Comparison of the Thermal Performance of Radiative and Convective Terminals: A Conceptual Approach

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ABSTRACT: During the last decades, the use of radiant terminals for heating or cooling buildings has rapidly increased and the activation of different construction elements has been tested: floor, ceiling or walls. Radiant systems are usually used instead of air conditioning systems: it is therefore of interest to compare the energy efficiency of the two types of terminals for heating and cooling buildings. Convective terminals (i.e. air conditioning systems) have been widely used in buildings, but the level of comfort is not always acceptable due to high air velocity. On the other hand radiant terminals can provide a better indoor climate, and be more energy efficient because they can make use of low-grade sources. The output of this conceptual approach is a better understanding of the advantages and drawbacks of the two technologies under different conditions. The analysis has been performed by simulating the energy consumption of an office room, located in Denmark. Different outdoor conditions have been tested, in order to compare their performance during the winter season and the summer season. Different types of activated surface have also been simulated. The results of this analysis show that the efficiency of convective terminals is highly dependent on the ventilation losses of the building. Radiant terminals are less sensitive to this parameter.

Keywords: Radiant terminal, Air conditioning, Energy efficiency, Heating, Cooling, Steady state, Solar Absorption

INTRODUCTION

Conventional forced-air systems have been widely used to control indoor climate, but the use of radiant systems has rapidly increased during the last decades [1]. During the 70’s, the use of floor heating systems became very popular, especially in the residential sector. In the 90’s, cooled radiant ceilings have been increasingly installed in European offices because of longer overheated periods during summer time. More recently radiant walls have been introduced on the market, principally for cooling. Radiant systems are an efficient way of transporting energy [2], mainly due to the higher heat capacity of water. Moreover the large surface of exchange of radiant systems allows the use of source temperature closer to the room temperature, especially in low energy buildings.

Differences between radiant and forced-air systems can also be observed in the way of emitting energy. In fact, instead of transferring heat only by convection, this occurs partly by radiation (or from) the neighbouring surfaces, and partly by convection to (or from) the indoor air with a radiant system [3]. The influence of the terminal type will be the focus of this paper. Therefore four different terminals are selected (air conditioning, radiant floor, radiant wall and radiant ceiling) and their thermal performance will be compared for an office room located in Denmark. The heating and cooling seasons will be studied through numerical simulations under steady state conditions. The output of this study will be a better understanding of the parameters influencing their performance and also the advantages and drawbacks of the different solutions.

CASE STUDY

In order to study the influence of the terminal type on the energy efficiency, an office room located in Denmark has been considered. The internal dimensions of the room have been chosen similar to the PASSYS test cell: 5×2.76×2.75 m (length × width × height), resulting in a floor area of 13.8 m² [4]. 25% of the south facing wall is glazed: the window is double glazed with a g-value of 0.6. The construction elements are selected in accordance to the Danish Building Regulation 2010, meaning that the building energy need is lower than 71 kWh/m² per year for heating, cooling, ventilation, lighting and domestic hot water. The thermal characteristics of the building components are given in Table 1: ε corresponds to the surface emissivity (considering long-wave radiation) and α to the surface absorptivity (considering short-wave radiation). It has to be noticed that the window has a low-emissivity coating (low-e).

Four different terminals for cooling have been tested: radiant walls, radiant floor, radiant ceiling and air conditioning. The three radiant systems can be compared at the same level, as the activated areas are similar. Radiant terminals are embedded close to the surface, so
that their radiant effect applies directly on the internal constructions. The cooling or heating power of these terminals is assumed to be constant over the entire surface, therefore not taking into account the inhomogeneity due to the pipes layout.

Two different ventilation strategies have been tested: fully mixed or with a temperature gradient. The system is not equipped with heat recovery.

Table 1: Thermal properties of the construction elements (U-value given without including surface heat transfer coefficient).

<table>
<thead>
<tr>
<th></th>
<th>U'  (W/m².K)</th>
<th>ε_{long-wave} (°)</th>
<th>α_{short-wave} (°)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wall</td>
<td>0.15</td>
<td>0.8</td>
<td>0.6</td>
</tr>
<tr>
<td>Window</td>
<td>1.40</td>
<td>0.2</td>
<td>0.14</td>
</tr>
<tr>
<td>Floor</td>
<td>0.10</td>
<td>0.8</td>
<td>0.6</td>
</tr>
<tr>
<td>Roof</td>
<td>0.10</td>
<td>0.8</td>
<td>0.6</td>
</tr>
</tbody>
</table>

The outdoor conditions have been selected from the weather data of the Design Reference Year (DRY) in Copenhagen. Using these boundary conditions, dynamic simulations of an office room facing south have been performed using the software BSim [5] and two cases have been selected: one representing the winter season, and another one the summer season. The studied cases have been chosen so that, 95% of the time, the climate is less severe and leads to a lower energy consumption. Therefore it makes possible to compare the capacity of different terminals to provide an acceptable indoor climate over the year with a tolerance of 5%; this value has been chosen according to [6] and can be interpreted as a design parameter. The selected cases are summarised in Table 2.

Table 2: Definition of the winter conditions (26th of December, 12 o’clock) and summer conditions (13th of July, 15pm).

<table>
<thead>
<tr>
<th></th>
<th>WINTER</th>
<th>SUMMER</th>
</tr>
</thead>
<tbody>
<tr>
<td>T_{ext} (°C)</td>
<td>0.0</td>
<td>23.0</td>
</tr>
<tr>
<td>Q_{Solar normal} (W/m²)</td>
<td>11</td>
<td>296</td>
</tr>
<tr>
<td>Q_{Solar diffuse} (W/m²)</td>
<td>33</td>
<td>389</td>
</tr>
<tr>
<td>Solar azimuth (°)</td>
<td>174</td>
<td>234</td>
</tr>
<tr>
<td>Solar height angle (°)</td>
<td>11</td>
<td>47</td>
</tr>
</tbody>
</table>

PHYSIC OF THE ROOM

Modelling accurately the heat transfer in the room is essential when studying the influence of the terminal type. In fact, the only difference between radiant terminals and air conditioning is the nature of the heat or cold provided to the room, i.e. either convective or radiative. But this parameter will highly influence the indoor climate.

In order to establish the room heat balance, the surfaces are discretized in a total of 61 sub-surfaces, and the room air is assumed to be fully mixed (single zone model). A nodal scheme is then used to solve the heat transfer between the different elements. The conduction (Q_{cond}) through walls is calculated using the U-values (Table 1). The heat transfer is assumed to be one-dimensional, and thermal bridges are not considered. The effect of thermal mass does not need to be treated as only steady state conditions are studied in this paper. The different models used for convection, long-wave radiation and short-wave radiation are explained in the following paragraphs.

On the external side of the walls, a combined heat transfer coefficient is used to model the convective and radiative exchange with the ambient environment. This coefficient has been chosen equal to 29.3 W/m².K, which corresponds to the value for brick or rough plaster [7].

On internal surfaces, the convective heat transfer (Q_{conv}) is assumed to be natural and is modelled by a constant coefficient selected according to the surface slope and the type of convective flow observed. For a horizontal surface, the flow can either be facing upward or downward, depending on the temperature difference between the air and the surface considered. For example, in the case of a heated floor, the relatively hot, lighter fluid has a tendency to be convected upward in the form of plumes, being replaced by colder, denser fluid from above. This configuration is gravitationally unstable and the buoyancy forces will drive the convective motion. The associated convective heat transfer coefficient will be rather high. On the other hand, in the case of a cooled floor, the fluid is gravitationally stable and it leads to the formation of a stable layer. The downward flow will result in a low convective heat transfer coefficient [8].

Table 3: Definition of the convective heat transfer coefficients at the internal surfaces of the room [9, 10].

<table>
<thead>
<tr>
<th></th>
<th>h_{conv} (W/m².K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vertical surface</td>
<td>2.5</td>
</tr>
<tr>
<td>Horizontal upward</td>
<td>4.0</td>
</tr>
<tr>
<td>Horizontal downward</td>
<td>0.7</td>
</tr>
</tbody>
</table>

The radiative exchange can be split in two categories: (1) Short-wave radiation, also called solar radiation, is coming from a body heated at 5800 K. 75% of the flux is therefore emitted at a wavelength lower than 1 μm. (2) Long-wave radiation is emitted by objects at ambient temperature, around 300 K, and is concentrated in the
infra-red region (wavelength around a few micrometres). For example a black body heated at 300 K has a peak emission at 10 μm. The material properties are changing with the type of radiation considered, i.e. the type of spectrum (Table 1).

Long-wave radiative heat transfer between surfaces \( Q_{rad \ j \rightarrow i} \) has to be modelled accurately when establishing the room heat balance. Diffuse grey surfaces are assumed, and the window is taken to be opaque to long-wave radiation. The difficulty comes from the non-linearity of radiative exchange, which implies the use of either simplified linear models or iteration techniques. The algorithm chosen in this study is the one developed by Clarke [11]: it is a non-linear model, including direct reflection between two surfaces, but also indirect reflections on a third surface. The use of this model implies several iterations before convergence is reached (assessed by a change in the surface temperature smaller than 10^{-7} %). This method has been compared to the exact method, the radiosity method, and the obtained radiative heat fluxes do not differ by more than 3% for the considered cases.

One of the main issues, when calculating the cooling load in a building, is the computation of the incoming solar radiation \( Q_{solar} \). There are three main steps in the calculation of the solar load: (1) determination of the irradiation on the south façade, (2) separation of solar radiation absorbed and reflected by the glazing, and transmitted to the room, (3) calculation of the solar repartition in the room. Once the solar radiation is distributed on the different internal surfaces, radiation is treated as long-wave radiation.

(1) The irradiation on a vertical surface is calculated according to [12]. It takes into consideration direct, diffuse and reflected solar radiation.

(2) The part of solar radiation absorbed by the glazing and transmitted to the room is calculated using the software Window 6 [13]. In this program, the detailed calculation of reflection between the panes and the absorption and transmission of each pane is performed hemispherically for diffuse radiation and in steps of 10° incidence angle for direct solar radiation. Thanks to this program, it is possible to define the part of solar radiation directly transmitted \( (T_{sol}) \) and the part absorbed and then reemitted \( (SHGC - T_{sol}) \), with \( SHGC \) Solar Heat Gain Coefficient. In the studied case, around 5% of the solar radiation is absorbed and 50-55% is transmitted to the room (these values vary with the solar angle).

(3) Finally the part of transmitted solar radiation has to be distributed on the room surfaces. In BESTEST [7], a method is proposed to calculate the interior solar distribution of short-wave radiation depending on the solar absorptivity of surfaces. This technique assumes that no solar radiation is directly absorbed by the zone air; all incident solar radiation initially hits the floor and is then reflected over the other surfaces according to their view factors. The remaining amount of original solar radiation is then assumed to be absorbed by all surfaces in proportion to their area-absorptance products. For the considered case, the distribution is: 63% at the floor, 10% at the ceiling, between 4 and 9% at each wall, and 1% lost through the window. This type of distribution is proper to cases with low diffuse radiation.

The internal heat gains from equipment, lights and people are modelled as sensible heat load (no latent heat load), assuming that 40% of this heat load is convective \( Q_{conv \ int} \), and 60% is radiative \( Q_{rad \ int} \) [7]. The radiative heat loads are distributed on the different surfaces according to their relative area; no radiative internal heat load is applied to the window (low-e).

Finally the temperatures at the different nodes are determined by solving the room heat balance composed of 62 equations:

\[
\begin{align*}
Q_{ventilation} &= \sum_i Q_{conv \ (i)} + Q_{conv \ int} + Q_{conv \ terminal}
Q_{cond \ (i)} &= Q_{conv \ (i)} + \sum_j Q_{rad \ j \rightarrow i} + Q_{solar \ (i)} + Q_{rad \ int \ (i)} + Q_{rad \ terminal \ (i)}
\end{align*}
\]

\( Q_{conv \ terminal} \) and \( Q_{rad \ terminal} \) represent the heating (if positive) or cooling (if negative) power of the terminal. The validity of the code has been checked by comparing the obtained results with the software BSim [5]. A good agreement has been observed despite differences in the model used for representing the room physics.

**EVALUATION OF COMFORT**

The thermal comfort has been evaluated at a global scale using the operative temperature, but local thermal comfort parameters have also been assessed.

The operative temperature is defined as the mean value of the radiant temperature and air temperature; this evaluation of the operative temperature is valid for relative air velocity below 0.2 m/s [14].

As we consider the case of an office building, the air temperature is calculated at a height of 0.6 m, which corresponds to the height of a seated person.

The mean radiant temperature \( T_{rad} \) is also evaluated for a seated person (orientation not fixed). It can be expressed by the following equation [14]:

\[
\frac{T_{rad}}{T_{AMB}} = \frac{1}{T_{AMB}} \int_{0}^{\infty} \frac{1}{\tau} \left( 1 + \frac{\tau}{T_{AMB}} \right) e^{-\frac{T_{AMB}}{\tau}} d\tau
\]
\[ T_{\text{rad}} = 0.13 \left( T_{\text{pr [up]}} + T_{\text{pr [down]}} \right) + 0.185 \left( T_{\text{pr [right]} + T_{\text{pr [left]} + T_{\text{pr [front]} + T_{\text{pr [back]}}}} \right) \]

with \( T_{\text{pr}} \) plane radiant temperature

\[ T_{\text{pr}} = \sum_i T_i^4 F_{\text{pr-i}} \] (for high emissivity)

\( F_{\text{pr-i}} \) view factors between the plane and the surface \( i \) (calculated according to [14])

The factors used in the calculation of the radiant temperature correspond to the projected area factor of a seated person in the 6 directions. This calculation method has been preferred to the area-weighted method because it leads to more accurate results. Otherwise the radiant temperature would have been underestimated in case of cooled floor and overestimated in case of cooled ceiling (with an error up to 2 K).

As the focus of this project is on radiant solutions, two local comfort parameters have been evaluated: the radiant temperature asymmetry, which corresponds to the difference between the plane radiant temperatures on two opposite sides of a small plane element, and the surface temperatures. In all the simulations performed, it has been verified that the local discomfort parameters are within the range of category II, corresponding to a normal level of expectation: the limit values can be found in [15].

**RESULTS FOR HEATING**

In order to study the influence of the terminal type on the heating consumption of buildings, several simulations have been performed under typical winter conditions (Table 2). Several parameters can influence the energy efficiency of terminals: outdoor temperature, air change rate, temperature gradient... Therefore a sensitivity analysis has been conducted by setting the operative temperature equal to 21°C for all cases, and observing the influence of the different parameters on the energy performance and local comfort. Internal heat loads are not considered in the winter case.

Figure 1 presents the influence of the air change rate on the energy efficiency of convective and radiant terminals. It can be observed that, at low air change rate, the differences between the four terminals are rather small; the convective terminal is slightly more energy efficient at low air change rate. Nevertheless, when increasing the air change rate, radiant terminals show significant advantages. They are taking advantage of the high proportion of radiative exchange, which is therefore not greatly influenced by the air temperature and the air change rate. No significant differences can be observed between the three radiant terminals. At 3 ACH, the energy need for a radiant system is around 10% lower than for a convective system.

Even though all systems achieve the same level of global comfort, the local comfort conditions are not identical. Local parameters are changing only in the case of radiant terminal. When increasing the air change rate, a higher asymmetry between the air and the surface temperature can be observed. The air temperature is decreasing whereas the radiant temperature is increasing: at 3 ACH, this asymmetry can be up to 5 K. The maximum surface temperature is therefore increasing, as it can be seen in Figure 2. The radiant floor has the lowest surface temperature due to its high convective heat transfer (Table 3), whereas the ceiling has the highest one. The local comfort criteria are respected in all cases, except for the radiant ceiling: at air change rate higher than 2.5 ACH, the vertical radiant asymmetry is larger than 5 K.

In order to study the influence of the type of ventilation system, a temperature gradient of 4 K between the floor and the ceiling has been imposed to the room air. Thanks to this model, it is possible to simulate the effect of displacement ventilation. The temperature gradient chosen ensures that the local thermal comfort is achieved, i.e. that the vertical air temperature between 1.1 m and 0.1 m is lower than 3 K [15]. The temperature gradient leads to an increase of the heating consumption (Figure 3) because of warmer outlet air. At 3 ACH, the
increase is of the order of 10 % for the heated wall; similar results are obtained with other types of terminal.

In the summer case, more differences appear between the different radiant terminals. Radiant floor is the most efficient system despite the low convective heat transfer coefficient: this is due to the direct solar absorption (Figure 5). Radiant ceiling is the least efficient radiant terminal despite the high convective heat transfer coefficient: it can be explained by the low amount of solar radiation reflected to this surface and by the location of the operative temperature sensor. Moreover it can be observed that the portion of radiation in the terminal heat balance is almost constant, around 50 %. Only the convective part varies greatly according to the type of terminal: for floor cooling, it represents only 7% of the heat exchange, whereas it represents around 30 - 40% for the other radiant terminals. Similar results have been obtained experimentally by Novoselac et al. [16] in their study about cooled ceiling.

When having a closer look to local comfort, it can be observed that all the solutions achieved an acceptable indoor climate. The minimum surface temperature is always above 21°C, and the temperature asymmetry is within the recommended range. But radiant terminals, and especially the cooled ceiling and the cooled wall, create a more uniform environment, i.e. with a lower temperature difference between air and surfaces.

Simulations with different outdoor temperatures have also been performed (Figure 6) and they showed that the energy efficiency of air conditioning is highly dependent on the ambient conditions. In fact, as air conditioning terminals are cooling down the air, more energy will be needed if warmer air penetrates the building. Radiant systems are less sensitive to the inlet air temperature because these systems are principally cooling down the surfaces and can achieve the same level of comfort than air conditioning with warmer air.
The influence of the type of ventilation has been studied (Figure 7). Contrary to the heating case, the cooling power needed to achieve thermal comfort is reduced with a temperature gradient in the room. At 3 ACH, the required power is decreasing by 30%.

Figure 8 presents the cooling consumption depending on the floor absorptivity with an air change rate set to 2 ACH. When the floor absorptivity is decreasing, the cooling consumption is decreasing. This is due to the part of solar radiation redirected outside the building, which is changing: for a floor absorptivity of 0.8, this part is only 0.7%, whereas it increases up to 2.9% for an absorptivity of 0.2. Because of the small window area and the shape of the room, the influence on the cooling consumption is not so important, but it can become rather high for highly glazed surfaces like atria [17].

CONCLUSION

The thermal performance of four types of terminal (air conditioning, radiant floor, radiant wall and radiant ceiling) has been investigated numerically. The case of an office room located in Denmark has been simulated under winter and summer conditions. Not only has the energy efficiency been analysed, but also the local thermal comfort.

The parameter, which is influencing the most the thermal performance of terminals, is the air change rate. During winter, the ventilation rate should be minimised in order to decrease heat losses; in that case the four types of terminal have similar efficiency. Only at high air change rate radiant terminals show better performance. It has to be noticed that the heated ceiling has limited capacity due to the constraint on radiant asymmetry.

During summer, radiant panels show significant advantages compared to air conditioning. In fact outdoor air is often a few degrees above or below the cooling set-point. For air conditioning, this characteristic is a drawback because the cooling consumption will be highly dependent on the outdoor temperature. On the contrary, radiant terminals are much less sensitive because they are operating at higher air temperature: the ventilation system is therefore often assisting the cooling process. Moreover radiant systems create a more uniform indoor environment. Significant differences between radiant terminals can only be observed in the summer case due to high solar radiation. Floor cooling is the most efficient solution thanks to the direct solar absorption, but the repartition of solar radiation plays an important role in this efficiency. The results and conclusions made in this paper might differ with other assumptions.

Finally the effect of displacement ventilation has been simulated. The temperature gradient created leads to an increase of the heating consumption, whereas the cooling consumption is reduced.
REFERENCES
12. ASHRAE (2009). Handbook of Fundamentals, Chapter 14 - Climatic Design Information, Atlanta GA.