Topic 8. Advanced or innovative HVAV&R systems and system components

# Comparison of Zone Cooling Load for Radiant and All-Air Conditioning Systems

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## SUMMARY

By actively cooling down one or more surfaces in a space, water-based embedded surface cooling (radiant) systems remove heat both by convection and radiation. Therefore, unlike the case of well-mixed air distribution systems where the cooling load is purely convective, the cooling load for radiant systems is comprised of both convective and radiant components. However, in current practice, the same design cooling load calculation methods for the radiant systems are used as the convection based air systems. The purpose of this energy simulation study was to investigate if there are differences in peak zone cooling load and 24-hour total zone cooling energy generated by all-air distribution systems in comparison to radiant cooling systems. Sensitivity studies were performed for different load and boundary conditions. Simulation results show that total zone cooling energy for the radiant cooling systems studied can be 2.7-6.5% higher than the all-air systems, and peak zone cooling load can be in the range of 10-40% higher depending on the load conditions. Future research is needed to assess the implications for design guidelines. In general, even if the total zone level cooling energy may be slightly higher for the radiant systems, there are verified advantages of using hydronic-based radiant systems such as improved plant side equipment efficiency with warmer cold water temperature, possibility of night pre-cooling and utilization of natural cooling resource, and energy efficiency in transporting heat with water compared to air. All of these factors combine to produce better overall energy performance in radiant cooling systems.

## INTRODUCTION

Water-based embedded surface cooling systems are gaining popularity as an energy efficient strategy for conditioning buildings (Olesen 2012). By actively cooling down one or more surfaces in a space, these systems remove heat by both convection and radiation. The presence of activated radiant surface(s) may have different impacts on building heat transfer dynamics. First, in a room with an active cool surface, the inside temperature of the non-active building walls may be lower than the walls when the zone is conditioned by convection based all-air system. This is important due to the fact that wall surface temperature is directly related to the conductive heat transmission through the building envelope, and therefore the total cooling energy. Second, the activated radiant cooling surface(s) removes radiant load (e.g. solar, radiant internal load) that is directly impinging on it and at the same time extracts heat from the non-activated building mass that is used to serve as thermal storage in

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convection-based air systems. This phenomenon may change the cooling load profile for the mechanical systems. These two impacts are important because they may directly influence the equipment design at zone and plant level, as well as the system operational strategies.

For the impact on wall surface temperature, there are several research studies on the radiant heating cases (Howell 1987; Chapman, Rutler et al. 2000). However, both studies focused on heating calculation under steady-state conditions, and the role of instantaneous heat gain and building thermal mass, which is particularly important in cooling load calculation, were not considered. Chen (1990) suggested that the heating load of a ceiling radiant heating system was 17% higher than that of the warm air heating system because of the role of thermal mass and higher heat loss through the building envelope due to slightly higher inside surface temperatures. Despite the potential difference in load calculations for radiant and air systems, in current radiant system design guidelines, such impacts are not considered or evaluated (CEN 2005; ISO 2012).

The impacts of activated radiant cooling surface(s) on the heat transfer dynamics of a space and the resulting cooling load profile has been reported in several papers. Babiak et al. (2007) suggested that the cooling capacity of a chilled radiant floor can be doubled when illuminated by direct solar radiation. Causone et al. (2010) investigated the difference in cooling load calculation for radiant cooling systems compared to an all-air system with the presence of solar load and developed a method to take it into account in cooling load calculation. But to the authors’s knowledge no comprehensive studies have been conducted regarding the interaction of the cooling surface(s) with non-activated building mass and radiant heat gain, and implications for cooling load profile and peak cooling load value.

The purpose of this study was to compare the cooling load profiles for radiant systems and convection based air systems, especially the peak cooling load value and the 24-hour total cooling energy.

## BACKGROUND

ISO 11855 (2012) has categorized the water based embedded surface cooling systems depending on system construction (pipe spacing, length, thermal resistance, etc) and construction of floor/wall/ceiling (finishing, number of active region, etc). Design and dimensioning of the systems is based on system cooling capacity that is characterized by the heat flux between the cooling surface(s) and the space at design conditions, which depends only on the cooling surface temperature and space conditions for a given type of surface (wall, ceiling, floor) and is independent of the type of embedded systems(heavyweight, lightweight). The cooling load is the rate at which sensible and latent heat must be removed from the space to maintain a design condition of temperature and humidity (ASHRAE 2009). Assuming there is no supplementary conditioning system for the space, the goal for designing and sizing a surface cooling system is to ensure the system cooling capacity can meet a design cooling load calculated by standardized methods, usually the peak design cooling load. The cooling surface heat flux is, however, different from the waterside load, which is the heat extraction rate based on an energy balance of the hydronic circuit. The differences depend on the specifications of the surface cooling systems, space load conditions, and operation strategy. The hydronic circuit load is important for sizing of waterside equipment, such as pumps, chillers and cooling tower, etc. Therefore, the heat flows at both the cooling surface level and hydronic level require careful investigation.

For heavyweight surface cooling systems, for example the thermo active building systems (TABS) where intermittent or nighttime operation is often implemented, the required cooling capacity for the system should not be directly evaluated based on the calculated peak-cooling load; instead the system should be sized based on the 24-hour accumulated design heat load, internal load pattern, hydronic loop operation schedule, as well as radiant system specifications (ISO 2012). Therefore, understanding both peak cooling load and 24-hour accumulated heat load is of significant importance for surface cooling system design.

## METHODS

EnergyPlus v7.0 was used for the simulation study because it performs a fundamental heat balance on all surfaces in the space and has been extensively validated (DOE 2012). Also since it is able to integrate the heat transfer calculation in the water-based embedded cooling systems with the changing space conditions, it is able to capture the transient behavior of the systems (DOE 2011). Two single zone models, one conditioned by an all-air system and one by a water-based embedded cooling system (radiant system) were developed for cooling energy and cooling load profile comparisons for the 99.6% cooling design day.

### Simulation Runs

Seven simulation runs were conducted in total, and they were grouped into three groups based on the test purpose (Table 1). To ensure equivalent comfort conditions between the two systems, all simulations of the air system were controlled to closely track the hourly operative temperature profile derived from the radiant system simulation for the identical input conditions.

*Table 1: Simulation runs summary*

| Case | Building | Int. load[1] | Window | Boundary conditions[3] |
|---|---|---|---|---|
| 1 | HW[2] | No | No | EW=Envir |
| 1a | HW_SmallR[2] | No | No | EW=Envir |
| 1b | LW[2] | No | No | EW=Envir |
| 2 | HW | Conv.: 1.0 | No | All=Adb |
| 2a | HW | Rad: 0.6,Conv.: 0.4 | No | All Adb |
| 2b | HW | Rad: 1.0 | No | All Adb |
| 3 | HW | No | Yes | EW=Envir |

Note:

1. Int. load= Internal load;
2. HW=Heavy weight, HW_SmallR=Heavy weight construction with half thermal insulation at exterior walls; LW= light weight construction, but similar thermal resistance with the HW case.
3. EW=exterior wall, Adb=adiabatic, Envir=outdoor environment; Conv=convective heat gain, Rad: radiant heat gain; Both roof and floor have boundary conditions set to adiabatic for simplicity

Group 1 was used to investigate the influence of wall transmittance (U-value) and thermal mass on both total cooling energy and peak cooling load. Case 1 serves as a baseline comparison. The purpose of Group 2 is to evaluate the impacts of internal load on both total cooling energy and peak cooling rate. For these cases, the building envelope is set to be adiabatic, creating an interior zone. Group 3 introduces two real windows on the south side exterior wall such that the impact of solar gain can be evaluated.

**Model Specifications**

The single zone model is developed primarily based on ASHRAE Standard 140 (ASHRAE 2007). System and design parameters for the radiant system are adopted from RADTEST (Achermann and Zweifel 2003). Additional details are summarized below.

<u>Building geometry</u>: The test case (Figure 1) is a rectangular single zone (8 m wide × 6 m long × 2.7 m high) with no interior partitions. The building is of heavy weight construction with characteristics as described below. Both the floor and roof boundary conditions are set to adiabatic to simplify the analysis. Only case 3 has 12 $m^2$ of south-facing windows. The overall U-Factor is 2.721 $W/m^2K$ with Glass SHGC at 0.788. <u>Weather:</u> the standard Denver weather data provided by ASHRAE 140 is used. <u>Material Specifications</u>: The baseline construction is based on case 900 (heavyweight), except that the roof construction has been modified so that water tubes can be embedded in the concrete layer. For case 1b with lightweight configuration, exterior wall and floor construction are based on Table 1 of section 5, Standard 140, and roof is exactly the same as heavy weight. <u>Internal heat gain and infiltration</u>: For group 2, the internal gain is 720 W from 6:00 to 18:00. The radiant and convective split is different for each run as specified in Table 1. There is zero air infiltration for all runs. <u>Radiant model:</u> The radiant system design specification is based on RADTEST case 2800. As for control, the goal is to maintain operative temperature setpoint at 22°C for 24 hours with a 2°C deadband (DOE 2011). <u>Air system</u>: “IdealLoadsAirSystem” EnergyPlus object is used to ensure that the same operative temperature as the corresponding radiant system and zone design condition can be maintained.

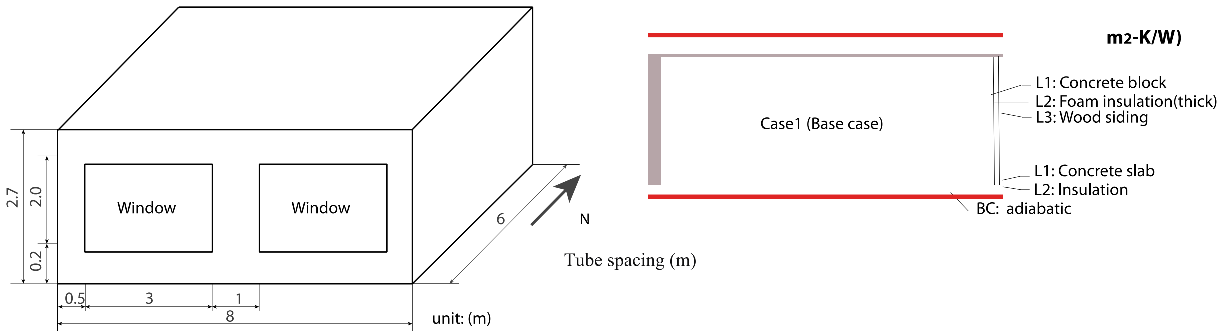


*Figure 1: Base case modeling specifications (Only Case 3 has windows)*

## PARAMETERS INVESTIGATED

For an all-air system, the sensible cooling load is calculated by equation (1),

$$\dot{Q}_{air} = \dot{m}_{air} C_{p,zone} (T_{zone} - T_{sa}) \tag{1}$$

where $\dot{m}_{air}$ is the zone air mass flow rate, $kg/s$; $C_{p,zone}$ is the specific heat of zone air, $J/kgK$; $T_{zone}$ is zone air temperature in °C; and $T_{sa}$ is supply air temperature , °C.

For the radiant cooling system, the cooling surface heat transfer rate can be calculated as a combination of convection and radiation, and equation (2) governs the heat transfer process at the surface level (DOE 2011),

$$\dot{Q}_{surf} = \dot{Q}_{conv} + \dot{Q}_{rad} = -\dot{Q}_{cond} \tag{2}$$

where, $\dot{Q}_{cond}$, $\dot{Q}_{conv}$, $\dot{Q}_{rad}$ are the conduction, convection and radiation heat transfer rate at the inside face of the cooling surface(s), respectively, W. A positive value means that heat flows into the inside face of the surface, and a negative value means heat leaves the inside face of the surface. EnergyPlus predicts conduction and convection heat transfer rates at inside wall surfaces, and the total surface-cooling load is calculated as the negative of the conduction heat transfer rate.

Hydronic cooling load can be calculated by equation (3) (DOE 2011),

$$\dot{Q}_{water} = (\dot{m}C_p)_{water}(T_{wi} - T_{wo}) \tag{3}$$

where, $\dot{m}$ is mass flow rate of water, $kg/s$; $C_p$ is the specific heat of water $J/kgK$; $T_{wi}, T_{wo}$ are the supply and return water temperature, respectively, °C. Table 2 lists the parameters to be studied for the simulation.

*Table 2: Parameters analyzed*

| | Peak cooling rate ($W/m^2$) | Total cooling energy ($kWh/m^2$) |
|---|---|---|
| Air system | Peak sensible cooling load (AirPeak) | 24 hour total sensible cooling energy (AirTotal) |
| Radiant system | Peak surface cooling load (RadSurfPeak) | 24 hour surface peak cooling energy (RadSurfTotal) |
| | Peak hydronic cooling load (RadHydPeak) | 24 hour total hydronic cooling energy (RadHydTotal) |

**RESULTS**

Simulation results for the summer design day are reported in Tables 3-5. Table 3 compares the heat transmission through building exterior walls and windows for Groups 1 and 3, and Table 4 shows the resulting 24-hour cooling energy and the percentage differences. Table 5 summarizes the peak cooling rates for the two conditioning systems.

*Table 3: Comparison of 24-hour heat transmission through building envelope*

| Case | Summation of heat conduction at inside face of exterior walls ($Wh/m^2$) | | | Net heat gain through windows ($Wh/m^2$) | | |
|---|---|---|---|---|---|---|
| | Air | Radiant | % Difference | Air | Radiant | % Difference |
| 1 | 133.6 | 138.8 | 3.8% | NA | NA | NA |
| 1a | 207.9 | 221.8 | 6.3% | NA | NA | NA |
| 1b | 143.2 | 148.7 | 3.7% | NA | NA | NA |
| 3 | 70.0 | 82.7 | 15.4% | 426.1 | 430.2 | 1.0% |

Note: Group 2 cases have adiabatic boundary conditions, therefore, no heat transmission through building envelope

*Table 4: 24 hour total cooling energy comparison for summer design day*

| Cases | AirTotal ($Wh/\mathrm{m}^2$) | RadSurfTotal ($Wh/\mathrm{m}^2$) | RadHydTotal ($Wh/\mathrm{m}^2$) | RadSurfTotal VS. AirTotal | RadHydTotal VS. AirTotal |
|---|---|---|---|---|---|
| Case 1 | 133.8 | 138.8 | 140.5 | 3.8% | 4.7% |
| Case 1a | 209.9 | 221.9 | 224.5 | 5.4% | 6.5% |
| Case 1b | 143.3 | 148.8 | 150.5 | 3.7% | 4.8% |
| Case 2 | 181.0 | 179.5 | 180.9 | -0.83% | -0.05% |
| Case 2a | 180.8 | 180.2 | 183.0 | -0.30% | 1.23% |
| Case 2b | 181.5 | 180.0 | 182.1 | -0.86% | 0.32% |
| Case 3 | 505.8 | 520.0 | 526.3 | 2.7% | 3.9% |

*Table 5: Peak cooling rate comparison for summer design day*

| Case | AirPeak ($\mathrm{W/m^2}$) | RadSurfPeak ($\mathrm{W/m^2}$) | RadHydPeak ($\mathrm{W/m^2}$) | RadSurfPeak vs AirPeak | RadHydPeak vs AirPeak |
|---|---|---|---|---|---|
| 1 | 7.28 | 9.06 | 9.89 | 19.7% | 26.4% |
| 1a | 11.22 | 14.09 | 14.67 | 20.3% | 23.5% |
| 1b | 11.87 | 15.20 | 15.68 | 21.9% | 24.3% |
| 2 | 12.84 | 14.09 | 19.22 | 8.9% | 33.2% |
| 2a | 11.12 | 13.12 | 16.11 | 15.5% | 31.0% |
| 2b | 10.01 | 12.23 | 12.77 | 18.2% | 21.6% |
| 3 | 23.74 | 39.32 | 29.13 | 39.6% | 18.5% |

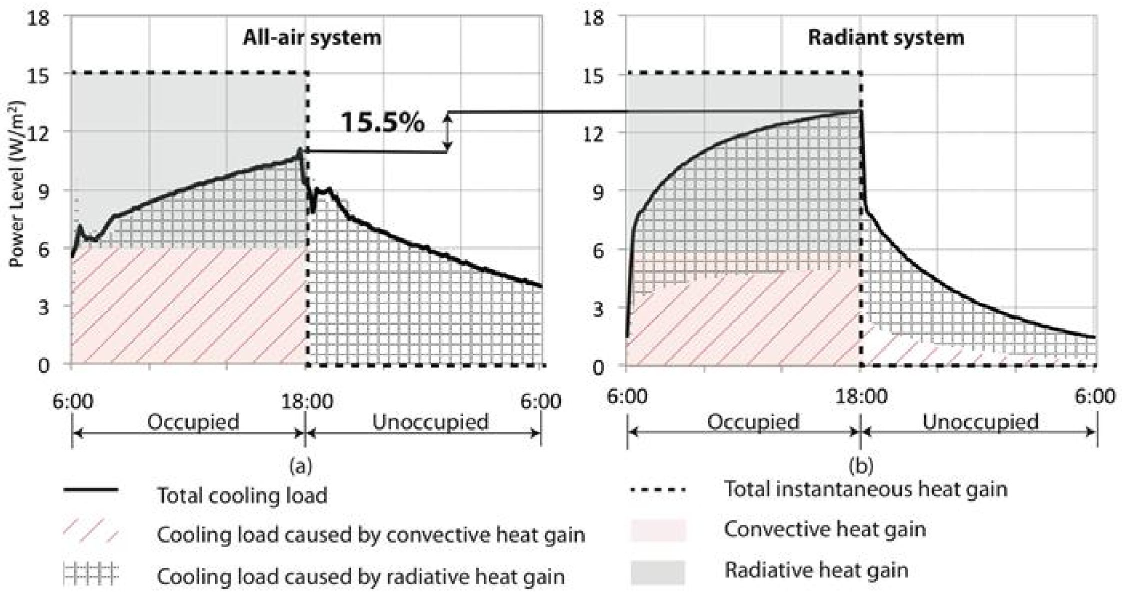


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*Figure 2: Comparison of cooling load and heat transfer breakdown for Case 2a: (a) all-air system and (b) radiant system (surface cooling)*

Figure 2 shows an example (Case 2a) of the cooling load profiles for the all-air and radiant systems. The figure compares the processes of how radiative and convective heat gains are converted into cooling load for the two systems. For Case 2a, the total internal heat gain (15 W/m$^2$ during occupied hours) were divided into convective heat gain (6 W/m$^2$) and radiative heat gain (9 W/m$^2$). The solid black lines are hourly cooling loads, which reach their peak value at the end of the occupied period for both systems. These predicted cooling loads represent the total amount of heat being removed by each system to maintain the same operative temperature profile. Note that the peak cooling load for the radiant system is predicted to be 15.5% greater than that for the all-air system. For the all-air system, the cooling load is 100% convective, while for the radiant system, the cooling load represents the total heat removed at the activated ceiling surface, which includes incident radiant loads, longwave radiation with non-activated zone surfaces and convective heat exchange with the warmer room air. As shown in Figure 2, the cooling load for both systems is composed of two components, one that originates as convective heat gain from internal loads, and one that originates as radiative heat gain from internal loads. In the case of the all-air system, cooling load matches the convective gain at the beginning of the occupied period and then ramps up during the day as radiative gains are absorbed by zone surfaces and re-released as convective load. The fact that building mass delays and dampens the instantaneous heat gain is well recognized by cooling load calculation methods. For the radiant system (Figure 2b), the strong influence of the active chilled ceiling is demonstrated by the large portion of total cooling load consisting of radiative heat gains. The convective gains actually contribute a smaller amount compared to the all-air system because a higher zone air temperature is reached to balance the cooler ceiling surface temperature, thereby maintaining an equivalent operative temperature.

## DISCUSSION

Based on Tables 4 and 5, the radiant hydronic loop 24 hours total energy is always higher than the load at the surface level. The difference is the energy used to cool the mass of the slab itself with adiabatic boundary condition for the roof. Compared to surface cooling rate, the hydronic loop cooling rates are delayed by the slab thermal mass. These observations are consistent with the general understanding of water-embedded cooling systems.

For Group 1, Table 3 shows higher conductive heat transfer through the exterior wall for the radiant system. The reason for this finding was the lower surface temperature (about 0.5 °C) at the inside face of the walls caused by the radiant system. The difference in conduction heat transfer explains why total daily cooling energy is 3.6-5.4% higher at surface level and 4.7-6.5% higher at hydronic level than the all-air system. Comparing Cases 1 and 1a, which have less thermal insulation, the percentage difference in total cooling energy increases from 3.6% to 5.4%, and for Case 1b, the change in total cooling energy from Case 1 is negligible. Therefore, even though thermal insulation affects the difference in total cooling energy between all-air and radiant systems, building mass does not affect the difference. Group 2 cases have adiabatic boundary conditions, and therefore, have no difference in total cooling energy. For Group 3, the total hydronic energy is 3.9% higher. Different heat gain through the building envelope is the only reason for the higher total cooling energy in radiant systems.

Table 5 shows that the peak radiant system cooling loads (both hydronic and surface) exceed that of the air system in all cases by 8.7% to 48.7% depending on load conditions. For the radiant cooling system, as shown in Figure 2(b), a large portion of the radiative heat gain converts to cooling load during the occupied period due to the presence of the cooling surface(s). Even with part of the convective heat gain shifted to night time, the peak cooling load is higher than the convection based air system depending on load conditions.

For Group 1, the radiant system demonstrates higher cooling rates compared to the all-air system in all cases. Surface peak cooling rate changes from 19.7% to 21.9% higher, and hydronic peak cooling rate ranges from 23.5% to 26.4%. Thermal mass does not have much impact on the difference in peak cooling rate. For Group 2, where the total internal loads are the same for all cases but with different radiant and convective splits for each case, the peak cooling rate differences range from 8.9% to 18.2% at the surface level and from 21% to 33% at the hydronic level. This implies that higher radiant/convective ratio in heat gain produces larger differences in peak loads between the two systems at the surface level. This is further demonstrated in Group 3, where solar gain causes an increase in the radiation/convection ratio of the load on the radiant surface(s). Differences in surface peak cooling rate increase to about 40% higher than the all-air system. More detailed discussions of this point will be described in future papers. While the high peak-cooling rate in Case 3 is regarded as enhancing the cooling capacity of the radiant cooling system when the space is exposed to solar gain, sizing of the waterside equipment must take this into account.

## CONCLUSIONS

For the simulated cases, total zone level cooling energy removed at the activated surface and by the hydronic loop are 2.7-5.4% and 3.9-6.5% higher, respectively, compared to the all-air system when equivalent comfort conditions (operative temperature) are maintained. This is due to the lower surface temperature at the inside face of the building envelope created by the active (cooled) radiant surface. Peak cooling load at the active radiant ceiling surface can be 8.9-39.6% higher than the air system depending on the load conditions. For interior spaces, the peak cooling rate differences range from 8.9% to 18.2% at the surface level and from 21.6% to 33.2% at the hydronic level. For perimeter zones or atrium spaces with direct solar gain the peak cooling load can be up to 40% higher.

In general, even if the total zone level cooling energy may be 2.7-5.4% higher for the radiant systems, there are verified advantages of using hydronic-based radiant systems such as improved plant side equipment efficiency with warmer cold water temperature, possibility of night pre-cooling and utilization of natural cooling resource, and energy efficiency in transporting heat with water compared to air. All of these factors combine to produce better overall energy performance in radiant cooling systems (Babiak, Olesen et al. 2007).

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