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New method for the design of radiant floor cooling systems with solar radiation

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Abstract

Impacts of solar shortwave radiation are not taken into account in the standardized design methods in the current radiant system design guidelines. Therefore, the current methods are not applicable for cases where incident solar is significant. The goals of this study are to: 1) use dynamic simulation tools to investigate the impacts of solar radiation on floor cooling capacity, and 2) develop a new simplified method to calculate radiant floor cooling capacity when direct solar radiation is present. We used EnergyPlus to assess the impacts of solar for different design conditions. The simulation results showed that the actual cooling capacities are in average 1.44 times higher than the values calculated with the ISO 11855 method, and 1.2 times higher than the ASHRAE method. A simplified regression model is developed to improve the predictability of ISO methods. The new model calculates the increased capacity as a function of the zone transmitted solar and the characteristic temperature difference between the hydronic loop and room operative temperature.

Keywords – Radiant floor cooling; air system sizing; cooling load and capacity; solar heat gain; radiant design standards

1   Introduction

Natural lighting and physical connection to our environment are integral to the design of functional residential and commercial buildings. Glass, creating a continuum of space between the outdoors and the living space, is a distinct and pervasive building material in modern architecture. Glass admits solar radiation, and it is usually a challenge for the traditional air-based HVAC systems to maintain thermal comfort in spaces with significant solar radiation. In contrast, water-based radiant floor cooling systems are considered as especially effective for conditioning those spaces. For standard applications of radiant floor cooling systems, the rule of thumb is the cooling capacity can be up to 50 W/m^2 [1, 2]. For applications that the sun illuminates the cooling surface, literatures show that the floor capacity can increase significantly, reaching 80-100 W/m^2 [3-5]. For this reason, floor cooling is increasingly designed in spaces with large glazed surfaces, such as atriums, airports, and entrance halls [6, 7].

The cooling capacity of radiant floor systems is limited primarily due to a relatively small convective heat transfer coefficient between the floor and air, risk of condensation and concern about discomfort caused by low floor temperature, radiant asymmetry, large

vertical air stratification and draft. Researcher have studied the cases with solar radiation. Odyjas [4] showed that cooling capacity of radiant systems largely depends on the type of cooling load occurring in the room, but they did not explain the phenomenon from a fundamental heat transfer perspective. Their numerical simulation results showed that with a minimum floor temperature at 20 °C, the cooling capacity of the simulated floor system was 14-22 W/m^2 for pure convective load, 19-30 W/m^2 for mixed radiant/convective load, and 150-226 W/m^2 for 100% direct shortwave solar load at steady-state conditions. The last case, however, would never occur in practice because it requires a supply water temperature of 4.5 °C, which is too low for radiant applications due to condensation and comfort concern.

Simmonds [8] looked at longwave and shortwave radiation separately in his calculation of total cooling capacity, and explained that the enhanced cooling capacity was due to solar radiation reaching the floor. However, in the calculations, he assumed that the amount of solar radiation absorbed by the floor was a known value. This is however hardly true in design practices. The calculations of absorbed solar radiation require not only knowing the transmitted solar through windows, but also floor surface temperature, surface material absorptivity, and room thermal conditions and properties. While the total transmitted solar can be obtained with well-defined calculation method or readily available computer programs, the latter factors are either uncertain during design or are variables that cannot be predicted accurately without using a heat-transfer based computer simulation tool. For designers, it would be useful to have a simplified method to estimate the amount of solar radiation absorbed by the radiant floor surfaces.

Causone [9] used a lighting simulation tool to quantify the ratio of the amount of solar radiation that is directly absorbed by radiant ceilings to the total solar heat gain to the space. He studied cases with different aspect ratios, window orientations, surface material absorptivity, and locations. However, his study focused on radiant ceiling applications. Ceilings normally receive a much lower direct solar radiation as compared to floors. In addition, there are limitations in his method: 1) the lighting tool can only figure out the "solar patches", i.e. the amount of radiation that arrives at the surface instead of the absorbed radiation; 2) the calculations were steady-state analysis. At one point in the paper, the authors proposed that a fictitious heat transfer coefficient maybe introduced to characterize the improvement of heat transfer due to solar radiation. The heat transfer coefficient could be a function of directly absorbed solar radiation and the temperature difference between floor surface and room operative temperature. This is a promising idea. However, they did not provide a method to estimate this heat transfer coefficient.

For designers, accurate prediction of radiant system cooling capacity is critical both for designing of the radiant system but also for the associated air system. As radiant systems provide only sensible cooling, air systems work in hybrid mode to provide ventilation, dehumidification and supplemental cooling if needed. The size of the air system depends on radiant system capacity, and underestimation of radiant system capacity can lead to oversizing the air system.

Therefore, the goals of this the paper are to: 1) verify and quantify the dynamic impacts of solar radiation on radiant system capacity, 2) develop a simplified method to calculate radiant floor cooling capacity , and 3) investigate the implications for sizing the associated air systems. In the first part of this paper, we theoretically analyze the

limitations in existing design methods, and explain why they fail to take into account solar shortwave radiation and radiation from internal load in the calculation process. Even though the internal radiative heat gain is also not properly considered, this paper focuses on the solar radiation because the impacts of internal load were evaluated to be less significant (5- 10% for studied simulation cases).

The following sections present the methods and results that quantify the dynamic impacts of solar radiation on radiant system capacity. Based on the simulation results, a simplified method is developed for predicting the radiant system capacity when there is solar load. The last section of the paper demonstrates how the designers may use the new method to size the associated air system.

2 Background

There are two primary types of water-based radiant systems: 1) suspended metal ceiling panels with copper tubing attached to the top surface (radiant ceiling panel, RCP); 2) prefabricated or installed-in-place systems consisting of embedded tubing in radiant layers (embedded surface system, ESS). Depending on pipe position and radiant layer constructions, ISO 11855, the standard titled *Building environment design -- Design, dimensioning, installation and control of embedded radiant heating and cooling systems*, further classifies the embedded system into seven types, from type A to G (Table 2 of part 2 of the standard) [10]. Radiant floor system can be one type of the ESS systems.

Designing a radiant system generally involves the determination of the following parameters: system specifications (tube diameter, spacing, surface finishing, insulation, total tube length, etc.), and design operating conditions (design surface temperature, flow rate, supply temperature, and pressure drop). The goal is to make sure the system capacity can satisfy the heating/cooling demand.

According to ISO 11855, design Cooling Capacity is defined as the thermal output at a cooling surface at design conditions. In practice, there is no standardized way to obtain the radiant system capacity. A survey of leading designers indicated that the design approaches include the direct use of numbers from manufacturer's product catalog, the use of designers' in-house calculation tools, which are developed mostly for steady state analysis, or conducting finite element or finite difference analysis which allows the evaluation of system dynamic performance. Occasionally, the most experienced designers use whole building simulation software, such as EnergyPlus or TRNSYS to assist in the design [11]. Dynamic simulation tools allow an evaluation of dynamic impacts of the thermal environment on system performance and an assessment of control sequences. For the panel systems, steady-state analysis method might be adequate because of the relatively small delay of the heat exchange between the environment and hydronic loop. For the embedded systems with thermal delay, dynamic solution is desirable for improved prediction accuracy. However, it is not always practical due to reasons such as lack of available skills or financial/time constraints. Simplified methods that are developed based on steady state calculations are still the most widely adopted practice.

There are two standardized ways to represent system capacity. The first one directly correlates surface heat flux to the temperature differences between room operative temperature and radiant surface(s), and this method explicitly requires the knowledge of the surface heat transfer coefficients. The second approach represents system capacity with a lumped thermal resistance and a mean temperature difference between the cooling medium and the space. In the design guidelines, there is no information about the application or differences between the two approaches. However, it appears that designers like to use the first approach as a quick way to check the feasibility of radiant system and as a basis for detailed design of system configuration and design operating conditions [7, 12]. The second approach uses product performance data from manufacturers to relate the thermal output to system configuration and waterside operation.

Regardless of the representation of system capacity, understanding the heat transfer process at radiant surface is critical to this design analysis. By investigating how the current calculation methods characterize this process, we can understand the limitations of the current methods.

2.1 Heat transfer at the radiant surface

Fig.1 shows the heat transfer balance at the radiant surface. If one defines the control volume as the inside face of the cooling slab, with a positive sign meaning heat is being transferred into the control volume and negative indicates heat leaving the control volume, the heat balance equation can be written as follows:

$$q''_{conv} + q''_{lw\_surf} + q''_{lw\_int} + q''_{sw\_sol} + q''_{sw\_int} + q''_{cond} = 0 \qquad (1)$$

In which,

q''_{conv} = Convection heat transfer at the exposed face of the cooling surface(s), W/m^2

$q''_{lw,surf}$ = Net longwave radiation flux to radiant active surface from other surfaces, W/m^2

$q''_{lw,int}$ = longwave radiant exchange flux from the internal load, W/m^2

$q''_{sw,sol}$ = Transmitted solar radiation flux absorbed at the surface, W/m^2

q''_{cond} = Conduction heat transfer at the exposed face of the cooling surface(s), W/m^2

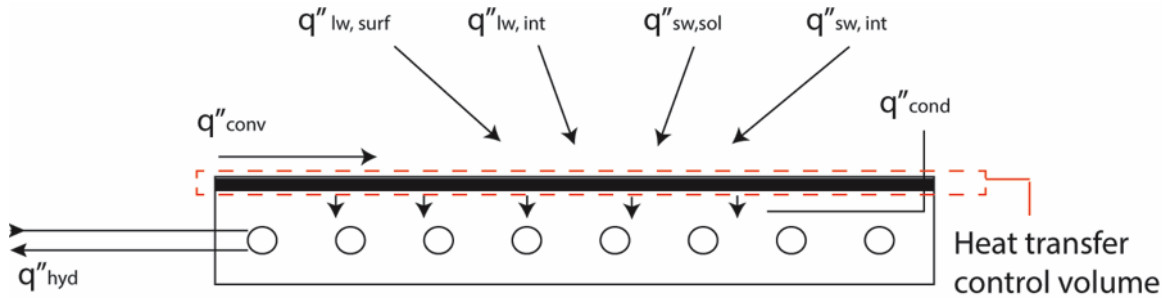q''_{hyd} = Specific heat rate removed by the hydronic loop, W/m^2

Figure 1: Heat transfer balance at the radiant surface and hydronic loop

The amount of heat removed by the activated cooling surface (cooling capacity) is a combination of convection and radiation, and can be theoretically calculated as below:

$$q_{surf}^{"} = -q_{cond}^{"} = q_{conv}^{"} + q_{lw,surf}^{"} + q_{lw,int}^{"} + q_{sw,sol}^{"} + q_{sw,int}^{"} \tag{2}$$

The last four terms on the right hand side, $q_{lw,surf}^{"}$, $q_{sw,sol}^{"}$, $q_{sw\_int}^{"}$, and $q_{lw,int}^{"}$ are radiation components, and can be summed up to obtain the total radiation capacity at the surface, $q_{rad}^{"}$. Radiation heat transfer is a significant contributor to the total heat transfer, usually higher than 50%.

Surface heat transfer can be calculated either separately for convection and radiation, as is the case in Chapter 6 of *ASHRAE Handbook, HVAC Systems and Equipment* [13] and recommended by researchers [14] or can be characterized using a combined heat transfer coefficient as is recommended by ISO 11855(2012). Usually, scientists are interested in the first approach, while designers prefer to use a combined heat transfer coefficient [15].

2.1.1 *Convective heat transfer*

The surface convective heat transfer can be calculated as:

$$q_{conv}^{"} = h_c \cdot (T_a - T_s) \tag{3}$$

Where, h_c is the convective heat transfer coefficient, W/m$^{2.}$K, T_a is the zone air temperature, °C, and T_s is the radiant surface temperature, °C.

Natural convection is usually assumed for radiant cooling systems. Some selected algorithms for the calculation of the convective heat transfer coefficient of cooled floor/heated ceiling are summarized in Table 1 below. In the last column of the table, the ranges of the heat transfer coefficient from each method are reported. The selection of the algorithm can have a significant impact on the calculation results. However, it is beyond the scope of this paper to verify if one is better than the others. In this study we selected the algorithm developed by Walton [16], which is implemented in EnergyPlus with the "TARP" heat transfer calculation option.

Table 1: Floor cooling convective heat transfer coefficient correlations

Correlation	Source	Range [2]
$h_c = 0.87 \, (T_a - T_s)^{0.25}$	ASHRAE recommendation: A. Kollmar [24]	0.8-1.46
$h_c = 1.0$	Olesen, B [25]	1.0
$h_c = 0.948$	Walton [23]: EnergyPlus *Simple* option	0.948
$h_c = 0.7589 \, (T_a - T_s)^{1/3}$	Walton [23]: EnergyPlus *TARP*[1] option	0.75-1.51
$h_c = \dfrac{0.704}{D^{0.601}} \, (T_a - T_s)^{0.133}$	Awbi [26]	0.46-0.61

1. We adopted this algorithm for the EnergyPlus simulations
2. The heat transfer coefficents are calculated with $T_a - T_s$ varied from $1 - 8$ °C

2.1.2 Radiation heat transfer

In the ASHRAE Handbook, *HVAC Systems and Equipment* [13]*,* the radiation heat flux for surface heating and cooling systems is approximately,

$$q^{"}_{lw\_surf} = 5 \times 10^{-8} \cdot [(T_s + 273.15)^4 - (AUST + 273.15)^4] \tag{4}$$

Where, $AUST$ is area-weighted temperature of all indoor surfaces of walls, ceiling, floor, window, doors, etc. (excluding active cooling surfaces), ℃.

A linear radiant heat transfer coefficient can be also defined to express the radiant heat exchange between a specific surface and all the other surfaces in the room.

$$q^{"}_{lw\_surf} = h_{rad} \cdot (T_s - T_{ref}) \tag{5}$$

Where, h_{rad} is a linear radiant heat transfer coefficient, and it can be considered constant when surface temperatures are within 15-30 °C. Most literatures suggest that the value of h_{rad} to be5.5 W/m^2·K [2, 13], but it may increase in spaces with large glazing area [17]. T_{ref} is a reference temperature, and there is no standardized definition for it yet. In most literatures, it is either the $AUST$ or the operative temperature at a reference point in the room.

Both equations (4) and (5) calculate longwave radiation heat transfer between the radiant cooling surface and its enclosure surfaces. The lack of consideration of the incident solar radiation and other internal heat gain on cooled surfaces was observed in calculation models documented in the standards and in several models developed by researchers [17-19].

2.1.3 Total heat transfer

To size HVAC systems, and especially radiant systems, a combined heat transfer coefficient is convenient. This means that a total heat transfer coefficient, depending on

system type (floor/wall/ceiling and heating/cooling), is used to calculate the surface heat flux.

$$q^{"} = h_t \cdot |T_s - T_{ref}| \tag{6}$$

where, h_t is the combined convection and radiation heat transfer coefficient, and typical values are reported in Table 2. Again, only convection and longwave radiation between surfaces are included in the total heat transfer calculation.

Table 2: Summary of floor cooling total heat transfer coefficients

h_t	Reference temperature (T_{ref})	Source
7	Operative temperature	ISO 11855 [22]
8.29	AUST	Chapter 6 ASHRAE HVAC Systems and Equipment [20]
7.5	Operative temperature	Olesen [25]

2.2 Cooling capacity estimation method

As mentioned, most manufacturers reported radiant system capacity in relation to a lumped thermal resistance, K, and a mean temperature difference between the cooling medium and the space, ΔT_h. The mathematical formula is in Eq (7).

$$q^{"} = K \cdot \Delta T_h^n \tag{7}$$

Where, n is a constant, and equal to 1 for the embedded systems according to ISO 11855. Both K and n are to be determined. The parameters that are included in the resistance, K, are surface heat transfer coefficients, resistance of the radiant conductive layers, resistance between water loop and pipe, etc. According to Zhang [17], surface heat transfer coefficient is the most important parameter among all other thermal resistances. Definition of ΔT_h depends on system types and applications. For embedded floor system, ΔT_h is calculated as:

$$\Delta T_h = \frac{(T_{wi} - T_{wo})}{ln\,[\,(T_{wi} - T_{opt})/\,(T_{wo} - T_{opt}\,)]} \tag{8}$$

Where, T_{wi}, and T_{wo} is the supply and return water temperature respectively, °C, and T_{opt} is the room operative temperature, °C.

Methods for the determination of the thermal resistance, K, also depend on system types . In general, they can be classified into calculation and testing methods. Testing methods involve evaluating radiant system performance by conducting laboratory tests following a codified procedure, and calculation methods involve using analytical or numerical methods based on known physical properties of the systems. For radiant ceiling panels, only testing methods are permitted (ASHRAE 138 [20] or EN 14240 [21] for cooling and

EN 14037 [22] for heating), while for embedded systems, both calculation and testing methods are allowed. However, the testing method for embedded systems, the "two plate" method described in EN 1264, was not widely adopted by manufacturers, and the calculation methods are on the other hand widely used [23].

Details about the calculation methods for different system types can be found in ISO 11855, which is consistent with the contents in EN 15377 [24] and EN 1264 [25]. Note that the ASHRAE Handbook: HVAC Systems and Equipment also provides a calculation method that correlates system capacity with characteristic panel thermal resistance, design radiant surface temperature, AUST (area-weighted indoor surface temperatures), water supply temperatures, etc. However, the methods specified in the ISO standard are the most widely adopted, and thus we report system cooling capacity following the ISO standard.

For the radiant floor cooling system that we studied in this paper, K can be calculated using the following steps. First, calculate the $K_{H,floor}$ for floor heating systems using the correlations for system type A-D [19].

$$K_{H,floor} = B \left(\prod_i \alpha_{i,}^{m} \right) \tag{9}$$

Here, B is a system dependent coefficient, $\prod_i \alpha_{i,}^{m}$ is the power product linking the parameters that can be calculated with the information of the floor construction using methods documented in Appendix A of ISO 11855 part 2. To obtain K for floor cooling system, a conversion has to be conducted:

$$K = K_H\left(\Delta R_{\alpha,}, R_{\lambda,B}\right) = \frac{K_{H,floor}}{1 + \dfrac{\Delta R_{\alpha,} + R_{\lambda,B}}{R_{\lambda,B}} \left(\dfrac{K_{H,floor}}{K^*_{H,floor}} - 1\right)} \tag{10}$$

Here, $\Delta R_\alpha = 1/\alpha - 1/10.8$, and α is the total heat transfer coefficient depending on surface type (floor/celling/wall) and application (heating/cooling), $R_{\lambda,B}$ is the thermal resistance of surface covering, $K^*_{H,floor}$ is the resistance when $R_{\lambda,B} = 0.15$.

In summary, a review of the existing design methods shows that impacts of shortwave solar radiation are not taken into account. As mentioned, even though the internal radiative heat gain is also not properly considered, this paper focused on the impact of solar load.

3 Methodology

3.1   Modeling approach

To verify and quantify the dynamic impacts of solar radiation on radiant system capacity, we uses Energy Plus v7.2 to model a single zone conditioned by a radiant floor system. We design a full matrix of simulation run to evaluate a wide range of design options.

EnergyPlus v7.2, a widely used whole-building energy simulation tool [26, 27], is selected because it employs the fundamental heat balance method for zone thermal

modeling and has been validated against experimental measurements and through comparative testing with BESTest suite [28]. More importantly, it is one of a limited number of tools that is capable of accurately simulating radiant system performance [29, 30]. EnergyPlus's Engineering Reference provides the details of solar simulation algorithm employed in the tool. . It is a "ray tracing" method that tracks the paths of the beam and diffuse solar coming through the fenestration systems. The heat balance algorithms can adequately capture the dynamics of the absorbed solar load and the operating conditions of the cold floors.

The single zone model used for this study is developed based on ASHRAE Standard 140 [31]. Only radiant systems are modelled when the study is focused on the fundamentals on cooling capacity. Radiant system design parameters were based on RADTEST [32]. The parameters and their variations we have investigated include: shortwave absorptivity of floor surface material (0.4/0.8), shading options (Interior blinds/No shading), window-to-wall ratio (40/55/70/95), various toping thickness of radiant slab system (5/7/10 cm), zone orientation (east/west/south), building aspect ratio (1.3/2) and supply water temperature (12/15/18 °C). Schematic of the radiant floor is shown in Fig.2. Interior blinds were controlled to be active when the incident solar on window is higher than 100 W/m^2. The total number of simulation runs was 864.

The test case model is a rectangular single zone with no interior partitions. For the cases with aspect ratio at 1.3, the model dimensions are 8 m wide × 6 m long × 2.7 m high, and for the aspect ratio 2, the width increased to 12 m. The base building construction is based on case 900 (heavyweight) [31], except that the floor construction has been modified so that water tubes can be embedded in the concrete layer when radiant floor systems are simulated. There is a window with an overall U-value of 2.721 W/m^2·K and SHGC of 0.788 and the total area of the window varies for each window to wall ratio. The TMY3 Denver weather data was used. We do not include internal load and infiltration in the model. During each simulation run, the radiant system was available 24 hours a day, and was controlled to maintain a zone operative temperature at 24 °C.

For each simulation run, we investigate the heat transfer process at the radiant surface. In this paper, the simulated radiant floor cooling capacity is the Surface Inside Face Conduction Heat Transfer Rate at the radiant surface. This is an EnergyPlus output. We compared the simulated radiant system cooling capacity to the capacity calculated using the ISO method. The goal is to numerically evaluate the applicability of the ISO method to cases with solar load.
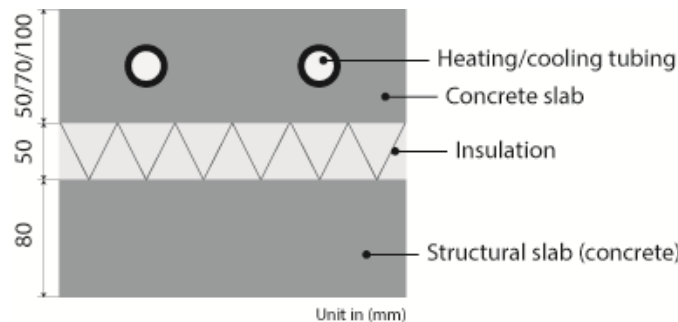

Figure 2: schematic of the radiant floor systems simulated

3.2 Statistical analysis method

Throughout this study, we need to gauge quantitatively how two sets of data compared to each other. The coefficient of variation of the root mean squared error (CVRMSE), was selected for this purpose, and it can be calculated using the following formulae:

$$CVRMSE = \frac{[\sum_{i=1}^{n}(y_i - \check{y}_i)^2/(n-p)]^{1/2}}{\bar{y}} \times 100\% \tag{11}$$

The Root Mean Square Error (RMSE) is a frequently used measure of the difference between values predicted by a model and the values actually observed from the environment that is being modelled. These individual differences are also called residuals, and the RMSE serves to aggregate them into a single measure of predictive power. CVRMSE is the non-dimensional form of the RMSE.

4 Impact of solar heat gain on cooling capacity

We have conducted a total of 864 runs for the 99.6 % cooling design day. Simulation results confirmed that radiation is the dominant heat transfer mechanism on the chilled floor, and is the main interest to this study, so we concentrated our analysis on radiation heat transfer rate. Fig.3 is a plot of the 24-hour radiation heat flux, $q_{rad}^{''}$, at the floor surface, including the total radiation and its breakdown into longwave and shortwave radiation. For each hour, a box-plot displays the range of floor surface radiant heat flux for the 864 runs. Because no internal load has been simulated, shortwave radiation consists of pure solar load, and the longwave radiation includes envelope load and part of the solar load that has been absorbed by building mass and reemitted toward the radiant floor.
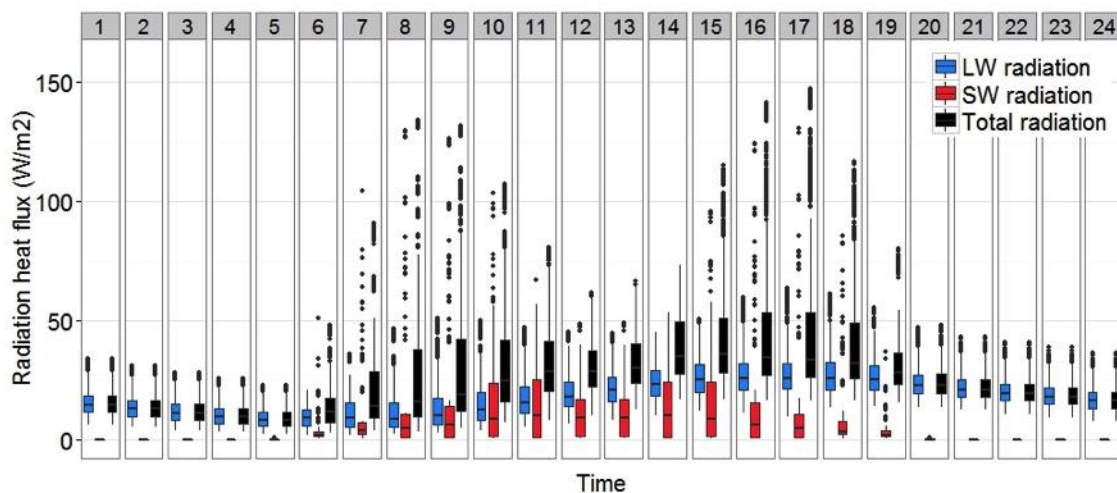

Figure 3: Cooling design day floor radiation heat flux breakdown for the 864 simulation runs

4.1 Surface radiation heat flux

In this paper we have hypothesized that, with the presence of solar heat gain, the actual heat transfer rate at a radiant floor cooling system will be higher than the values calculated using either ISO or ASHRAE methods. The enhancement is the result of absorption of shortwave radiation at the cooling surface. To numerically demonstrate this, we defined two indices, total radiation ratio (TRR) and longwave radiation ratio (LWRR), as listed in Table 3. Total radiation ratio is defined for a direct comparison of EnergyPlus simulated total radiation heat flux at the radiant surface and the value calculated using ISO method (Equation 5) and ASHRAE method (Equation 4).. The longwave radiation ratio (LWRR) is defined as the ratio of simulated longwave radiation heat flux at the cooling surface to the radiation calculated using ISO and ASHRAE approaches. Fig. 4 shows the boxplots of TRR and LWRR of the 864 runs.

Table 3. Parameters analyzed

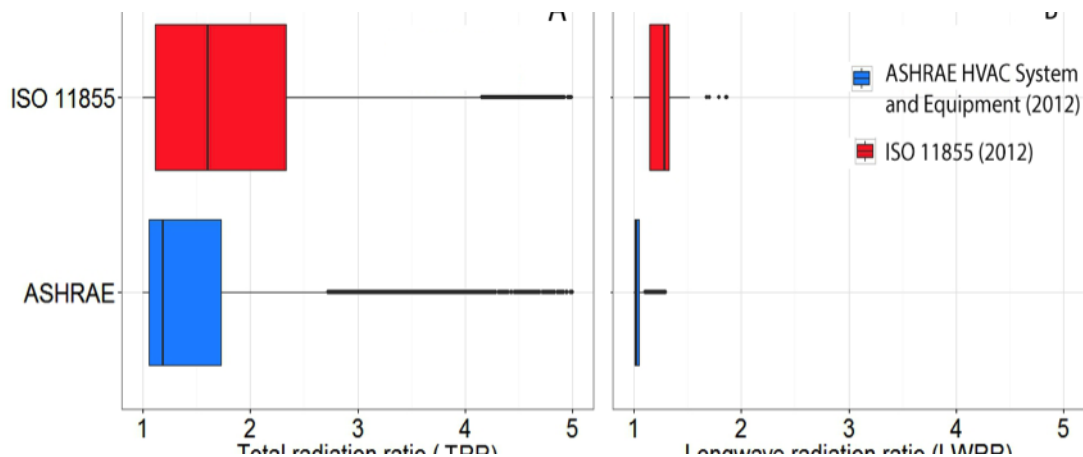|	Compared to ISO	Compared to ASHRAE
Total radiation ratio (TRR)	$RR_{ISO} = \dfrac{q''_{rad\_EPlus}}{q''_{rad,ISO}}$ $q''_{rad,ISO} = 5.5 \cdot (T_s - T_{opt})$	$RR_{ASHRAE} = \dfrac{q''_{rad\_EPlus}}{q''_{rad,ASHRAE}}$ $q''_{rad,ASHRAE} = \text{Eq. (6)}$
Longwave radiation ratio (LWRR)	$LWRR_{ISO} = \dfrac{q''_{rad\_lw\_EPlus}}{q''_{rad,ISO}}$	$LWRR_{ASHRAE} = \dfrac{q''_{rad\_lw\_EPlus}}{q''_{rad,ASHRAE}}$

Figure 4: Comparison of radiation heat flux at radiant surface between Energyplus and ISO/ASHRAE method using box-plot of the 864 simulation runs: (A) total radiation ratio; (B) longwave radiation ratio

Fig.4 (A) shows the range of total radiation ratio (RR). If the ratio is higher than 1, the simulated total radiation heat flux is higher than calculated values. The right plot (B) shows LWRR. The scale of x-axis is limited to 5 to achieve better resolution for the

interquartile range. The box plot shows that LWRRs are close to 1, indicating the longwave radiation heat flux calculated using the standard methods is almost the same as that simulated, thus the increase of total radiation can be attributed to shortwave radiation. Fig.4 demonstrates that the median of the simulated cooling capacity is 1.44 times higher than the ISO 11855 method and the interquatile range (IQR) is from 1.06 to 2.44, and when compared to ASHRAE, the simulated cooling capacity is at median 1.2 times higher and the IQR of RRASHRAE is 1.06 to 1.86. Cooling capacity estimated using ASHRAE method coincides better with the results from EnergyPlus.

4.2 Cooling capacity

Fig.5 compares the simulated and calculated (Eq.7-10) cooling capacity for two cases. The first case (A) is with interior blind, i.e. without shortwave solar hitting the radiant floor, and the second case (B) is without any shade, i.e. with shortwave solar radiation. Each dot in the figure represents one hourly simulation result. In Fig.5 (A) the calculated capacity curve can predict cooling capacity reasonably well, with CVRMSE at 25.0%. At a standard design temperature difference of $\Delta T_h = 10$ °C, cooling capacity of the simulated radiant floor systems range from 35.6– 44.0 W/m^2, which is consistent with the numbers reported in the literature in standard applications. Fig.5 (B) shows the comparison for cases with direct solar, and system capacity can increase up to 130-140 W/m^2 at a standard system $\Delta T = 10$ °C. The CVRMSE was 54.1%.
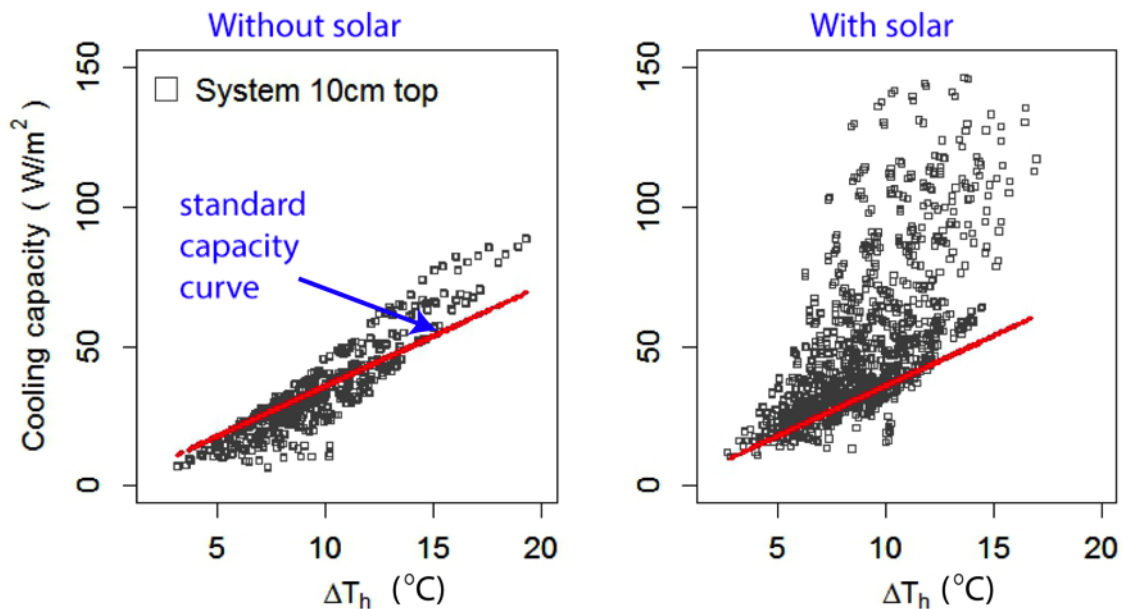

Figure 5: Comparison of simulated cooling capacity with cooling capacity calculated using ISO -11855 method (Eq.7-10) for system Type A-D: (A) with interior blind, i.e. no shortwave solar radiation; (B) without shade, i.e. with shortwave solar radiation.

Even though the actual cooling capacity of a radiant floor is higher than the standard recommended values when there is solar radiation, it is essential to minimize solar radiation admitted into the building for a successful radiant system design. Excessive solar radiation causes high overall cooling energy consumptions. However, for those

cases such as atria, airports, and perimeter areas when large area of glass is desirable, the impacts of solar need to be properly considered to achieve optimal sizing of radiant floor cooling systems. In addition, the sizing of their associated air system depends on an accurate prediction of floor capacity. We will discuss this issue in more detail in Section 5.

5   Simplified model for radiant floor cooling capacity estimation

To improve the predictability of the current cooling capacity estimation method for cases with direct solar heat gain, the following new equation is proposed:

$$q^{"} = K \cdot \Delta T_h + q^{"}_{sw\_sol} \qquad (12)$$

$q^{"}_{sw\_sol}$ is the cooling capacity enhancement caused by absorbing of direct solar radiation. Eq. (13) is the proposed correlation to calculate $q^{"}_{sw\_sol}$ and the derivation of this correlation is described in the Appendix A.

$$q^{"}_{sw\_sol} = 1.993 \cdot (q^{"}_{sol,win})^{0.7476} - 5.038 \cdot (q^{"}_{sol,win} \cdot \Delta T_h)^{0.2793} \qquad (13)$$

Where, $q^{"}_{sol,win}$= total transmitted solar heat flux into the space, W/m^2; ΔT_h can be calculated by Eq. (8). This model has an R^2 adj = 0.82.

The $q^{"}_{sol,win}$ includes both the beam and diffuse solar heat flux that are transmitted through the windows into the space. If there is no complex fenestration system, i.e. shading devices or light shelf, it can be obtained by multiplying the incident solar heat flux on the window and the window's total solar transmittance. If complex fenestration systems are installed, more advanced simulation tools may need to be used for the calculation.

Theoretically speaking solar transmittance is a variable that is directionally and spectrally (wavelength) dependent. The calculation of total transmitted solar radiation with high accuracy requires detailed information of window's optical properties, which could be obtained using tools, such as WINDOW [33]. However, in practice, there are usually simplifications. For example, in DOE-2 and with the *simple glazing* option in EnergyPlus, the total solar transmittance at normal incident is used. For designers who only have information about window's solar heat gain coefficient (SHGC) and U values, the total solar transmittance at normal incident could be calculated using the regression models developed by researchers at Lawrence Berkeley National Lab [34] .

Note that the heat transfer process with incident solar radiation is highly dynamic; the proposed calculation method can be used for rough estimation of the system design capacity and is not intended to replace detailed dynamic simulation.

6 Implications for sizing of associated air system

As mentioned before, radiant systems are typically designed to operate in hybrid with an air system. Designers have to size the air system to provide supplemental cooling if the space total cooling load exceed the maximum capacity of the radiant system. Thus, the desired capacity of the air system is directly related to the radiant system cooling capacity. Underestimation of radiant system capacity when solar load is significant can lead to oversizing of the air system. In this section, we conducted a new series of simulations of the hybrid radiant and air system, and the goals are to validate the applicability of the proposed approach to the estimation of radiant system cooling capacity and demonstrate how to use it for sizing of the associated air system.

The same single zone model described in Section 3.1 was used but with an idealized air system simulated. During each simulation run, both radiant and air systems were available 24 hours for the cooling design day. While the radiant system was controlled to maintain a zone operative temperature of 24 °C, the air system was controlled to maintain the setpoint at 26 °C. Design parameters investigated were the same as listed in Section 3.1, except that only cases without any shade were investigated (432 runs).

Fig.6 shows the zone operative temperatures of all runs for the design day, and they were maintained below 26 °C with supplemental cooling capability of the air system.
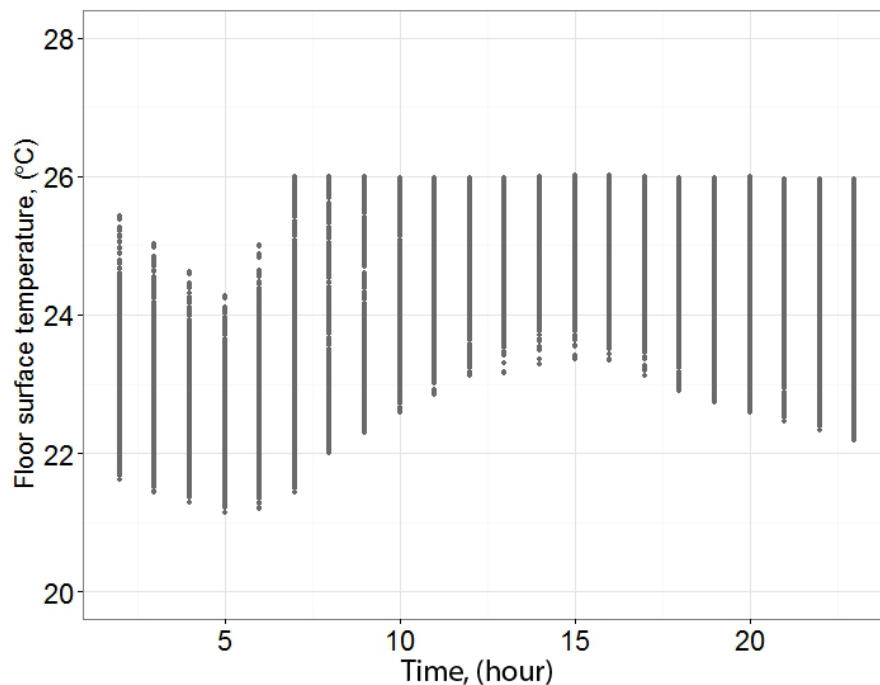

Figure 6: Zone operative temperature ranges during all simulation runs

Fig.7 compares the predicted and EnergyPlus simulated floor capacity. The dots will fall on the red line if they are the same. The CVRMSE was 22.1 %. If the ISO method was used, the CVRMSE would be 58.4 %.
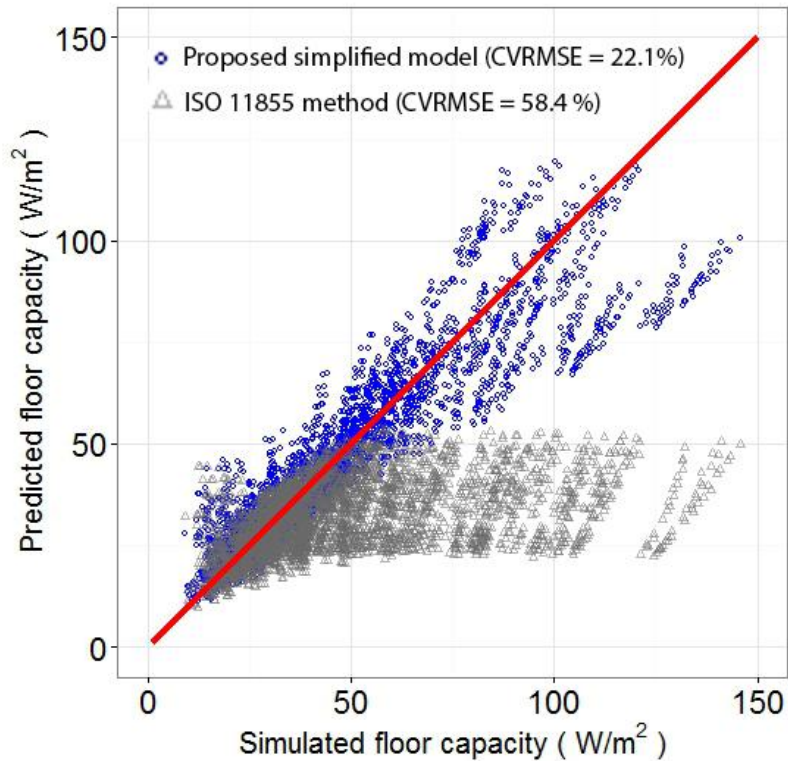
Figure 7: Comparison of simulated capacity and calculated capacity using Eq.(8, and 12-13)

Fig.8 shows an example of the sizing process if designers estimated the cooling capacity of the radiant floor system using the ISO 11855 method. In this case, the peak total cooling load was estimated to be 135 W/m^2, and if ISO 11855 method was used, they have to size the air system to be able to handle 93 W/m^2. However, the actual capacity of the floor as calculated with EnergyPlus simulation was 109 W/m^2, and thus the air system only needs to be sized to 26 W/m^2. The ability to accurately estimate the cooling capacity of the radiant system is the key to solving this issue.

Now, we use the same example to demonstrate how to use the new model for the sizing of air system, see Fig.9:

Step 1: for the summer cooling design day, calculate the total cooling load and the peak load. For this example it is 135 W/m^2.

Step 2: calculate the radiant floor available capacity use Eq (8, 12-13), assuming the designer has information about the radiant system in consideration, transmitted solar radiation, design supply and return water temperature and design room operative temperature.

Step 3: hourly air system capacity is obtained by subtracting the predicted floor system capacity from total cooling load. The required air system design capacity is the peak value, here in this case 31 W/m^2.
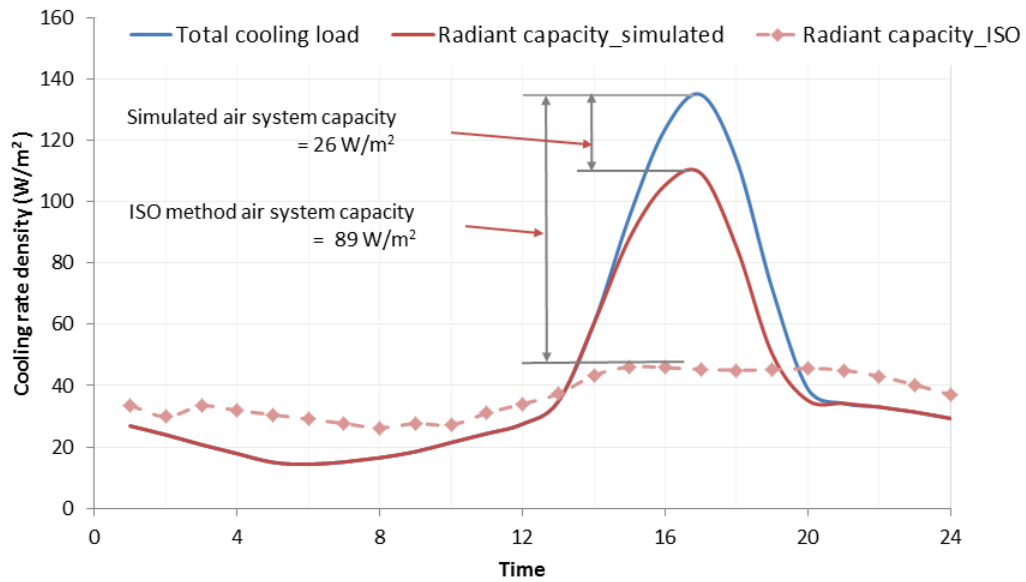
Figure 8: Example of how enhanced cooling capacity impact sizing of air system
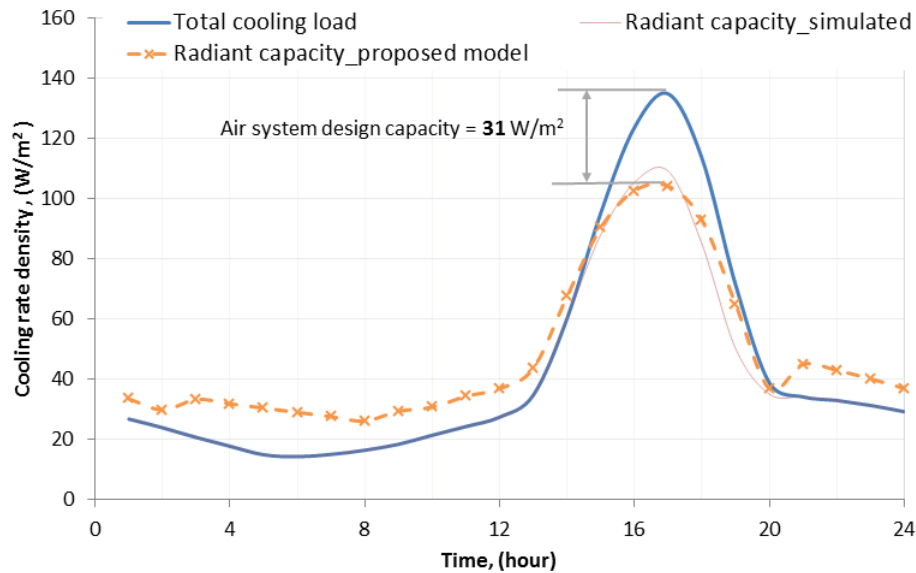

Figure 9: Example of using the proposed method for sizing of air system

7 Conclusions

We provided evidences that existing radiant cooling capacity estimation methods are theoretically insufficient when the system is exposed to radiation. This is because only convective heat transfer and radiant heat exchange with warmer building surfaces (longwave radiation), not solar radiation or radiation from lighting, are considered in the current calculation methods.

The parametric runs results showed that for cases with direct solar, and system capacity can increase up to 130-140 W/m^2 at a standard system $\Delta T = 10$ °C. The median of the actual surface radiation heat flux is 1.44 times higher than the values calculated with ISO 11855 method, and it is 1.2 times higher compared to ASHRAE. The ASHRAE method, which calculates surface radiation and convection heat flux separately, has better predictability than the ISO method, which calculates surface heat flux using a combined heat transfer coefficient.

A new equation is proposed to estimate system capacity enhancement due to direct solar absorption. The new simplified model calculates the enhanced capacity as a function of window transmitted solar radiation and a mean temperature differences between the hydronic loop and room operative temperature. The new regression model has an adjusted $R^2_{adj} = 0.82$. The new simplified model enables the designers to more accurately size the associated air system and therefore avoid oversizing the air system by a significant amount.

8   Acknowledgments

Appendix A: Derivation of correlation for calculating q''_{sw_sol}

Grouped scatterplots were used for initial evaluation of the correlation of q''_{sw_sol} with other design and operational parameters (Fig. A-1). It can be seen that there is a direct linear relationship between q''_{sw_sol} and the total window transmitted solar heat flux ($q_{sol,win}$), a parameter that can be easily obtained using energy simulation tools. Besides total window transmitted solar heat flux ($q_{sol,win}$), the following parameters are also selected to be included in the initial investigation: mean temperature difference as calculated by Eq (8) (ΔT_h), design supply water temperature (CWS), radiant floor surface material shortwave absorptivity (Abs), window-to-wall ratio (WWR), orientation (OR), aspect ratio (AP), and radiant toping slab resistance (K).
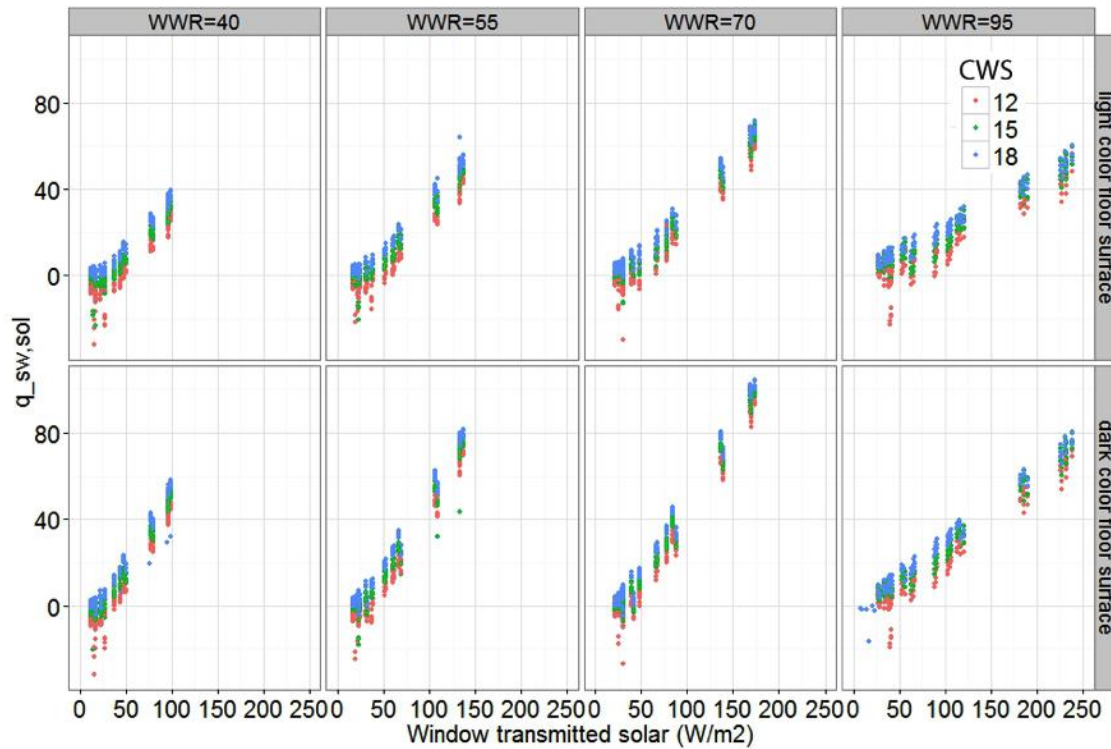
Figure A-1: Scatter plot of $q_{sw\_sol}^{"}$ vs. windows transmitted solar

As a starting point, a multi-variable linear model of $q_{sw,sol}$ that has included all parameters was derived, and the adjusted R^2 was 0.85. To evaluate the significance of each parameter in the model, ANOVA tests were conducted for models with less independent variable. Based on these tests, orientation and aspect ratio were dropped from further evaluation. Further reduction of independent variables was tested (Table A-1). However, the plot of residual over fitted value showed non-linear relationship, and thus transformation of independent variables was explored.

Table A-1: Summary of multi-variable linear models for prediction of $q_{sw\_sol}^{"}$

Model	Linear models for $q_{sw\_sol}^{"}$	Adjusted R^2
1.	$q_{sw\_sol}^{"} = -19.232 + 0.3870 * q_{sol,win}^{"} + 16.936 * Abs$	0.804
2.	$q_{sw\_sol}^{"} = -30.7853 + 0.38815 * q_{sol,win}^{"} + 1.4379 * CWS$	0.805
3.	$q_{sw\_sol}^{"} = 0.095 + 0.4205 * q_{sol,win}^{"} - 1.2947 * \Delta T_h$	0.795

Some non-linear models tested were shown in Table A-2. Models were generated using the curve fitting tool in Matlab 2013. Instead of general linear least square method, robust

regression method was applied. The latter is one type of the weighted regression methods, which gives less weight to points that behave as outliers but are not excluded for model development due to lack of compelling reasons. The independent variables are kept at two to reduce the complexity of the model, as increasing the number of independent variables does not significantly improve the model quality. Cross validation was applied for selection of model type. Cross validation is a model validation technique for assessing how the results of a statistical analysis will generalize to an independent data set. It is mainly used in settings where the goal is prediction, and one wants to estimate how accurately a predictive model will perform in practice. The procedure for cross validation involves assigning all data randomly to a number of subset. We uses 10 folds here. Each subset is removed, in turn, while the remaining data is used to re-fit the regression model and to predict at the deleted observations. The overall Mean Square, which is a corrected measure of prediction error averaged across all folds.

Table A-2 provides the test results. We can see Model #4 has slightly better prediction capability

Table A-2: Summary of models for prediction of $q_{sw\_sol}^{"}$

Model	Nonlinear models for $q_{sw\_sol}^{"}$	Adjusted R^2	Overall MS
1.	$q_{sw\_sol}^{"} = -7.681 - 2.469 \cdot \Delta T_h + 0.8409 \cdot q_{sol,win}^{"} + 0.1881 \cdot$ $* (\Delta T_h)^2 - 0.03855 \cdot (q_{sol,win}^{"} \cdot \Delta T_h)$	0.8322	96.6
2.	$q_{sw\_sol}^{"} = -16.32 + 0.5382 \cdot \Delta T_h + 0.7472 \cdot q_{sol,win}^{"} - 0.0308$ $\cdot (q_{sol,win}^{"} \cdot \Delta T_h)$	0.8301	96.5
3	$q_{sw\_sol}^{"} = 0.7504 \cdot q_{sol,win}^{"} - 0.02935 \cdot (q_{sol,win}^{"} \cdot \Delta T_h) - 18.09$ $* (\Delta T_h)^{-0.1289}$	0.8296	98.1
4	$q_{sw\_sol}^{"} = 1.993 \cdot (q_{sol,win}^{"})^{0.7476} - 5.038 \cdot (q_{sol,win}^{"} \cdot \Delta T)^{0.2793}$	0.8202	95.9

List of Symbols

D	Hydronic diameter, m
ESS	Embedded surface systems
h_c	Convective heat transfer coefficient, W/m$^2\cdot$K
h_{rad}	Linear radiant heat transfer coefficient, W/m$^2\cdot$K
h_t	Combined convection and radiation heat transfer coefficient, $W/m^2\cdot K$
K	Lumped thermal resistance between hydronic loop and space, W/m^2.K

$K_{H,\,floor}$ Lumped thermal resistance between hydronic loop and space for floor heating, W/m^2·K

n Constant

$q^{"}$ Specific system capacity, W/m2

$q^{"}_{cond}$ Conduction heat transfer at the exposed face of the cooling surface(s), W/m^2

$q^{"}_{conv}$ Convection heat transfer at the exposed face of the cooling surface(s), W/m^2

$q^{"}_{rad}$ Radiation heat transfer at the exposed face of the cooling surface(s), W/m^2

$q^{"}_{rad\_lw}$ Longwave radiation heat transfer at the exposed face of the cooling surface(s), W/m^2

$q^{"}_{lw\_surf}$ Net longwave radiation flux to radiant active surface from other surfaces, W/m^2

$q^{"}_{lw\_int}$ Longwave radiant exchange flux from internal load, W/m^2

$q^{"}_{sw\_sol}$ Transmitted solar radiation flux absorbed at surface, W/m^2

$q^{"}_{sw\_int}$ Net shortwave radiation flux to surface from internal load (lights) , W/m^2

$q^{"}_{sol,win}$ Total transmitted solar heat flux into the space. W/m^2

RCP   Radiant ceiling panels

T_s Radiant surface temperature, ℃.

T_{ref} Reference temperature, ℃.

T_a Zone air temperature, ℃

ΔT Reference temperature difference, °C

T_{wi} Supply temperature of cooling medium, °C

T_{wo} Return temperature of cooling medium, °C

T_{opt} Design space operative temperature, °C

T_{opt} Operative temperature at a reference point in the room, ℃.

Subscript

ASHRAE   Variables calculated using the ASHRAE method

Eplus   Variables calculated using EnergyPlus simulation

ISO   Variables calculated using the ISO method

Abbreviations

Abs	Absorptivity of material
AUST	Area-weighted temperature of all indoor surfaces of walls, ceiling, floor, window, doors, etc. (excluding active cooling surfaces), ℃.
ASHRAE	American society of heating refrigeranting and air-conditioning engineers
CVRMSE	Coefficient of variation of the root mean squared error
CWS	Cold water supply
LWRR	The longwave radiation ratio, is defined as the ratio of simulated longwave radiation heat flux at the cooling surface to the radiation calculated using either ISO and ASHRAE methods
OR	Orientation
RR	The radiation ratio, defined as the ratio of simulated radiation heat flux at the cooling surface to the radiation calculated using either ISO and ASHRAE methods
SHGC	Solar heat gain coefficient
TABS	Thermally activated building systems
WWR	Window to wall ratio

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