usual cooling load procedure
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W m-2
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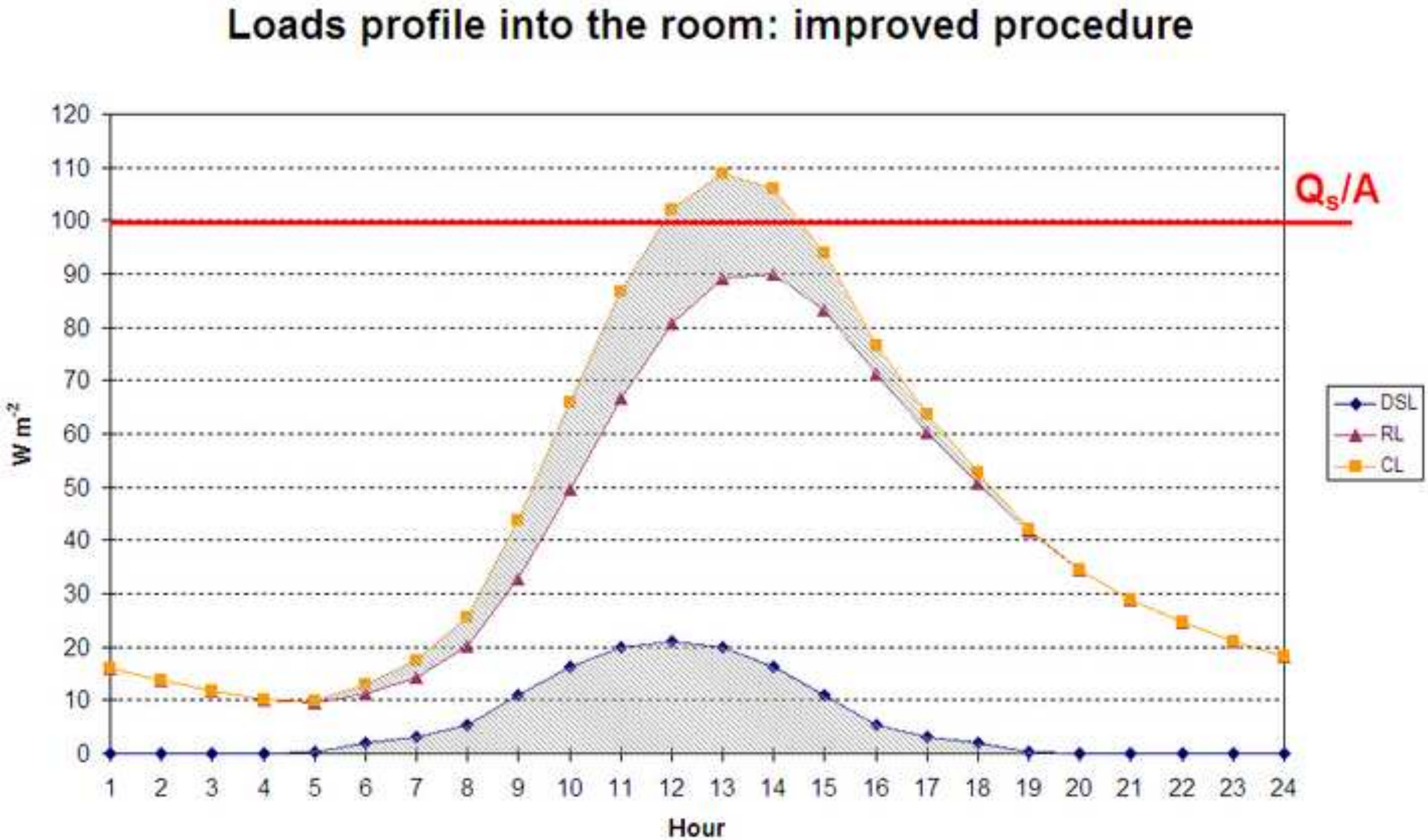
Figure 7



Loads profile into the room: improved procedure
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Hour
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DSL
RL
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