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Arghand, T., Dalenbäck, J., Trüschel, A. et al (2019). Some aspects of controlling radiant and convective cooling systems. E3S Web of Conferences, 111.  
<http://dx.doi.org/10.1051/e3sconf/201911105008>

N.B. When citing this work, cite the original published paper.

# Some aspects of controlling radiant and convective cooling systems

Taha Arghand<sup>1,\*</sup>, Jan-Olof Dalenbäck<sup>1</sup>, Anders Trüschel<sup>1</sup>, Saqib Javed<sup>1</sup>

<sup>1</sup>Chalmers University of Technology, Department of Architecture and Civil Engineering, 412 96 Gothenburg, Sweden

**Abstract.** Designing appropriate control systems for radiant heating and cooling terminals entails an understanding of their dynamic behaviour. This study experimentally investigates the dynamic response of a room with convective and radiant cooling systems. The experiments were performed in a 12.6 m<sup>2</sup> large test room outfitted as a single-office room. The main cooling system was radiant ceiling panels which covered 70% of the ceiling area. The thermal performance of the radiant system was compared to that of a fan-coil unit (FCU). The results from the step response test showed that the time constant of the room for the radiant system was shorter than for the convective one, indicating faster changes in room temperature by the radiant system. Furthermore, controlling the FCU with similar control system tuned for ceiling panels increased the hysteresis gap in the room air temperature from 0.4 K to 0.8 K. This indicates that control systems for low-mass radiant systems and convective systems might be applied to each other, but on-site tuning is required to omit the offset (persistent error). In this study, controlling room temperature with ceiling panels did not benefit from using an operative temperature sensor to provide feedback signal to the control system. However, the pump energy use was moderately decreased by 14%.

## 1. Introduction

The application of ceiling cooling panels as a part of building cooling systems has been investigated in many studies and its advantages regarding energy use of the system, thermal comfort, and utilizing free-cooling sources have been reported [1–5].

Designing control systems for ceiling cooling panels involves considering condensation prevention measures as well as proper cooling capacity control methods to maintain the room temperature. Over the past years, a fairly large body of literature has been developed on the preventive control methods to avoid condensation from forming [1,6,7]. However, little has been done to understand the cooling capacity control of the ceiling panels.

The very initial step to design a control system for a room terminal unit is to know the steady-state and transient thermal behavior of that terminal. The steady-state thermal behavior of a terminal usually deals with its overall cooling capacity when the input parameters are constant. But the transient behavior is mainly defined by how fast the cooling capacity of the terminal can change to keep the room temperature within the defined limits. This requires analyzing the response time of the system.

Not many studies have investigated the response of the ceiling panels. Jeong and Mumma [8] indicated that the response time of the ceiling panels is very short (<5 min). But this assumption was supported with no

reference or experimental results. Ning et al. [9] simulated the response time of different radiant cooling systems and classified the ceiling panels as a quick response system, with an average response time of 4 minutes. The criterion used to evaluate the response time of the radiant systems was the time taken for the surface temperature of the radiant system to reach to the new surface temperature when a change in control system was applied. This definition of response time is only focused on the response of the terminal, but the interaction of the internal heat gains and the cooling system is neglected in this way. When it comes to the design of the control system, considering this interaction is of especial importance. Convective cooling systems typically aim to maintain an air temperature set-point using a two-state feedback control algorithm, often with a small hysteresis gap between the value that initiates an "off" state and the value that initiates an "on" state. For a first-order system, this algorithm necessarily results in an oscillating saw-tooth shaped error between the set-point and the actual air temperature. ASHRAE-Systems and Equipment and ISO 18566 suggested controlling low-mass radiant systems, e.g. ceiling cooling panels, with traditional control technologies commonly applied to hydronic convective cooling systems [10,11]. These control strategies generally involve adjusting supply water temperature or flow rate with indoor feedback control. In addition, to improve the occupants' comfort,

\* Corresponding author: [arghand@chalmers.se](mailto:arghand@chalmers.se)

it is recommended to use operative temperature as the controlled variable.

However, our hypothesis is that with the same algorithm and the same hysteresis gap, the amplitude of the oscillating error for a radiant cooling system would be larger than an air system, whereas the cycling frequency would be smaller. This would cause longer on-off periods. The hypothesis is based on the expectation that when a controller initiates an "on" state, the room air temperature will change more rapidly for an air system than for a radiant system. This is in-line with the findings in other studies that convective- and radiant-based cooling terminals serve the room differently in terms of heat extraction rate and its magnitude [12–14]. Thus, designing a control system for ceiling cooling panels may require different considerations.

This study experimentally investigates the time response of a test room with a ceiling cooling panel system and analyze it in comparison with the response of the same test room with a convective cooling system. In addition, the influence of using operative temperature as a controlled variable for controlling the cooling capacity of the ceiling panel system and pump energy use is studied.

## 2. Definition of the time constant

Dynamic behavior of a linear first-order system is characterized by its time constant ( $\tau$ ). Generally, the time constant for first-order systems is the time taken for the output to reach 63.2% of its final value after a step change has been applied [15]. A cooling system consists of various components, each of which has different dynamic characteristics, and so different time constants. For instance, the time constant for temperature sensors[16], radiant terminals [9], building envelope[17], and control components have been documented in different studies. Our hypothesis is that radiant and convective cooling systems treat the heat gains differently. Furthermore, considering the experimental setup in this study, the time constant of the non-active room components, i.e. walls, floor and ceiling which are not involved in the heat exchange process, can be expected to be so long that the time constant of other components in the cooling system can be assumed negligible. The time constant in this study is defined as the time when the room air/operative temperature reaches approximately 63% of its total changes between two steady-state conditions. Changing of the room temperature at different rates, for example, due to using different terminal units in the room, would result in different room time constants, which, in turn, would change the proportional gain of the control system.

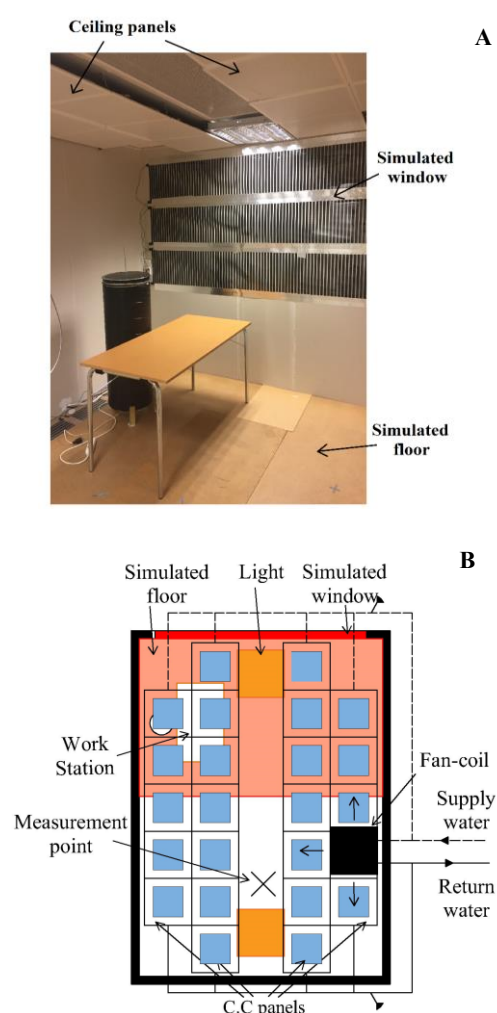
## 3. Methods

### 3.1. Experimental facility

The study was carried out in a test room with the dimensions of 4.2 m × 3.0 m × 2.4 m (L× W× H) and

outfitted as a typical office room. The test room was made up of polystyrene panel walls with gypsum board finish and light-weight compressed glass wool ceiling panels with an external layer of glass wool insulation. The floor was made of 30 mm expanded polystyrene and 6 mm of plasterboard and a layer of 22 mm of fiberboard finishing, to minimize the heat loss through the floor. The test room was located in a large lab hall, in which the temperature was constant at about 21 °C during the experiment period. Therefore, the heat loss through the test room envelope was insignificant.

Internal heat sources in the test room included a thermal dummy, lights, and simulated warm floor and window. The simulated floor and window were representing the effect of solar heat gain in the room and were made up of electrical heating foils, see Figure 1.



**Figure 1.** A) Photo of the test room, B) schematic of the experimental setup in the test room

Two cooling systems were installed in the test room. The primary cooling system used for the experiments was ceiling cooling panels. The second cooling system was a cassette-type fan coil unit (FCU). The thermal performance of the FCU was compared with the thermal performance of the ceiling panels. The ceiling panels comprised of copper pipes, embedded into a graphite-based material with high thermal conductivity, with

topside insulation [18]. The ceiling panels were hung 0.20 m below the main ceiling. Each panel was 0.60 m × 0.60 m in size and on whole the panels covered approximately 70% of the ceiling area. The ceiling panels were connected in four parallel loops, with each loop containing 5-7 panels in series, as shown in Figure 1.

The room air temperature was measured using a probe-type PT-100 sensor located at the height of 1.10 m above the floor, see Figure 1. The accuracy (bias) of the instrument was  $\pm (0.1 + 0.0017 \times T)$  °C. The time constant of the air temperature sensor was 3 sec. The operative temperature was also measured at the same location and height with another PT-100 sensor which had its measuring section placed inside a gray-painted Ping-Pong ball [19].

### 3.2. Experimental approach for the dynamic response test

Room terminals often operate under non-steady-state, also known as dynamic thermal conditions, due to changes in the room heat balance. The dynamic condition has three time-frames: the beginning of the change, the transient state between the two steady-state conditions, and the final state where the output settles. Designing an ideal controller requires information about all these three time-frames. A dynamic response test provides this information.

The dynamic response test with the ceiling panels was accomplished by conducting a step response test. To provide steady-state room temperature in the test room, water supply temperature and flow rate to the panels, and internal heat gains were kept constant for approximately 12 hours prior to the experiment. The water flow rate and supply water temperature prior and during the test were  $4.8 \pm 0.1$  l/min and  $15.2 \pm 0.3$  °C, respectively. The step change tests carried out by increasing the internal heat grains. The basic internal heat gain of the room was 450 W (36 W/m<sup>2</sup>). Then, the step increase in the heat load of 300 W was added to the basic room heat load to reach a total heat load of 750 W (60 W/m<sup>2</sup>). This step increase represents the heat dissipated by 4 people. The increase in the heat gain was performed using floor electrical foil. The measurements continued until the new steady-state room temperature was reached. The step response test was performed using the FCU and the ceiling panels under the same experimental conditions.

### 3.3 Experimental condition for periodic internal heat gain tests

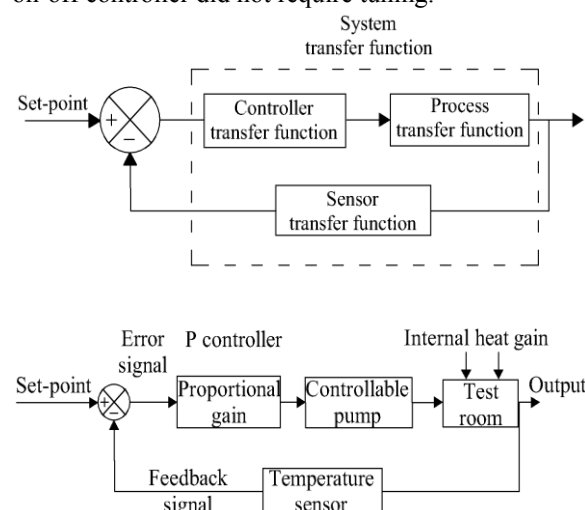
The objective of performing periodic internal heat gains was to test the performance of the control system from the standpoint of maintaining room temperature under transient heat gain condition. To simulate the transient heat gain condition in the test room, some part of the internal heat production was intermittently generated. The heat load was 200 W (16 W/m<sup>2</sup>) and 700 W (55 W/m<sup>2</sup>) during the “low” and “high” heat gain cycles,

respectively. The duration of each cycle was 120 minutes.

The cooling capacity of the panels was controlled by water flow control method. Depending on the test, the control method was implemented through either P controller (modulating), or on-off controller (two-position). To investigate any differences in control of ceiling panels and FCU systems. P controller was employed in the control system. The results are presented in section 4.2.1. The on-off controller was used to investigate the effect of sensor type in controlling the cooling capacity of the ceiling panels. The results of this part is discussed in section 4.2.2. Only one controller was applied at the same time. The set-point for air and operative temperatures in the periodic internal heat gain experiments was 25.0 °C.

The P controller adjusted the pump speed to provide appropriate water flow rate based on the difference between the set-point and the measured room temperature. This difference, called the control error, was then multiplied by the proportional gain to adjust the pump speed, see Figure 2. The proportional gain for a controller can be obtained through different tuning methods, such as step response method. The proportional gain, in fact, determines the responsiveness of the system and controls to regulate the feedback signal to appropriately run the pump/valve to reach the set-point, as shown in Figure 2.

The on-off controller simply drove the pump relative to the set-point at either “on” or “off” mode. If the measured room air/operative temperature rose above the set-point, the pump started circulating the water. The pump continued operating until the room air temperature dropped below the set-point. It is worth noting that the on-off controller did not require tuning.



**Figure 2.** Transfer function (top) and block diagram (bottom) of a typical feedback control system with P controller

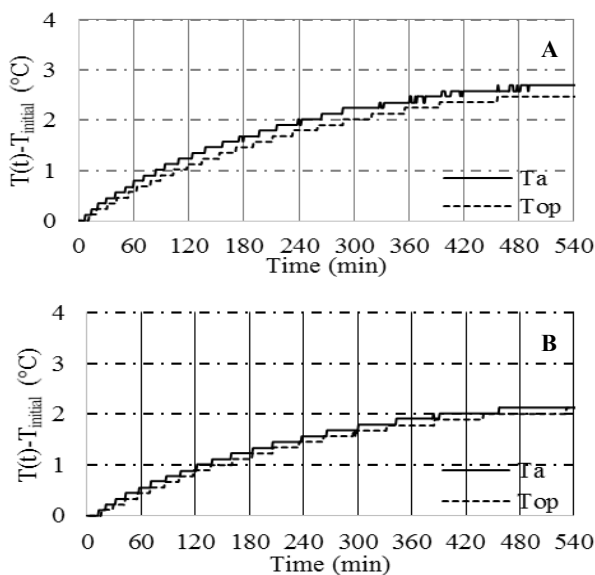
## 4. Results

### 4.1. Dynamic response test

Usually, the temperature of a conditioned space in a building is not constant. This instability comes from

changes in the heat balance of the conditioned space with time. To understand how a cooling terminal unit copes with variations in internal heat gain, performing dynamic response test on the cooling system is necessary. This section deals with the dynamic analysis and time response of the ceiling panels and the FCU.

Figure 3 shows the net rise in room air and operative temperatures due to a step increase in internal heat gain with ceiling panels and FCU. The net temperature rise is described as the difference between the initial and the instantaneous room temperature at a given time. With both terminals, increase in room temperature started with a short delay after applying the step change. It then kept increasing until new steady-state condition prevailed in the room. Room temperature trajectory shown in Figure 3 followed the natural logarithm form, which is the typical response form of first-order systems.



**Figure 3.** Net rise in air and operative temperatures as the response to the step increase in internal heat gain of the test room cooled by A) ceiling panels, and B) FCU

In Table 1, time constants of the room with FCU are higher than that for ceiling panels. Interpreting this result in the control context suggests that the proportional gain of the controller would not be similar for the two systems. In practice, using a similar controller for both terminals resulted in longer transient time to reach steady-state condition with FCU. In addition,  $\tau$  of operative temperature for both terminals was moderately higher than that of air temperature. This means that applying operative temperature as the feedback signal involves prolonging the response of the system.

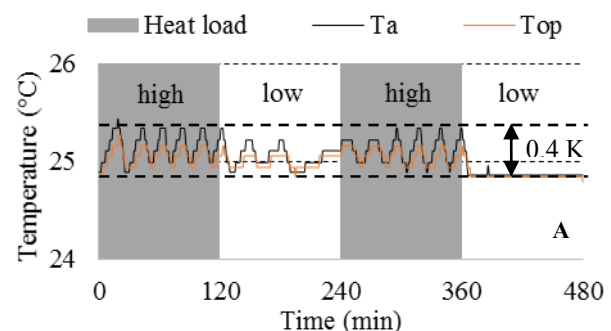
**Table 1.** Dynamic characteristics of ceiling panels and FCU in relation to changes in room air ( $T_a$ ) and operative ( $T_{op}$ ) temperature as the response to the step change in internal heat loads

System design		Ceiling panels	FCU
Time constant ( $\tau$ ) (min)	$T_a$	200	226
	$T_{op}$	240	270
Net increase in room temperature (K)	$T_a$	2.9	2.2
	$T_{op}$	2.6	2.1

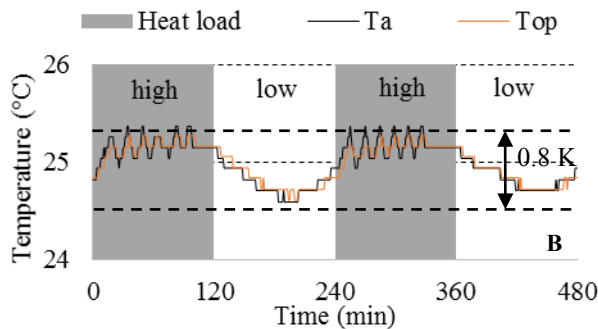
## 4.2. Periodic internal heat gain test

### 4.2.1. Performance of P controller

Results in Figure 4 and Table 1 indicate that using a controller input tuned based upon the response of one of the terminals cannot be a very accurate controller input to control the other terminal. To examine this hypothesis, a proportional (P) controller was tuned and used to the ceiling panels to maintain room air/operative temperature at 25.0 °C. The controller was then applied to the FCU to compare the performance of two cooling terminals in terms of the room temperature control. Figure 4 shows the air and operative temperatures in the room where the P controller was used to adjust the water flow rate to FCU and ceiling panels. For ceiling panels, the maximum overshoot of 0.3 K took place during the “high” heat gain period, and the minimum undershoot measured of 0.1 K occurred during the “low” heat gain period. Using the same control algorithm for the FCU increased the hysteresis gap in room air temperature from 0.4 K to 0.8 K. While the overshoot remained constant for the two terminal units, reduction in undershoot amplitude of about 0.4 K contributed mostly to increase in the hysteresis gap for the FCU. We expected that using the same controller inputs for radiant and convective systems would lead to different hysteresis gap, due to various parameters, including different transfer functions of the room. However, the amplitude was smaller than expected. This might be due to the fact that the added heat gain generated by the simulated floor gradually manifested itself as the heat load in the room. If the added heat gain had more convective characteristics, i.e. capability to increase the room temperature directly instead of heating the floor, the hysteresis gap would have been wider. To minimize the magnitude of the error between the set-point and actual air temperature, the proportional gain coefficients for a control algorithm need to be tuned for each scenario. In practice, this usually requires on-site tuning, because such detailed behavior is very difficult to predict with simulations.



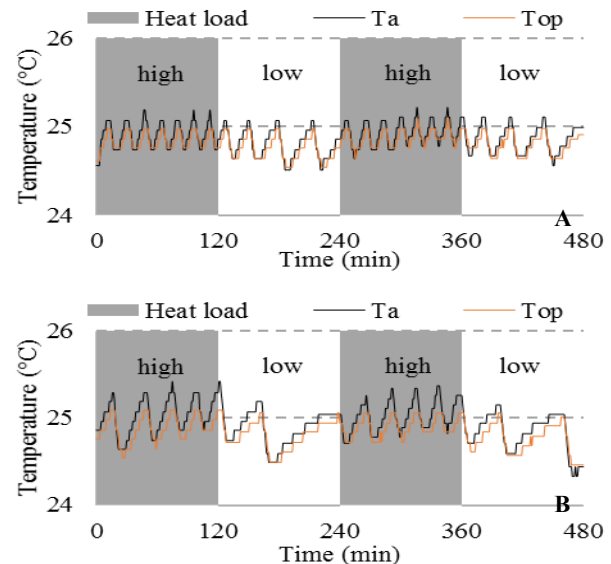




**Figure 4.** Comparison of air ( $T_a$ ) and operative ( $T_{op}$ ) temperature between A) ceiling panels, and B) FCU system tests. The similar proportional gain for P controller was used for both cooling systems to adjust the flow rate to control the room temperature. The proportional gain was tuned only for ceiling panels. The set-point was 25 °C air temperature

#### 4.2.2. Influence of sensor type

To study the influence of the sensor type, the feedback signal from either air or operative temperature sensor was used as the input signal to the controller. The set-point was 25.0 °C, for both sensor types. The on-off controller was used to operate the pump. Both air and operative sensors were located at the same place and height. The results in Figure 5 show air and operative temperatures in the test room handled by the ceiling panels and controlled based on the feedback signals from the air or operative temperature sensors. Overall, no significant difference can be seen in room air/operative temperature pattern between the two cases. It appears that using operative temperature as the feedback signal prolonged the operation time of the pump from 24 minutes to 38 minutes during the “high” heat gain period. In fact, using the operative sensor made the system slower to react to changes in room heat gain. It is also interesting to note that there was no difference between  $T_a$  and  $T_{op}$  larger than 0.1 K when air temperature sensor was used as the reference signal, see Figure 5A. But the difference was more significant, about 0.3 K, when the operative temperature was providing the feedback. This occurs because in most circumstances operative temperature changes more slowly in response to heat gain than the air temperature [13]. This slower change is partly because an operative temperature sensor has a larger time constant than an air temperature sensor, but also partly because, in most circumstances, mean radiant temperature actually changes more slowly than the air temperature. For the non-active room components with higher thermal capacity, the difference between  $T_a$  and  $T_{op}$  would be larger during the transient condition, since it would take longer time for the non-active building components to change their temperature.



**Figure 5.** Air ( $T_a$ ) and operative temperature ( $T_{op}$ ) in the test room with ceiling panels when the control signal to the on-off controller was provided by A) air temperature sensor, and B) operative temperature sensor. The set-point temperature was 25 °C.

Table 2 shows the influence of the feedback signal type on the energy demand of the circulating pump. The cumulative pump working period was moderately longer for 14 min when air temperature sensor was used as the feedback signal. Longer operating time resulted in higher total flow circulated by the pump and higher pump energy use. This can be explained by analyzing the air temperature differentials in Figure 5. The air temperature overshoot during the high heat gain period was higher (for about 0.3 K) when the operative temperature sensor was used. Keeping the room temperature closer to the set-point temperature required longer water circulation time, which explains the difference in the pump energy use with different sensors.

**Table 2.** Pump energy use in relation to the feedback signal type provided by the air temperature sensor or operative temperature sensor. The values were measured for the period of 8 hours under cyclic heat gain condition

Terminal type	Ceiling panels	
	Air	Operative
Sensor type providing the feedback signal		
Pump working period (min)	92	78
Total circulated flow (l)	442	374
Pump energy use (Wh)	49	42
Average room air/operative temperatures (°C)	24.9/24.8	24.9/24.8

Based on the results presented in Figure 5 and Table 2, using the operative temperature sensor did not change the average room temperature, but the pump energy use was moderately reduced.

## 5. Discussion

Results in this study revealed that the dynamic response of the room in relation to heat gain variations was different, depending on the cooling system type. Different heat extraction rates by convective and radiant systems were previously studied in [12,20]. It was concluded that radiant cooling systems removed the heat from the sources earlier than the convective ones. This was because the radiant systems absorbed the radiation directly from the source and reduced the accumulation of heat in the non-active building components. The findings of our study agree well with the previous findings, but we also suggest that these differences shall be considered in control design of convective and radiant cooling systems. Therefore, the application of radiant system control may benefit from custom tuning, especially if the system is operating in the following conditions:

- For buildings with high thermal mass materials or not very well insulated envelope.
- Systems with long delay time, for instance, due to the transport of water.
- Systems with high non-linear characteristics of HVAC equipment such as valves, coils, etc.
- Systems operating for a large part of the year, when they often work at very low loads, which aggravate non-linearities of the system.

In the newly-built offices, installing efficient window glasses along with sunscreen and other protective measures reduce the direct solar heat gain through the transparent building envelope. In addition, temperature sensors in the room are preferably installed in the location not to be exposed to direct sunlight. Therefore, there shall not be a considerable difference between the air and operative temperatures. In this study, we did not find that room temperature control may benefit from using the operative temperature sensor to provide a feedback signal to the control system, as opposed to the suggestions in [10,11]. However, using the operative temperature sensor may be useful in buildings where a large proportion of heat gain comes from radiant sources, [21,22].

The results from the step change test in our test room using ceiling panels and FCU showed that the dynamic behavior of the test room followed the characteristics of the first order systems in both cases. Understanding the response type is helpful when it comes down to the systems' controller design. For systems which do not behave as a first order system, complete analysis of their dynamic behavior under different thermal conditions is required for designing a controller. But for the first order systems, this can be done by merely having the time constant of the system.

## 6. Conclusions

This study investigated the dynamic response of a room cooled by a radiant ceiling cooling panel system and an

FCU system. The results from the dynamic response test were compared and used to show the differences in the room temperature control by these two cooling systems. The following concluding remarks are made:

- Both cooling systems behaved like a first-order system in relation to a step increase in the room heat gain. However, the time constant of the room relative to the heat gain variations was shorter with the ceiling panels, indicating that ceiling panels would react faster to room temperature changes.
- Different cooling loads for ceiling panels and FCU affected the time constant of the room and consequently the proportional gain of the control system of the cooling systems tested. In this study, controlling FCU with the control system tuned for ceiling panels increased the hysteresis gap in the room air temperature from 0.4 K to 0.8 K. This means that traditional control systems for convective cooling systems might be applicable for low mass radiant systems, without sacrificing the indoor thermal environment. However, on-site tuning for different indoor temperatures is recommended to reduce the offset (the persistent error) in the control system.
- In this study, using operative temperature as the feedback variable for a two-position controller resulted in a lower cycling frequency, and a larger mean absolute error between set-point and actual temperature. However, no significant difference in the air temperature was observed by controlling the ceiling panels with either air temperature or operative temperature as the feedback signal. The pump energy use was moderately lower by 14% when ceiling panels were controlled with operative temperature. Nevertheless, controller type (modulating or two-position), thermal characteristics of building material, and heat gain type might influence the results to some extent.

## Acknowledgement

This work was financially supported by Swedish Energy Agency through its national program EFFSYS EXPAND and in kind-contribution of laboratory facilities by UPONOR and LGG Inneklimat AB. We would like to thank Jonathan Woolley (Center for the Built Environment, University of California, Berkeley) for his valuable contribution in reviewing this article. We are particularly thankful to Håkan Larsson for lab assistance.

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