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Publication date:
2017

Document Version
Peer reviewed version

Cooling load calculations of radiant and all-air systems for commercial buildings

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Abstract

The authors simulated in TRNSYS three radiant systems coupled with a 50% sized variable air volume (VAV) system and a 50% sized all-air VAV system with night ventilation. The objective of this study was to identify the differences in the cooling load profiles of the examined systems when they are sized based on different levels of the maximum cooling demand. The authors concluded that for high thermal mass radiant system nocturnal operation was adequate for providing an acceptable thermal environment even when the radiant system was sized based on the 50% of the maximum cooling demand. The 50% all-air system alone was able to provide comfort if night cooling was implemented. On the other hand, radiant cooling panels (low thermal mass) should be operating during the occupancy period. When sizing a high thermal mass radiant cooling system, the effect of thermal inertia and the response time should always be taken into account.

Nomenclature

Symbol	Unit	Quantity
a	$W/(m^2 \cdot K)$	Heat transfer coefficient
C_p	$KJ/(kg \cdot K)$	Thermal capacity
K_H	$W/(m^2 \cdot K)$	Radiant system dependent coefficient
m	kg/h	Water flow rate
q_{hydr}	W/m^2	Heat flow on the hydronic side
q_{surf}	W/m^2	Heat flow on the radiant surface
T_{des}	$°C$	Desired room temperature
T_r	$°C$	Water return temperature
T_s	$°C$	Water supply temperature

Introduction

Load calculations are an important step when sizing a heating or cooling system to avoid under- or over-sizing it. According to ASHRAE Handbook of Fundamentals, cooling loads are the rate of energy required to be removed to maintain the indoor environment at the desired temperature and humidity conditions (ASHRAE 2013c). Although the procedure of cooling load calculations is accurately defined for all-air systems, this is not the case for radiant systems. This is because all-air systems are purely convective, while radiant systems interact with the room through both convection and radiation. Therefore, the heat exchange in the case of radiant systems is more complex. Most of the studies comparing radiant and all-air systems focus on indoor environmental quality or energy use, but not on the load calculations (Olesen & Mattarolo 2009; Fabrizio et al. 2012; Karmann et al. 2017). Energy simulations and experimental data aimed at proving that actual definition of cooling load is inadequate for radiant systems and showed that all radiant systems types have higher cooling loads than air system if they use the same control and comfort objectives (Feng et al. 2013; Feng et al. 2014). Thermally active building systems (TABS) can be operated during the night and therefore are able to shave and shift the load resulting in a lower cooling load than air systems (Lehmann et al. 2007; Rijksen et al. 2010). In addition to that, radiant systems cannot be categorized and operate all in the same way, since high thermal mass systems such as thermally activated building systems (TABS) or embedded surface systems (ESS) operate differently from lightweight systems such as radiant ceiling panels (RCP) due to the different response times. Furthermore, radiant systems can remove only sensible heat gains and they should operate in combination with a variable air volume (VAV) system, which provides fresh air to the occupants and removes latent loads and pollutants. In addition to that, the VAV will also remove an amount of the sensible cooling loads if the air supply temperature is lower than the room air temperature, Therefore, when sizing a radiant cooling system, the operation of the VAV should be taken into consideration.

The objectives of this simulation study were to identify and compare the differences in cooling loads for all-air and three types of radiant systems (TABS, ESS, RCP), when they are sized based on the maximum cooling demand, 70% and 50% of it.

Methodology

Simulation model geometry and heat loads

The software TRNSYS was utilised to model an office room. The dimensions of the room were 8 m x 6 m x 2.7 m and this geometry was suggested by ASHRAE Standard 140 (ASHRAE 2012). On the south façade there were two windows with dimensions 3 m x 2 m each (a window to wall ratio of 0.28). The U-value of the windows was 1.3 W/m²K while the Solar Heat Gain Coefficient was 0.3.

Wilkins & Hosni (2000) defined different diversity and load levels for office buildings. Based on this study and the room size, the authors concluded that six occupants would be a realistic occupancy level. The occupancy period was typical office hours, namely 8:00 – 18:00 and the heat load of each occupant was 75 W of sensible heat gains and 75 W of latent heat gains based on the ISO Standard 7730 (EN ISO 2006). The internal heat gains from the occupants' equipment were 420 W (8.75 W/m²), while the heat gains from the artificial lighting were 240 W (5 W/m²). The peak heat gains of the design day were 2922 W, namely 60.8 W/m². The authors followed a weekly schedule, consisting of five working days and weekends. The time step of the simulations was 3 minutes and the timebase 0.5, which ensured smooth curves without steps. For all the simulations, the authors used the International Weather for Energy Calculations (IWEC) file for Oakland, USA.

Table 1 shows the properties of the materials of the layers of the radiant systems. For TABS and ESS the embedded pipes were in the middle of the concrete and the lime plaster layer respectively, while in the case of RCP the pipes were attached on the aluminum surface.

Table 1: Properties of the material layers of radiant systems

Material	**Thickness, m**	**Specific heat capacity, kJ/kgK**	**Density, kg/m³**	**Conductivity, W/mK**
TABS				
Concrete	0.08	1	1400	1.13
Insulation	0.112	0.84	12	0.04
Roof deck	0.019	0.9	530	0.14
ESS				
Lime plaster	0.026	0.84	1050	0.7
Insulation	0.05	1.21	56	0.02
Concrete	0.08	1	1400	1.13
Roof deck	0.019	0.9	530	0.14
RCP				
Aluminum panel	0.001	0.91	2800	273
Insulation	0.05	1.21	56	0.02
Concrete	0.08	1	1400	1.13
Insulation	0.112	0.84	12	0.04
Roof deck	0.019	0.9	530	0.14

Sizing the radiant systems

Using this office, the authors simulated three radiant cooling systems; a TABS floor system, an ESS floor system and an RCP system. Those systems were the ones used by Feng et al. (2013).

In order to size TABS and ESS, first, the authors ran an annual simulation with unlimited cooling capacity. From that simulation, the authors identified the maximum cooling demand and used it to calculate the water flow rates using the following three equations:

$$q_{surf} = a \cdot (T_{surf} - T_{des}) \qquad (1)$$

$$q_{surf} = K_H \cdot \frac{T_s - T_r}{ln\frac{T_s - T_{des}}{T_r - T_{des}}} \qquad (2)$$

$$m = \frac{q_{hydr}}{C_p \cdot (T_s - T_r)} \qquad (3)$$

From Equation 1, the authors calculated the maximum heat load the radiant surface can remove in steady state conditions, by considering that the temperature of the floor surface was 19°C. This is the minimum floor temperature allowed for comfort according to ISO Standard 11855-1 and ASHRAE Standard 55 (ISO 2012; ASHRAE 2013b). K_H is a coefficient depending on the properties of the radiant system, and it was calculated based on ISO Standard 11855-2 (ISO 2012a) for the TABS and the ESS. From Equation 2 the water return temperature was calculated for different water supply temperature values from 10°C to 22°C. For each set of water supply and return temperatures, the authors calculated the corresponding water flow rate from Equation 3, taking into consideration that when steady state is reached, q_{surf} and q_{hydr} are considered equal. Afterwards, the lower and upper limits for the water flow rate were defined based on the following two conditions:

- The maximum water speed should be 1.2 m/s (ISO 2012)
- The Reynolds number should be at least 4000 to have a turbulent water flow (ISO 2012)

Equation 2 is not applicable for RCP since it describes only systems with high thermal mass. For the RCP, the authors set the minimum surface temperature to 17.5°C, and the RCP was covering only 50% of the ceiling surface.

TABS and ESS were operating during the unoccupied period, namely from 18:00 to 08:00 while RCP was operating during the occupancy period. In addition to the time constraint, the radiant systems deactivated if the operative temperature was below 23°C or if the floor surface temperature dropped below 21.5°C to avoid overcooling. This setpoint was defined based on trial & error. In Table 2 the properties of the pipes of the radiant system are presented.

Table 2: Piping properties

Pipe spacing, m	Pipe outer diameter, m	Pipe thickness, m	Pipe thermal conductivity, W/mK
0.15	0.02	0.005	0.35

Table 3 shows the water flow rate and supply temperature for the three radiant systems as the authors calculated them using Equations 1 to 3.

Table 3: Water flow rate and supply temperature for radiant systems

Cooling power level		TABS	ESS	RCP
100%	Flow rate kg/h	234	205	192
	Supply temperature, ˚C	14	13	14
70%	Flow rate kg/h	193	212	135
	Supply temperature, ˚C	17	17	14
50%	Flow rate kg/h	234	224	96
	Supply temperature, ˚C	20	20	14

Sizing the VAV system

A VAV system was used to provide ventilation and cooling. The minimum air flow rate was defined by the equation 6.2.2.1 of the ASHRAE Standard 62.1 (ASHRAE 2013a) which requires a minimum air flow rate of 4.8 L/s per person (0.8 ACH). The authors calculated the maximum air flow rate by Equation 3, where C_p was the thermal capacity of the air, and it was 69 L/s per person (1500 kg/h or 12 ACH). A PI controller adjusted the air flow rate. The input to the PI controller was the room operative temperature, and the setpoint was 26°C. The authors arbitrarily decided to have the maximum flow rate for the VAV system that was operating with the radiant systems set to 50% of the maximum air flow rate of the all-air system. During the unoccupied hours, the VAV that was operating along the radiant systems had to deliver in total two air volumes, equivalent to 0.14 ACH. During the unoccupied hours, the all-air system was operating at 50% of the maximum air flow rate, but only if the floor surface temperature was above 20°C and the air temperature 2°C higher than the outdoor air temperature. During that period the heating and cooling coils were deactivated and the VAV was supplying directly outdoor air. During the occupancy period, the air supply temperature was between 15°C and 20°C depending on the outdoor air temperature. If the outdoor air temperature was below 15°C, the heating coil would operate to increase the air supply temperature to 15°C, while if the temperature was above 20°C the cooling coil would activate to reduce it to 20°C. When the outdoor air temperature was between 15°C and 20°C, both coils were deactivated.

Results

Thermal performance

Since it is difficult to visualize annual performance in a single figure, the following figures illustrate the 15th of November since the highest cooling demand occurred on that date. Figure 1 presents the temperature performance for the four systems examined with the maximum cooling power.

In Figure1a and 1b, the radiant floor systems had almost equal air and operative temperature. A 1°C difference was observed in the case of TABS only when the air flow rate increased to deal with the excess heat in the room. That was observed also in the case of the RCP. In the case of the all-air system, due to the night-time ventilation the previous night, the minimum air flow rate was sufficient for keeping the operative temperature setpoint below 26°C for almost half of the occupancy period. In all the cases the operative temperature was at the setpoint, therefore all four systems provided equal comfort.

Figure 2 shows the temperature performance for the four systems examined for the simulations with 70% of the maximum cooling power. In the cases of TABS and ESS, no significant differences were observed with the previous case. On the other hand, the decrease in the cooling power of the RCP required a higher airflow to maintain the room operative temperature at 26°C. Due to the night-time precooling, the all-air system was able to maintain the operative temperature at the setpoint of 26°C despite the reduction of the maximum cooling power. For the all-air system, despite the reduction of the air flow rate during the night-time, the air flow rate was almost identical compared to the previous case.

Figure 3 shows the temperature performance for the simulations with 50% of the maximum cooling power. In the case of TABS, the VAV system was operating at a higher air flow rate than before, which explains the difference between air and operative temperature. A higher air flow rate was observed in the case of ESS for most of the occupancy period. In the case of RCP, an insignificant increase in the operative temperature was observed. All three radiant systems and the all-air systems were able to maintain the operative temperature at 26°C.

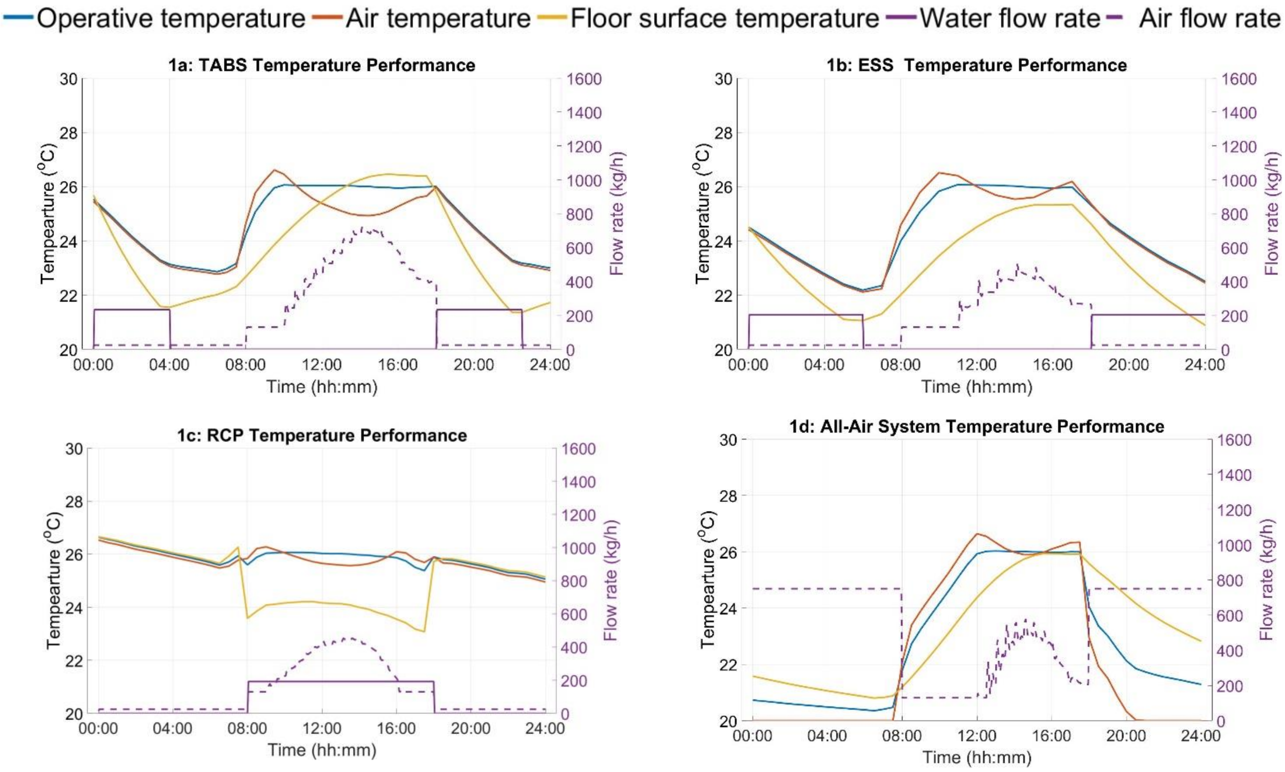

Figure 1: Temperature performance of the simulations with 100% of the maximum cooling power

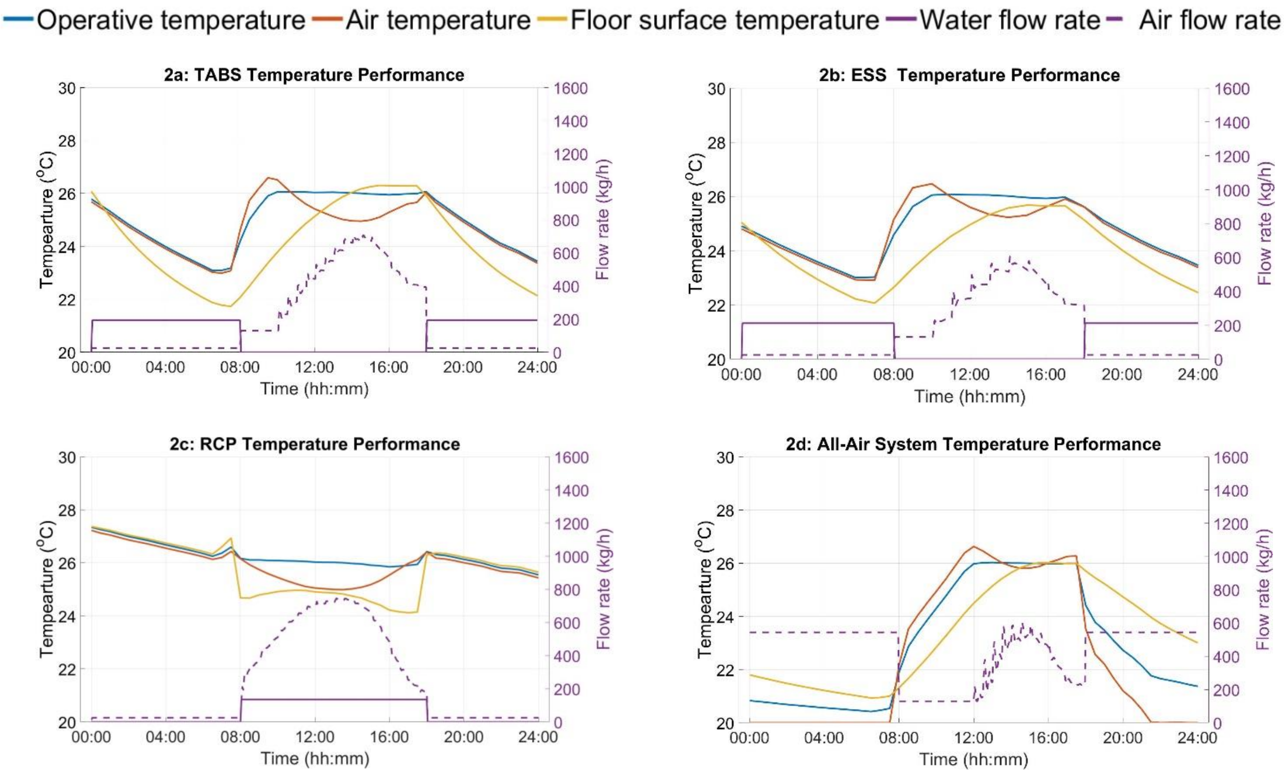

Figure 2: Temperature performance of the simulations with 70% of the maximum cooling power

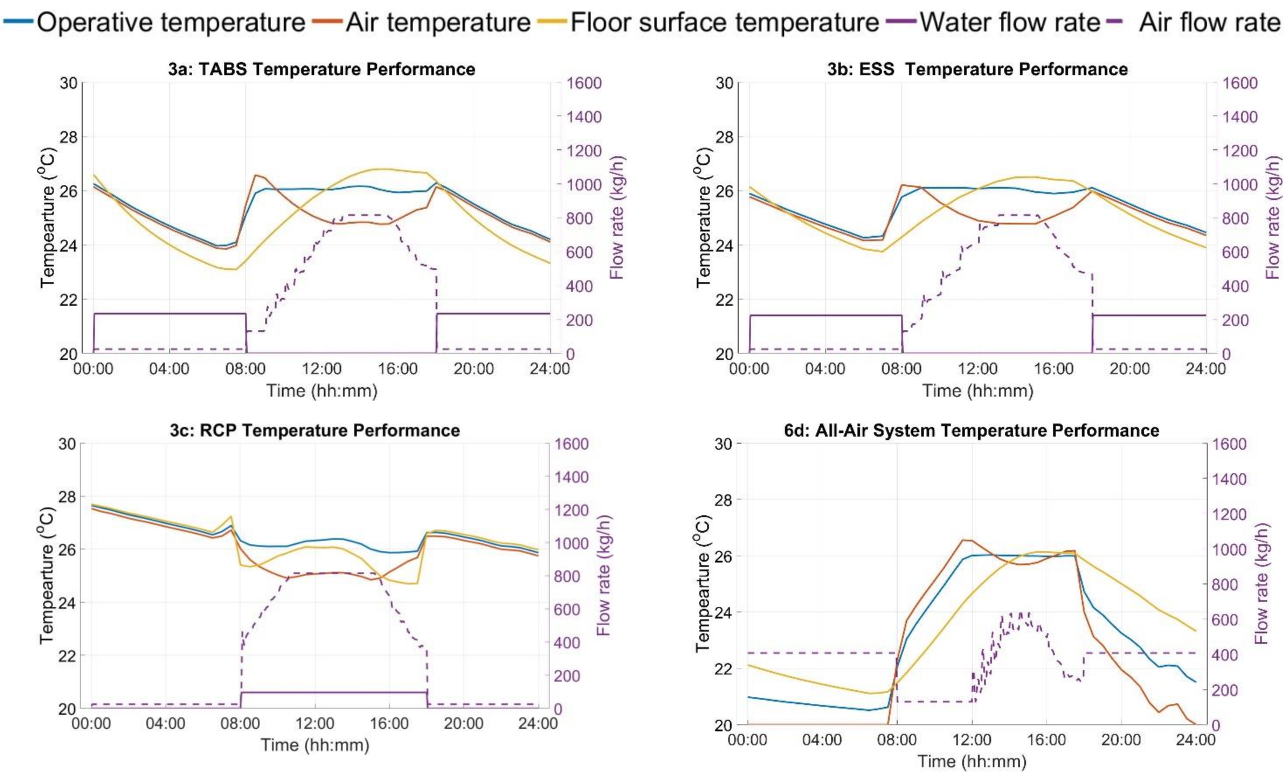

Figure 3: Temperature performance of the simulations with 50% of the maximum cooling power

Cooling power performance

Figure 4 shows the cooling performance of the four systems examined with the maximum cooling power. The authors compared the simulated systems in terms of cooling power of the VAV system and cooling power on the hydronic and the surface side. The purple curve illustrates the heat gains during that day, namely the sum of the solar heat gains, the heat from the occupants, the office equipment and the artificial lighting.

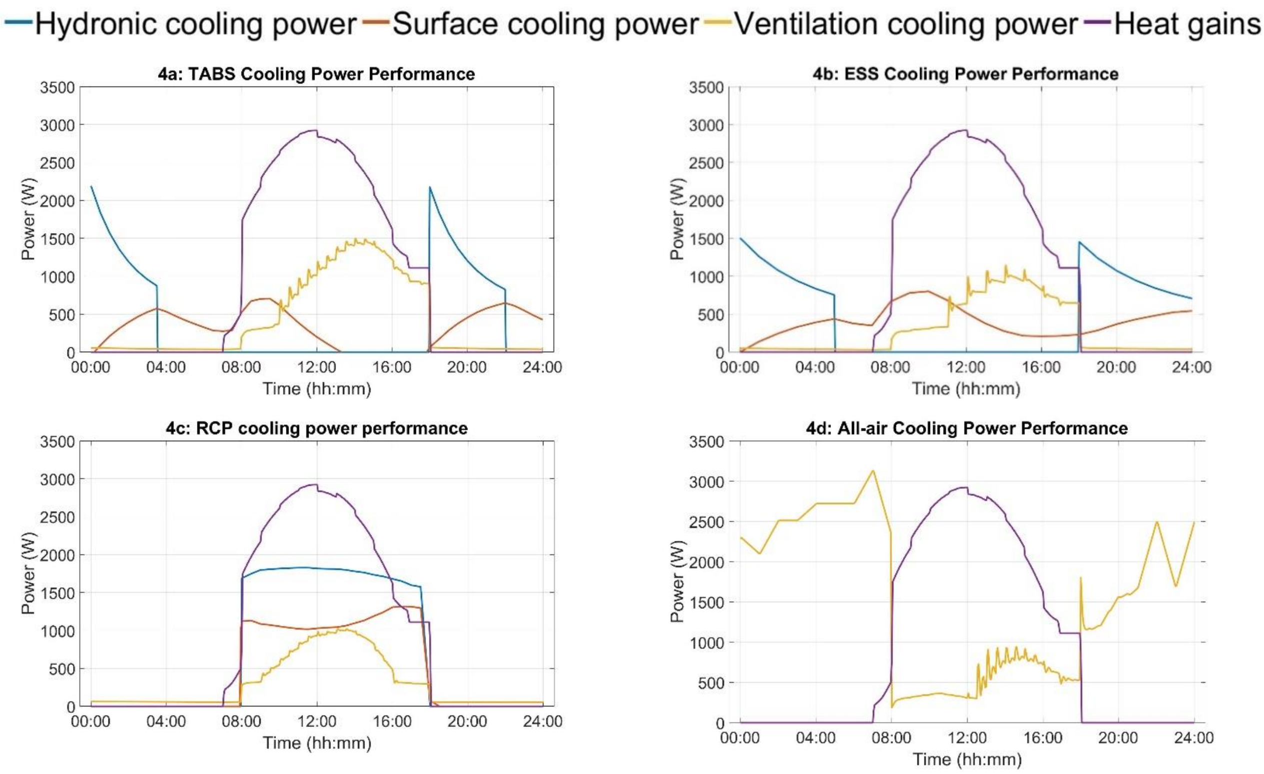

Figure 4: Cooling power performance of the simulations with 100% cooling power

Both TABS and ESS achieved a significant reduction in the peak cooling power to provide the required thermal environment. In contrast to that, the peak cooling power of the RCP was almost the same as the maximum heat gains if the hydronic and the VAV cooling power are added. For the all-air system, the daily profile was slightly lower than the air system couple with the radiant systems, at night the values are high, but in this climate, the cooling does not require a refrigeration cycle. Figure 5 illustrates the cooling performance of the four systems examined with 70% of the maximum cooling power of the examined system.

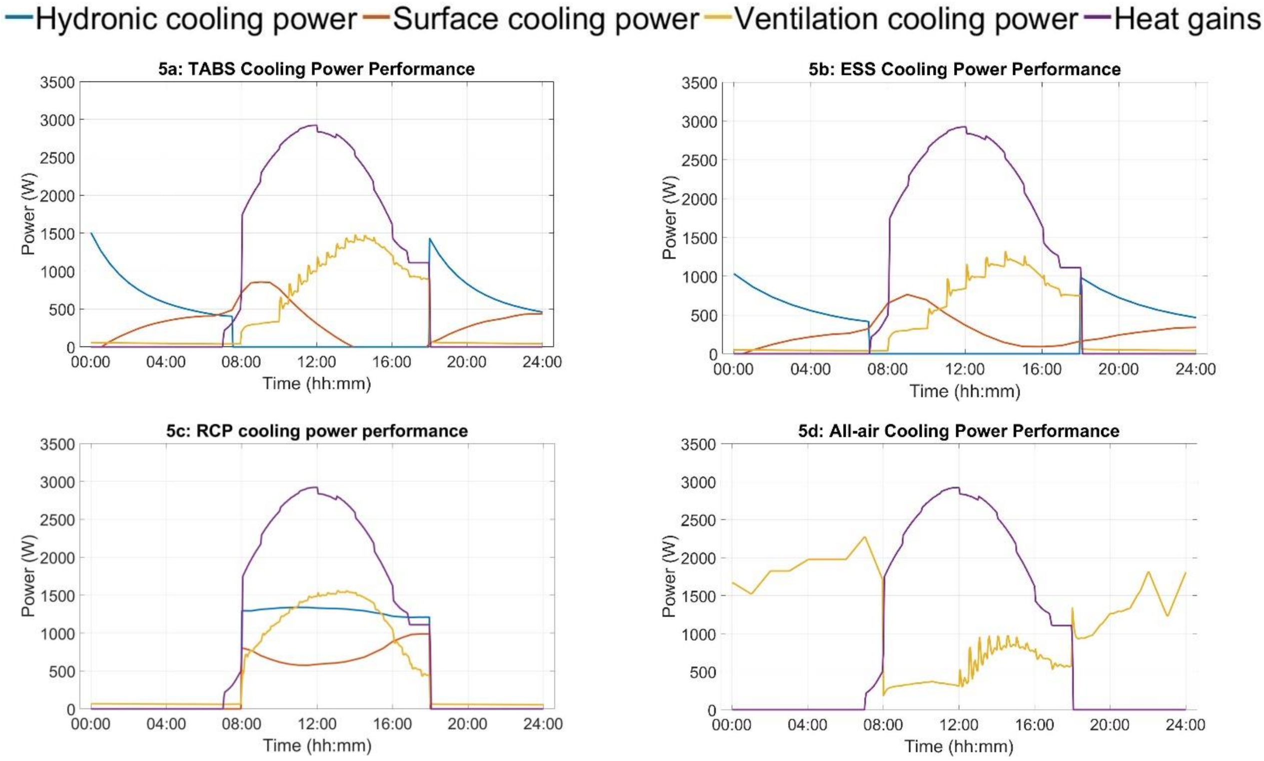

Figure 5: Cooling power performance of simulations with 70% cooling power

A further reduction in the maximum cooling power of TABS was achieved, since the radiant system delivered almost the same cooling energy with a lower peak power but in longer period. That is why insignificant differences are observed in the surface cooling power and the VAV system cooling power is almost the same as in the simulation with the maximum cooling power. The authors observed a similar trend on the hydronic side also for the ESS system. In contrast to TABS, in the case of ESS a significant reduction in the cooling power on the surface side was observed, and the cooling power of the VAV system increased to provide the designed operative temperature. As before, in the case of RCP the sum of the cooling power in the hydronic side and the VAV was comparable to the peak of the heat gains. all-air system, the cooling power during the occupancy period was identical to the previous case, while the peak cooling power during night-time was reduced substantially.

Figure 6 shows the cooling performance of the four systems examined when the authors sized the system based on the 50% of the maximum cooling.

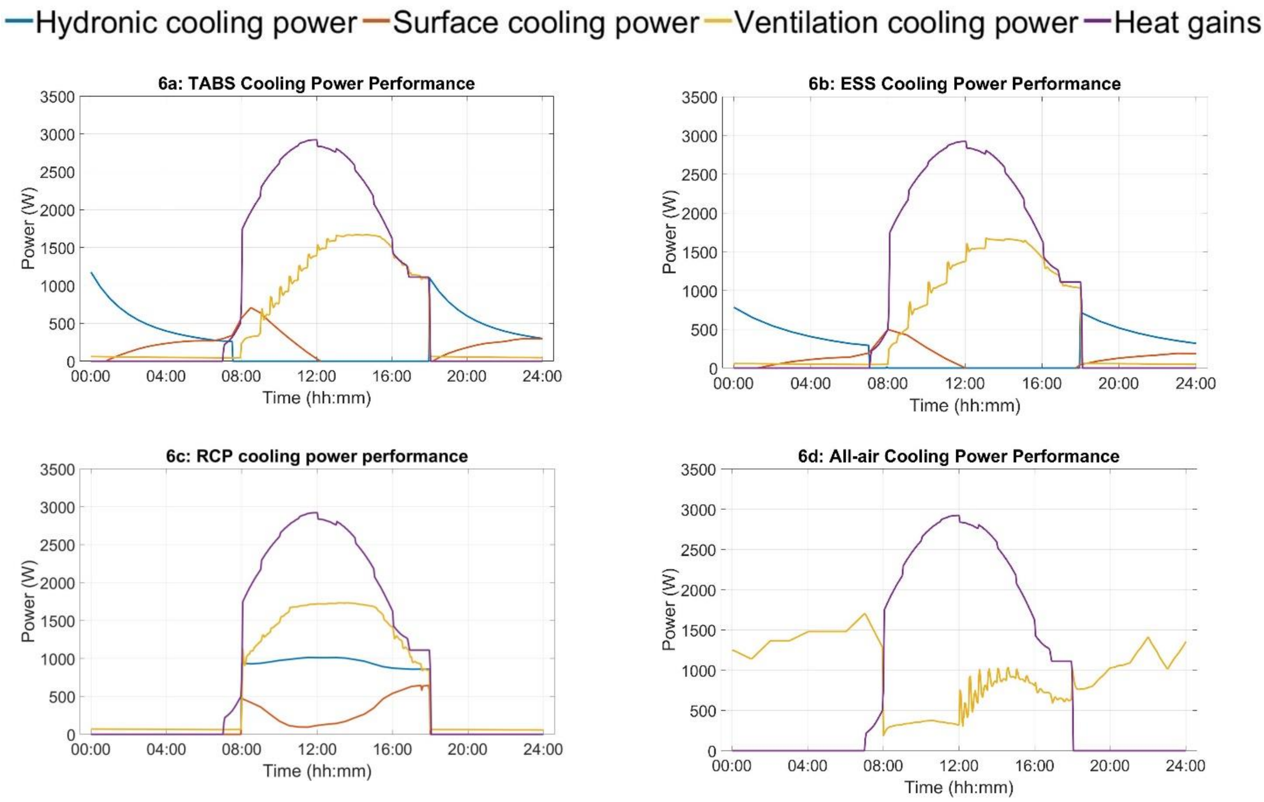

Figure 6: Cooling power performance for the simulations with 50% of the cooling power

A further reduction was observed in the hydronic cooling power of TABS, which was followed by a significant reduction on the surface side, which resulted in a considerable increase in the cooling power of the VAV system. In the case of ESS, the cooling power on the surface side is almost insignificant. This can be explained by Figure 3b, where the surface temperature is very close to the operative temperature, which results in very small ΔT for Equation 1. As before, in the case of the all-air system the cooling power changed insignificantly during the occupancy period and the peak during night-time was reduced substantially.

Table 4 shows the maximum cooling power that was calculated for each system. The maximum cooling demand was 2922 W, namely 60.9 W/m^2.

Table 4: Maximum cooling power of the examined systems

Cooling power	TABS, W		ESS, W		RCP, W		All-air, W*
	Radiant system	VAV system	Radiant system	VAV system	Radiant system	VAV system	
100%	2192	1500	1504	1146	1828	1519	3130
70%	1507	1478	1031	1319	1339	1561	2273
50%	1176	1670	783	1678	1013	1733	1701

* For the all-air system, the peak value was always obtained in the night when there is not the need of compression-based cooling. During daytime, the peak was around 1000 W

Table 5 shows the percentage of the maximum cooling power that each system delivered, compared to the maximum cooling load required. The underlined numbers indicate when the cooling power of the VAV system was higher than that of the hydronic system.

Table 5: Percentage of the maximum cooling power compared to the maximum cooling loads

Cooling power	TABS, %	ESS, %	RCP, %	All-air, %
100%	75	51	97	107
70%	52	<u>45</u>	<u>99</u>	78
50%	<u>57</u>	<u>57</u>	<u>94</u>	58

Cooling energy

Table 6 shows the cooling energy of the four simulated systems, namely the energy removed by the four systems. In the cases of the radiant systems, the values are the sum of the hydronic side and the VAV system.

Table 6: Cooling energy of the simulated systems

Cooling power	TABS, kWh	ESS, kWh	RCP, kWh	All-air, kWh
100%	20.3	18.4	24.4	36.5
70%	20.0	17.2	25.3	28.6
50%	20.0	19.6	24.9	23.6

Table 7 shows the percentage of the energy on the hydronic side that was removed on the surface side for the three radiant systems.

Table 7: Percentage of energy on the hydronic side removed on the surface side

Cooling power	TABS, %	ESS, %	RCP, %
100%	73	85	65
70%	69	79	56
50%	47	32	33

Discussion

TABS

The high thermal mass of TABS resulted in providing an acceptable thermal environment even when the system was sized based on the 50% of the maximum cooling power. TABS dealt with the base of the cooling loads while the VAV system, as a fast response system dealt with sudden variations in the heat gains as it can be seen from the small spikes in the VAV system curve in Figures 1a, 2a and 3a. Since TABS had a long response time, when the water pump was activated, a significant amount of time had to pass before a change was observed on the surface side, even when the maximum cooling power was used. Furthermore, the peak of the cooling power on the surface was significantly lower than the peak on the hydronic side for the whole period, since the system did not reach steady state. That difference was 68%, 44% and 40% for the simulation with 100%, 70% and 50% of the maximum cooling power, respectively. When TABS was sized based on the maximum cooling load, it deactivated several hours before the beginning of the occupancy period. That is a strong indication that the system was oversized.

ESS

When ESS was sized based on the 100% or 70% of maximum cooling load, a low air flow rate was enough to maintain the desired thermal environment. For that reason, air temperature and operative were equal for almost the whole occupancy period as Figure 1b shows. That effect deteriorated when the system was sized based on the 50% of the maximum cooling load. Similarly, to the case of TABS, there was a significant time lag between the hydronic and the surface heat flux. The peak cooling power on the surface side was 47%, 27% and 36% lower than the cooling power on the hydronic side for the simulation with 100%, 70% and 50% of the maximum cooling power, respectively. As in the case of TABS, ESS was considred oversized when it was operating based on the maximum cooling load.

RCP

In Figures 1c, 2c and 3c, RCP required the lower air flow rate out of the three radiant systems, since RCP was operating during the occupancy period. When sized based on the 50% of the maximum cooling demand, temperature was slightly above 26°C. Since both systems were operating at the same time, the peak cooling power was the sum of the two systems. Figures 4c, 5c and 6c show a significant difference in the cooling power in the hydronic and the surface side of the RCP. This is because TRNSYS provided one ceiling surface temperature as an output, taking into consideration both the fraction of the ceiling containing radiant cooling panels and the fraction that was not. That resulted in a higher surface temperature compared to the temperature of the surface of the RCP alone. As the maximum cooling power of the hydronic system was reduced, the cooling power of the VAV was increasing, to supplement the difference in the cooling power required to maintain the operative temperature at the desired temperature of 26°C.

All-air system

All-air system was able to maintain the operative temperature at the desired setpoint due to the night-time ventilation in all three examined cases. The air flow rate illustrated in Figures 1d, 2d and 3d during the occupancy period was identical, despite the reduction in the air flow rate during the night-time. That shows that when night-time ventilation is applicable, sizing an all-air system based on the maximum cooling demand results in oversized systems. Although the outdoor temperature was in favour of utilizing night-time ventilation in the simulated location, this is not always the case, and in some climates night-time ventilation is not applicable, unless the supply air temperature is conditioned by activating the cooling coil, which would increase the energy use of the air handling unit substantially.

Radiant systems cooling power

Table 5 shows that the systems with the highest thermal mass (TABS and ESS) achieved the highest percentage reduction in the cooling power. Nevertheless, when TABS was sized based on the 50% of the maximum cooling load, the VAV system was the system with the higher cooling power, between hydronic and VAV system. In the case of ESS, VAV had higher peak cooling power even when it was sized based on the 70% of the maximum cooling demand. Therefore, TABS and ESS alone were not sufficient for providing the desired thermal environment and the VAV system had a considerable contribution in removing the sensible heat gains. In the case of the RCP, the peak cooling power was comparable to the maximum cooling load in all three cases. As the cooling power of the RCP was getting lower the impact of the VAV was increasing to compensate for the lost cooling power of the radiant component.

Cooling energy

Cooling energy is the energy removed by the systems, so in the case of the radiant systems it is the sum of the cooling energy on the hydronic side and the VAV system. Cooling energy should not be confused with electrical energy use. Due to the outdoor conditions, some of the cooling energy could be "free", namely no refrigeration cycle would be required.

Table 6 shows that the cooling energy of TABS did not change significantly between the three cases. That was because as the cooling energy on the surface side was reducing, the cooling energy of the VAV was increasing. That was the case also for ESS and RCP, although the highest cooling energy was calculated when they were sized based on the 50% and the 70% of the maximum cooling load, respectively. When all-air system was sized based on the 100% or 70% of the maximum cooling load it had the highest cooling energy, while when it was sized based on the 50%, RCP had the highest cooling energy among the four simulated systems.

As Table 7 shows, for TABS and ESS there was a notable difference between the energy calculated on the hydronic and the surface side. That difference was the energy accumulated in the slab to cool it down. For that reason, it is very important to take Equation 2 into consideration when calculating the water flow rate. Going directly from Equation 1 to Equation 3 could possibly result in false water flow rate calculations. Table 7 shows a significant difference between the energy on the hydronic and the surface side for the RCP which was caused because TRNSYS provided only one temperature for the ceiling surface, as it was mentioned earlier.

Temperature control

Figure 1, 2 and 3 show that TABS and ESS had almost equal air and operative temperature during the occupancy period. They would be even closer if the heat gains were lower and a higher proportion of cooling load was handled by the radiant system. Therefore, for those systems an air temperature sensor would be adequate for the control of the thermal environment. On the other hand, in the cases of RCP and all-air system the two parameters varied substantially and thus, an operative temperature sensor should be used. As it is mentioned earlier, the authors controlled radiant systems based also on the surface temperature. This is a simplification compared to reality, where the surface temperature is difficult to measure and a temperature sensor is installed inside the slab instead. In that case, the time lag between the location of the sensor inside the slab and the surface should be taken into consideration, to avoid overcooling and discomfort due to too low surface temperature. Another factor that would affect the temperature control in reality, is the presence of furniture and other "obstacles" inside the conditioned space. In these simulations, the effect of furniture was not simulated.

The possibility of a new standard

As it was already known, radiant systems should operate with a ventilation system that would remove latent heat gains and pollutants and provide fresh air to the occupants. In this paper, the authors showed that the operation of the VAV system can be of high importance also for removing sensible heat gains, and it is a matter of combining the two systems for providing the desired thermal environment. For that reason, the publication of a holistic standard which provides the methodology for sizing both the radiant and the ventilation component of a complete cooling system should be taken into consideration.

Conclusions

When sizing a high thermal mass radiant cooling system, the effect of thermal inertia and the response time should be taken into consideration to avoid overcooling. The implementation of high thermal mass radiant systems shifted a significant amount of the cooling demand on the night-time. Same happened with the all-air systems that was able to cool down the building with night ventilation. Nocturnal operation for high thermal mass radiant cooling systems was sufficient to provide an acceptable thermal environment when there was a 50% sized VAV system operating during the day even when they were sized based on the 50% of the maximum cooling load, while RCPs should operate during the occupancy period, instead. As the cooling power of the radiant system was reduced, the dependence on the VAV system for providing supplementary cooling increased beyond the minimum ventilation required. Due to the utilization of night-time ventilation, all-air system was able to maintain the desired operative temperature even when it was sized based on 50% of the maximum cooling demand. Meaning that the 50% size all-air system was able to provide comfortable condition as the 50% sized massive radiant system couple with a 50% air system. The availability of free night cooling for the air and the radiant system depend on the outdoor climate.

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