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Load calculations of radiant cooling systems for sizing the plant

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Abstract

The aim of this study was, by using a building simulation software, to prove that a radiant cooling system should not be sized based on the maximum cooling load but at a lower value. For that reason six radiant cooling models were simulated with two control principles using 100%, 70% and 50% of the maximum cooling load. It was concluded that all tested systems were able to provide an acceptable thermal environment even when the 50% of the maximum cooling load was used. From all the simulated systems the one that performed the best under both control principles was the ESCS ceiling system. Finally it was proved that ventilation systems should be sized based on the maximum cooling load.

Keywords: Radiant cooling systems; TABS; ESCS; thermal environment; cooling plant; sizing; high temperature cooling

1. Introduction

In 2007 the European Union set some very ambitious targets regarding climate and energy, known as the “20-20-20 targets” [1]. In order for the member countries to achieve those targets, drastic changes have to be realized in the building sector since it is responsible for the 38.5% of the energy consumption in the European Union [2]. An effective solution that would lead towards this direction is the increased use of radiant heating and cooling installations, as they are considered more efficient compared to conventional HVAC systems. First of all, water has a higher heat capacity compared to air, and pumps are more efficient compared to fans [3]. Furthermore, the water supply temperature is close to room temperature [4]. In addition to the high energy efficiency, radiant systems outweigh conventional HVAC installations in terms of indoor climate, since they provide a better and more uniform thermal environment [5] and the risk of draught is limited [6]. Although radiant heating systems have been used for decades, only recently they have become popular in cooling installations [7]. Radiant cooling systems are separated into three categories, the Thermally Active Building Systems (TABS), the Embedded Surface Cooling Systems (ESCS) and the Radiant Cooling Panels [8].

In order to avoid over- or under dimensioning the cooling plant, which will lead in higher acquisition and operation and maintenance cost or in overheating in building respectively, particular attention should be paid during the design phase. Nonetheless, only a few studies have been conducted regarding sizing radiant cooling systems. In one of these studies, Feng et al. [9] compared several radiant cooling systems with a ventilation system. In this research, the latent heat loads were not taken into consideration. This decision bears the risk of leading to inaccurate results since the latent heat loads are related to absolute humidity which is affecting the air temperature and the risk of condensation occurrence on the radiant surface [10]. Furthermore, Feng et al. [9] used the maximum water flow rate when using the radiant cooling system and one of the conclusions was that a radiant system requires a system with higher cooling capacity than a ventilation system.

The aim of this study was to prove that a radiant system should not be sized based on the peak cooling load but at a lower value, and that this technique cannot be applied in a conventional ventilation system. In contrast to the procedure followed by Feng et al. [9], a more detailed approach was followed, in which the latent heat loads were included, and thus the corresponding ventilation system to remove them.

2. Methodology

In order to prove the aforementioned hypothesis the software TRNSYS was used, in which an office room was simulated. The dimensions of the office were 8m x 6m x 2.7m (L x W x H) and this is the geometry suggested by ASHRAE standard 140 [11]. On the south façade there were two glazing surfaces with dimensions 3m x 2m (L x H). This geometry, the properties of the construction elements and the radiant systems were the ones used by Feng et al. [9] to establish a valid comparison.

In this office, six radiant cooling systems were tested; three TABS systems (two floor and one ceiling systems) and three ESC systems (two floor and one ceiling systems) and afterwards the radiant systems were replaced by a ventilation system. The two floor systems and the two ceiling systems were the ones used by Feng et al. [9], while the two additional floor systems are identical to the ceiling systems but reversed. For each of these systems, three simulations were run and in these simulations the systems were sized based on the max cooling load, 70% of the max cooling load, and 50% of the max cooling load. For all the simulations the International Weather for Energy Calculations (IWEC) file for Copenhagen was used.

According to Wilkins and Hosni [12], the sum of the peak heat gain of all equipment separately is higher than the actual peak heat gain, due to usage diversity. Based on this study it was decided that the office will be simulated with 6 occupants, whose heat loads were 75 W of sensible heat gain and 75 W of latent heat gain based on the standard ISO 7730 [13] and the occupancy period was typical office hours, namely 8:00 – 18:00. The internal heat gains caused by the corresponding equipment for 6 occupants, namely computers, monitors, printer and facsimile equipment, were 759 W which refers to heavy load, which is the most demanding case that could be simulated.

The ventilation system that should be operating along with the radiant system in order to remove latent heat loads and pollutants was sized based on category II of the standard EN 15251 [14]; the flow rate was 272 m^3/h (2.1 ACH) and the supply air temperature was 20^oC and was operating only during occupancy period.

In order to size the cooling system, an annual simulation was run with unlimited cooling capacity. From the initial simulation the maximum cooling demand was extracted as a result and was used to calculate the maximum water and air flow rates. Afterwards the cooling system with the unlimited cooling capacity was initially replaced by the radiant systems and finally by the ventilation system.

For the radiant systems two control methods were examined, namely time and temperature control. In the time control the cooling system was operating consecutively from 6:00 to 18:00 every day. They were set to operate prior to the occupancy period, because of the time lag due to the thermal mass of the radiant systems. Therefore for the examined period the radiant systems were operating for 2568 hours in total. Although TABS are considered to be suitable for operating during night time [8], it was decided, due to the high internal heat gains, to follow this operation profile which was the same as the one used by Park et al. [3]. In the temperature control simulations the system would be turned on if the operative temperatures exceeded 25^oC and would continue operating until it dropped below 23^oC. With the temperature control methods only the radiant ceiling systems were tested. The properties of the piping system are presented in Table 1 and are identical to the ones used by Feng et al. [9].

Regarding the ventilation system simulations, the ventilation system was operating only during occupation period since this is the common practice, with air supply temperature of 20^oC. Thus, the ventilation system was operating in total for 2140 hours for the whole simulated period.

In this study it was not attempted to compare directly the radiant systems with the ventilation system, but to examine whether the hypothesis of reducing the capacity is applicable in the two systems. Therefore, a simple control for the ventilation system was considered adequate to prove that this idea cannot be applied on a ventilation system. Finally, for all the simulations the time step was 6 minutes.

Table 1. Piping system properties

Pipe spacing (m)	Pipe outer diameter (m)	Pipe thickness (m)	Pipe thermal conductivity (W/mK)	Maximum water flow rate (kg/h)*	Water supply temperature (^oC)
0.15	0.02	0.005	0.35	658	15

*The maximum flow rate was defined by the initial model simulation

3. Results

From the initial model simulation the maximum cooling demand was calculated to be 3.8 kW. As it was mentioned before, based on this value the water and the air flow rates were calculated, and the results are presented in Table 2. For the radiant systems the ΔT was estimated to be 5^oC while for the ventilation system 6^oC.

Table 2. Required water and air flow rates

	Water flow rate (kg/h)	Air flow rate (m^3/h)	Air flow rate (ACH)
Maximum cooling demand	658	1878	14.5
70% of maximum cooling demand	461	1315	10.1
50% of maximum cooling demand	329	939	7.2

The cooling period is considered from the 1st of May until the 30th of September, but due to the high cooling demand outside this period, it was decided to extend it from the 1st of April to the 31st of October, and this was the period that was simulated in the rest of the simulations.

In Table 3 the results from the temperature control simulations are presented when the maximum cooling demand was used, while in Tables 4 and 5 the results from the simulations where the 70% and the 50% of the maximum cooling demand was used are presented, respectively. The extracted results consist of the hours where the operative temperature was below 23.5°C, 23°C and 22°C and above 25.5°C, 26°C and 27°C in percentage of the total occupied hours during the simulated period. These temperature values are defined by the standard EN 15251 [14] as ranges for category I, II and III. Furthermore the average relative humidity (%), the minimum radiant surface temperature (°C) and the maximum absolute humidity (kg_w/kg_{air}) were extracted. The minimum radiant surface temperature and the maximum absolute humidity were used to estimate whether there is risk of condensation occurrence on the radiant surface as it is specified in EN 1264-3 [10].

Table 3. Results from time control simulations with 100% of maximum cooling demand

	Category I		Category II		Category III				
Radiant system	Hours below 23.5°C (%)	Hours above 25.5°C (%)	Hours below 23°C (%)	Hours above 26°C (%)	Hours below 22°C (%)	Hours above 27°C (%)	Surface temperature (°C)	Aver. rel. humidity (%)	Max. abs. humidity (kg_w/kg_{air})
TABS floor	3.6	2.6	3.1	1.0	2.2	0.0	18.1	47.5	0.008
TABS floor 2	3.4	3.0	3.0	1.2	2.0	0.1	18.4	47.4	0.008
TABS ceiling	3.4	1.6	3.2	0.7	2.1	0.0	18.4	47.9	0.008
ESCS floor	3.3	3.6	2.8	1.5	1.8	0.2	18.4	46.9	0.008
ESCS floor 2	3.5	2.6	3.0	0.9	2.0	0.0	18.1	47.3	0.008
ESCS ceiling	4.3	0.0	3.9	0.0	3.0	0.0	18.2	49.8	0.008

Table 4. Results from time control simulations with 70% of maximum cooling demand

	Category I		Category II		Category III				
Radiant system	Hours below 23.5°C (%)	Hours above 25.5°C (%)	Hours below 23°C (%)	Hours above 26°C (%)	Hours below 22°C (%)	Hours above 27°C (%)	Surface temperature (°C)	Aver. rel. humidity (%)	Max. abs. humidity (kg_w/kg_{air})
TABS floor	3.2	5.4	2.7	2.5	1.8	0.5	18.2	47.3	0.010
TABS floor 2	3.1	5.2	2.6	2.5	1.7	0.5	18.4	46.5	0.008
TABS ceiling	3.3	3.6	2.8	1.7	1.8	0.2	18.5	47.0	0.008
ESCS floor	2.9	6.7	2.5	3.4	1.5	0.7	18.5	46.0	0.008
ESCS floor 2	3.2	4.9	2.8	2.3	1.7	0.4	18.1	46.5	0.008
ESCS ceiling	3.9	0.7	3.5	0.2	2.5	0.0	18.3	48.8	0.008

Table 5. Results from time control simulations with 50% of maximum cooling demand

	Category I		Category II		Category III				
Radiant system	Hours below 23.5°C (%)	Hours above 25.5°C (%)	Hours below 23°C (%)	Hours above 26°C (%)	Hours below 22°C (%)	Hours above 27°C (%)	Surface temperature (°C)	Aver. rel. humidity (%)	Max. abs. humidity (kg_w/kg_{air})
TABS floor	2.7	10.3	2.3	5.9	1.5	1.4	18.4	46.1	0.008
TABS floor 2	2.7	10.5	2.2	6.0	1.4	1.4	18.6	45.3	0.008
TABS ceiling	2.8	7.3	2.3	3.9	1.5	0.9	18.7	45.9	0.008
ESCS floor	2.5	12.3	2.0	7.1	1.3	1.9	18.7	45.0	0.008
ESCS floor 2	2.7	10.3	2.2	6.0	1.4	1.2	18.3	45.3	0.008
ESCS ceiling	3.7	1.5	3.2	0.6	2.2	0.0	18.4	48.1	0.008

In Tables 6, 7 and 8 the results from the temperature control simulations are presented for 100%, 70% and 50% of the maximum water flow rate respectively. In these tables one additional result has been added, the hours of operation of the radiant system, since in theses simulations it was not a constant value as before.

Table 6. Results from temperature control simulations with 100% of maximum cooling demand

	Category I		Category II		Category III					
Radiant system	Hours below 23.5°C (%)	Hours above 25.5°C (%)	Hours below 23°C (%)	Hours above 26°C (%)	Hours below 22°C (%)	Hours above 27°C (%)	Hours of operation	Surface temperature (°C)	Aver. rel. humidity (%)	Max. abs. humidity (kg_w/kg_{air})
TABS ceiling	0.3	14.1	0.1	6.5	0.0	0.1	1310	22.3	41.2	0.008
ESCS ceiling	0.2	0.0	0.1	0.0	0.0	0.0	1161	21.9	41.8	0.008

Table 7. Results from temperature control simulations with 70% of maximum cooling demand

	Category I		Category II		Category III					
Radiant system	Hours below 23.5°C (%)	Hours above 25.5°C (%)	Hours below 23°C (%)	Hours above 26°C (%)	Hours below 22°C (%)	Hours above 27°C (%)	Hours of operation	Surface temperature (°C)	Aver. rel. humidity (%)	Max. abs. humidity (kg_w/kg_{air})
TABS ceiling	0.3	18.9	0.1	10.0	0.0	0.8	1414	22.39	41.0	0.008
ESCS ceiling	0.2	7.2	0.1	2.1	0.0	0.0	1272	22.1	41.6	0.008

Table 8. Results from temperature control simulations with 50% of maximum cooling demand

	Category I		Category II		Category III					
Radiant system	Hours below 23.5°C (%)	Hours above 25.5°C (%)	Hours below 23°C (%)	Hours above 26°C (%)	Hours below 22°C (%)	Hours above 27°C (%)	Hours of operation	Surface temperature (°C)	Aver. rel. humidity (%)	Max. abs. humidity (kg_w/kg_{air})
TABS ceiling	0.3	25.0	0.1	15.1	0.0	2.4	1543	22.5	40.6	0.008
ESCS ceiling	0.2	13.9	0.1	6.6	0.0	0.4	1404	22.2	40.9	0.008

In Table 9 the results from the simulations with the ventilation system are presented. Since the system failed to provide acceptable thermal environment when the 70% of the maximum air flow rate was used, it was decided that it was unnecessary to run a simulation with 50% of the maximum air flow rate.

Table 9. Results from the simulations with the ventilation system

	Category I		Category II		Category III			
Air flow rate level	Hours below 23.5°C (%)	Hours above 25.5°C (%)	Hours below 23°C (%)	Hours above 26°C (%)	Hours below 22°C (%)	Hours above 27°C (%)	Aver. rel. humidity (%)	Max. abs. humidity (kg_w/kg_{air})
100% of the maximum	2.4	14.2	1.9	15.7	1.3	4.8	36.4	0.008
70% of the maximum	0.9	36.8	0.7	24.0	0.3	8.7	35.2	0.007

4. Discussion

As it can be observed from the results from the initial model simulation, there is a very high cooling demand, which led to very high water and air flow rates. This was caused because there was no attempt to decrease the sensible heat gains, since the purpose of this study was to prove that a radiant cooling system can operate satisfactorily under random conditions even if sized based on a lower value than the maximum cooling demand. For example, although the windows are facing south, no solar shading system was implemented.

From Tables 3, 4 and 5 it can be seen that all four floor models are in category II, II and III of standard EN 15251 [14] when sized based on the 100%, 70% and 50% of the maximum cooling demand respectively. On the other hand, the ceiling systems performed much better since the TABS ceiling system was respectively in category I, II and III while the ESCS ceiling model drop to category II only when the 50% of the maximum cooling demand was used. The better performance of the ceiling cooling systems was expected from the literature [8, 15].

From the results of temperature control simulations shown in Tables 6, 7 and 8, it can be seen that the TABS ceiling system is always in category III, while ESCS ceiling system is in category I, II and III when sized based on the 100%, 70% and 50% of the maximum cooling demand respectively. The reason why the ESCS model performed better is its lower thermal mass compared to the TABS model, which requires more time to adjust to temperature changes.

Based on the minimum radiant surface temperature and the maximum absolute humidity results from Tables 3, 4 and 5 it can be concluded that there is no risk of condensation occurrence since in all cases the radiant surface temperature is above 18°C and the absolute humidity is below 12 kg_w/kg_{air}, as it is suggested from the standards EN 1264-3 and EN 15251 [10, 14]. That was the result also when the temperature control was used, as it can be seen from Tables 6, 7and 8.

Although the two models examined under the temperature control principle did not perform as well as in the case of time control principle regarding the thermal comfort, it can be observed that they operated for significantly less hours when the temperature control was used.

When the ventilation system was used, only when the maximum air flow rate was used the system’s performance was in category III of standard EN 15251 [14].

As it can be seen from a relative humidity point of view, all the examined systems, either radiant or ventilation, were evaluated as category I based on standard EN 15251 [14].

In this study energy performance/consumption was not considered, but the energy investigation is crucial to reach a definitive conclusion regarding the overall performance of the different radiant cooling systems. In addition to that, as it is mentioned before, a simplified control was used for the all the examined systems, therefore further investigation is required in terms of the control principle of both the radiant and the ventilation systems. In this study a simple control was considered as adequate to prove the initial hypothesis that was examined.

5. Conclusion

In this simulation study six radiant cooling systems were examined under two control principles, namely time and temperature control, to prove that a radiant cooling system should not be sized based on the maximum cooling load but at a lower value.

Based on the conducted simulations, it was concluded that all tested radiant systems performed satisfactory in terms of thermal environment even when the 50% of the maximum water flow rate was used. Consequently, radiant cooling systems should be avoided to be sized based on the 100% of the maximum cooling demand since this will lead to oversized systems which results in higher acquisition, operation and maintenance costs and higher energy consumption. On the other hand, this hypothesis cannot be implied on ventilation systems where the systems should be always sized based on the maximum cooling load. These conclusions come in contrast to the results from Feng et al. [9] who claimed that a radiant system requires higher cooling energy compared to an ventilation system.

Furthermore, among all the tested systems, the one that performed the best with both control methods was the Embedded Surface Cooling ceiling system.

As previously mentioned, when the time control principle was used the radiant cooling systems performed better in terms of thermal comfort but were operating for remarkably longer periods. Therefore it depends on the needs of the occupants whether to choose a control principle that will provide a better thermal environment or prefer to minimize the energy consumption.

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