Topic A4. Advanced or innovative HVAC&R systems and system components

A Classification Scheme for Radiant Systems based on Thermal Time Constant

Baisong Ning[1, 2, *], Stefano Schiavon[2], Fred S. Bauman[2]

[1]College of Civil Engineering, Hunan University, Changsha, China
[2]Center for the Built Environment, University of California, Berkeley, CA, USA

[*]*Corresponding email: baisong03@berkeley.edu*

Keywords: Radiant system, Thermal time constant, Classification, Computational fluid dynamics, Heat flux

SUMMARY

Radiant system design standards and guidebooks classify radiant systems as a function of their structure and geometry. We assume that design approaches, testing methods, and control strategies of radiant systems can be more clearly described and classified based on thermal parameters. We calculated using computational fluid dynamics method the thermal time constants for 66 radiant system cases. We found pipe spacing, thickness and material properties of structural layer have a significant impact on time constant, while operative temperature, water temperature and water flow regime do not. We found time constants of radiant ceiling panels vary between 30-91 s; for embedded surface systems between 0.25-3.5 h; for thermally activated building systems with pipes embedded in the massive concrete slabs, between 2.4-7.7 h; and for capillary pipes embedded in a layer at the inner surface, between 0.1-0.6 h. A preliminary radiant system classification scheme based on thermal time constant is proposed.

INTRODUCTION

Current hydronic radiant heating and cooling system design standards (e.g., ISO 11855 2012) and guidebooks (Babiak et al. 2007) categorize them as a function of their structure and geometry. The main categories are: radiant ceiling panels (RCP), embedded surface systems (ESS), and thermally activated building systems (TABS). Based on the position of the embedded pipes, ESS is sub-classified as Type A, Type B, Type C, Type D, and Type G. Type A is the system with pipes embedded in screed or concrete; Type B contains pipes embedded outside of the screed; Type C contains pipes embedded in screed below the splitting foil layer; Type D contains capillary mats in a thin (e.g., gypsum) layer with insulation separating it from the building structure; Type G contains pipes embedded in the sub-floor of a wooden construction. TABS are sub-classified as Type E and Type F. Type E is the system with pipes embedded in massive concrete slabs, and Type F is the system with capillary pipes embedded in a thin layer that can be thermally connected to a massive slab.

Thermal parameters for radiant systems include steady-state and dynamic ones. The most commonly used steady-state thermal parameters are cooling and heating capacity and thermal resistance. It is expected that steady-state parameters will be most suitable for the evaluation of quick acting radiant systems, such as radiant ceiling panels. However, these parameters alone may be insufficient to characterize the operational performance of radiant systems involving larger amounts of thermal mass, including both ESS and TABS. For instance, Feng et al. (2013) found that the cooling loads for radiant systems and air systems have significant differences, and the cooling load calculation for RCP and light-weight ESS are different compared to TABS. We assume that design approaches, testing methods, and control strategies of radiant systems can be more clearly specified based on thermal parameters, not just structure and geometry.

To describe the dynamic characteristics of radiant systems, we decided to use the thermal time constant, τ, a parameter initially used in the lumped system analysis method. In this study, the thermal time constant is defined as the time required by the radiant system to reach 63.2% of the difference between the original and new values (e.g., surface temperature, heat flux) after a step change. In current radiant system design standards (e.g., ISO 11855 2012), there is only an approximate qualitative description about the response time of radiant systems (fast or slow), as no quantitative description is available. Several studies described the dynamic behavior of radiant systems using both theoretical and experimental methods. Thomas et al. (2011) showed with laboratory experiments that a light-weight radiant floor achieved 80% of its maximum emitting power after more than 30 min of operation. By using numerical methods, Weitzman (2004) modeled building integrated heating and cooling systems and found that the time constants for light-weight floor heating systems range from 30 min to 2 h, while 3-6 h for heavy floor heating systems. Larsen et al. (2010) developed an analytical solution to calculate the dynamic surface temperature and heat storage for ESS. Zhao et al. (2014) used an analytical method to calculate the time constant for light-weight and heavy-weight radiant floors. However, only part of all radiant system types have been studied and these theoretical studies can be used for the description of ESS with insulation separating the building structure, but are not appropriate for TABS, which may use both floor and ceiling for heating and cooling purposes. Moreover, the effect on calculated thermal time constant of thermal capacity and conductivity of different material layers, fluid temperature, fluid flow rate, pipe spacing, room operative temperature, surface heat transfer coefficients, etc. are not well understood.

Therefore, the objectives of this study are to: (1) investigate the effect on thermal time constant of radiant systems for the following parameters: thermal capacity and conductivity of different material assemblies, fluid temperature, fluid flow rate, pipe spacing, room operative temperature, and surface heat transfer coefficient; (2) calculate the thermal time constant for the typical radiant system types.

METHODS

In this study, computational fluid dynamics (CFD) method is used for the calculation of thermal time constant. CFD method can be used for the calculation of radiant system types that are not easy to be calculated by analytical methods. After a comparison of the CFD software Fluent 6.3 (finite volume method) with Heat 2 8.0 (explicit finite difference method) and Therm 6.0 (finite element method), we found that Fluent 6.3 was preferred because Therm 6.0 cannot conduct dynamic calculations, while the pipe resistance cannot be properly taken into account in Heat 2 8.0. To simplify the heat transfer model, the following assumptions were applied, as shown in Figure 1: (1) the materials of each layer are homogeneous and their thermal properties are constant; (2) the pipes have a symmetric layout; a calculation unit with two symmetric pipes is used for each calculation case; (3) the water temperature along the pipe is uniform, average water temperature is used for each pipe of the calculation unit, so that the 3D model can be simplified as 2D model; and (4) the flow regime of the water in the pipe is turbulent.

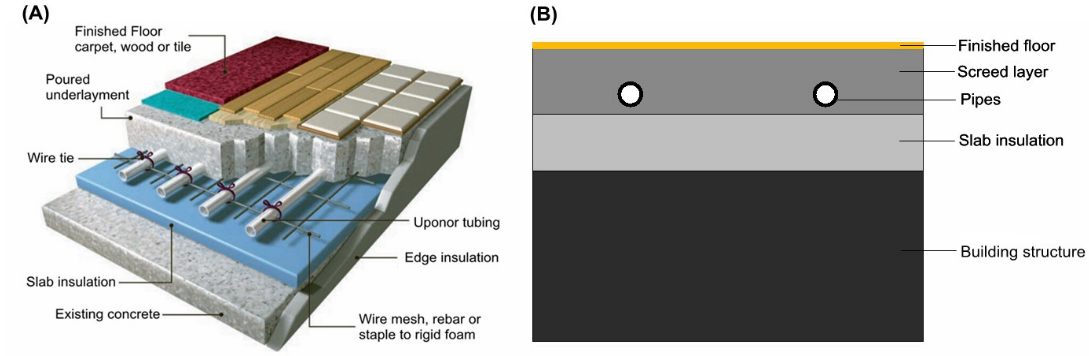

Figure 1. Example of a radiant system（A）3D graphical representation of type A (Uponor, 2013) and (B) schematic representation of the 2D calculation model.

The boundary conditions are as follows: for radiant heating system, room set-point operative temperature is 20 °C, average water temperature is 35 °C; for radiant cooling system, room set-point operative temperature is 26 °C, average water temperature is 17 °C. Total heat exchange coefficient h (combined convection and radiation) between radiant surface and space are as follows (Babiak et al. 2007): for floor heating, h_{FH} = 11 W/(m^2 K); for floor cooling, h_{FC} = 7 W/(m^2 K); for ceiling heating, h_{CH} = 6 W/(m^2 K); for ceiling cooling, h_{CC} = 11 W/(m^2 K). The left and right boundary conditions are adiabatic due to the symmetric layout. The convective heat transfer coefficient for the water and pipe surface is calculated based on the method given by ASHRAE Handbook of Fundamentals (Chapter 26, ASHRAE 2012).

Input simulation parameters include geometric parameters and thermal parameters (density, thermal conductivity, and specific heat). These data are obtained from several sources as radiant system design guidebooks (Babiak et al. 2007) and engineering manuals (Uponor 2013; Price 2011; BEKA 2014; Viega 2013). The input parameters for the case named type A in radiant system design standard (ISO 11855 2012) shown in Figure 1 are listed in Table 1. The input parameters for all the other radiant systems modeled cannot be reported here due to the space constraints. They are available at http://bit.ly/CobeeRadiant.

Table 1. Input parameters for type A: radiant floor

Type A Radiant Floor	Layer name	Material name	Thermal properties source	Density (kg/m^3)	Thermal conductivity (W/(m K))	Specific heat J/(kg K))	Thickness (mm)
1	Floor covering	Tiles Ceramic	ISO 10456:2007(E)	2300	1.3	840	5
2	Weight bearing, and thermal diffusion layer	Cement mortar	C 26.5 ASHRAE (2012)	1600	0.97	840	50
3	Pipes	Cross linked polyethylene (PEX)	UPONOR (2013)	936	0.38	1470	OD=19.05 ID=14.58 Su=25 T=150
4	Thermal insulation	Glass fiber board	C 26.5 ASHRAE (2012)	160	0.032	1670	60
5	Building structure	Aggregate concrete	C 26.5 ASHRAE (2012)	2400	1.4	800	150

Note: "OD" means outside diameter; "ID" means inside diameter; "Su" means the distance from the center of the pipe to the bottom of the embedded layer; "T" means pipe spacing.

RESULTS

Calculation procedures of thermal time constant

There are two main thermal time constants for radiant system, the one based on surface temperature and the one based on heat flux; both can be used for our purpose. In this paragraph, we present an example of how the time constants are calculated. The detailed input parameters are listed in Table 1. Figure 2 shows the change of average surface temperatures (part A) and area weighted surface heat flux (part B), and for radiant floor heating.

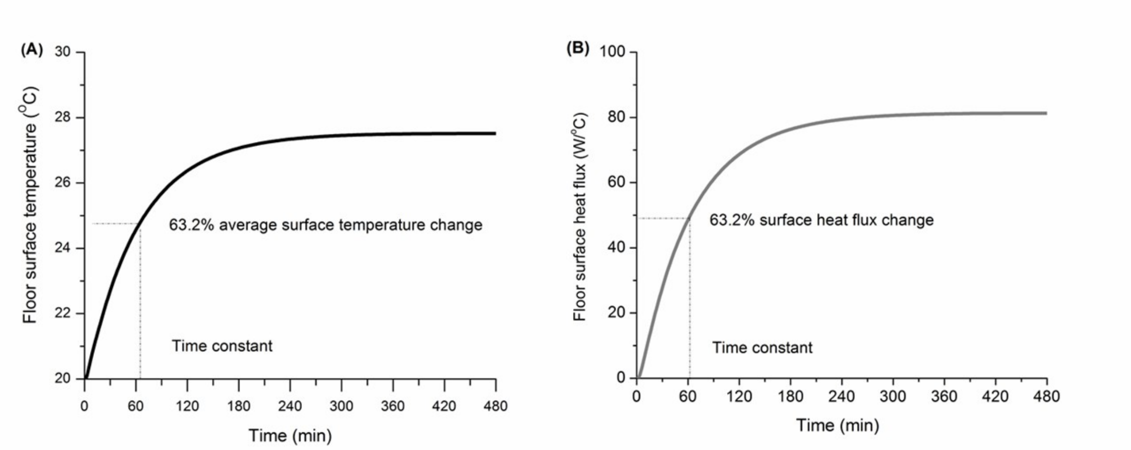

Figure 2. Thermal time constants for (A) average surface temperature and (B) surface heat flux.

We can see from Figure 2 that at first the average surface temperature increase very fast, then the rate of increase becomes slower; and after about 300 minutes, it goes to steady-state. The time (about 66 minutes) when the average surface temperature reaches 63.2% of the steady-state value is the thermal time constant. After about 4 time constants, the average surface temperature goes to steady-state. The time constant for the surface heat flux is the same with average surface temperature. We decided to use the average surface temperature time constant (hereinafter as “time constant”) as the calculated time constant in this study.

Impact factor analysis for typical radiant system

Geometric parameters, thermal parameters and boundary conditions can impact the time constant. Many possible combinations could happen and to simplify the analysis and reduce the number of simulations, we first did a simple sensitivity analysis to study the most important parameters. We use the above described embedded surface system Type A as the basic calculation case. Room operative temperature, average water temperature, water flow Reynolds number, pipe spacing, pipe diameter, thickness of screed layer, pipe embedded layer material, floor covering material are considered for the potential impact factors (as listed in table 2). In the sensitivity analysis cases, only the value of the above impact factors are changed compared with the basic calculation case. We use the standard deviation of the calculated time constant for the analysis of the sensitivity of impact factors, see equation (1). Table 2 has listed the time constant and the standard deviation for each studied factor. Besides, the cooling and heating capacity when radiant systems reach steady states are also listed in the table because they are important thermal parameters for the design and operation of radiant systems.

$$s = \sqrt{\frac{1}{N-1}\sum_{i=1}^{N}\left(t_i - \bar{t}\right)^2} \quad (1)$$

Where s is the standard deviation of time constant, N is the number of calculated cases for each impact factors, τ_i is the calculated time constant for different cases, $\bar{\tau}$ is the average of τ_i for different cases of one impact factor.

Table 2 Impact factor analysis for type A radiant floor

Type A radiant floor	Cooling				Heating			
Impact factors	Variable value	Cooling capacity (W/m^2)	Time constant (min)	Standard deviation	Variable value	Heating capacity (W/m^2)	Time constant (min)	Standard deviation
Operative temperature	24 °C	29.4	77.6	0.7	18 °C	92.5	65.4	0.6
	26 °C	37.9	76.9		20 °C	81.6	65.9	
	28 °C	46.3	76.3		22 °C	70.7	66.5	
Average water temperature	16 °C	42.1	76.6	0.4	30 °C	54.4	67.4	1.3
	17 °C	37.9	76.9		35 °C	81.6	65.9	
	18 °C	33.6	77.3		40 °C	108.8	64.8	
Water flow Reynolds number	4500	37.7	77.9	0.7	4500	81.2	67.7	1.1
	7000	37.9	76.9		7000	81.6	65.9	
	9500	37.9	76.7		9500	81.8	65.6	
	12000	38.0	76.4		12000	81.9	65.3	
Pipe spacing	150 mm	37.9	76.9	19.6	150 mm	81.6	65.9	13.8
	225 mm	30.8	100.3		225 mm	63.6	88.2	
	300 mm	25.1	115.8		300 mm	50.5	91.2	
Pipe diameter	OD=15.88 mm ID=12.07 mm	36.9	79.8	2.3	OD=15.88 mm ID=12.07 mm	78.9	68.5	2.1
	OD=19.05 mm ID=14.58 mm	37.9	76.9		OD=19.05 mm ID=14.58 mm	81.6	65.9	
	OD=22.23 mm ID=17.04 mm	38.5	75.2		OD=22.23 mm ID=17.04 mm	83.5	64.3	
Thickness of screed layer	40 mm	39.4	58.3	22.4	40 mm	86.1	49.3	17.0
	50 mm	37.9	56.9		50 mm	81.6	65.9	
	60 mm	36.6	96.3		60 mm	77.9	83.3	
Pipe embedded layer material	Cement mortar	37.9	76.9	6.4	Cement mortar	81.6	65.9	5.9
	Low-mass aggregate concrete	37.0	86.8		Low-mass aggregate concrete	79.4	74.9	
	Sand and stone aggregate concrete	41.5	88.9		Sand and stone aggregate concrete	91.9	77.1	
Floor covering material	Tiles Ceramic/ porcelain	37.9	76.9	6.3	Tiles Ceramic	81.6	65.9	5.0
	Wood	33.6	75.3		Wood	70.1	66.9	
	Tiles ceramic and carpet	28.7	87.0		Tiles ceramic and carpet	65.3	75.0	

From Table 2 we can conclude that pipe spacing, thickness and material of pipe embedded layer material, and floor covering material have a significant effect, while room operative temperature, average water temperature and water flow Reynolds number have a small effect on time constant.

Recommended time constant range for typical radiant systems

To simplify the calculation, for each radiant system type, we make a design of specified parameters of radiant system for the minimum and maximum time constant calculation cases. For the minimum calculation cases, the significant impact factors values are set to the value with a shorter time constant. For the maximum calculation case, the significant impact factors values are set to the value with a larger time constant. Besides, engineering practice is taken into consideration when choosing the value of the significant factors. The input parameters for each calculation case, the resulting time constants, and cooling and heating capacities can be found at http://bit.ly/CobeeRadiant.

Except for radiant ceiling panels, most of the modeled radiant systems are radiant floors. For the radiant system type like Type D that can be installed as both radiant ceiling and radiant floor system, the time constants are calculated separately. For TABS, like Type E, as heat is transferred to both the floor and ceiling, two time constants are calculated, one for the floor and one for the ceiling. Based on the calculation, the thermal time constant ranges (minimum and maximum values) for radiant heating and cooling system types modeled in this paper and described in the current design standards are shown in Figure 3.

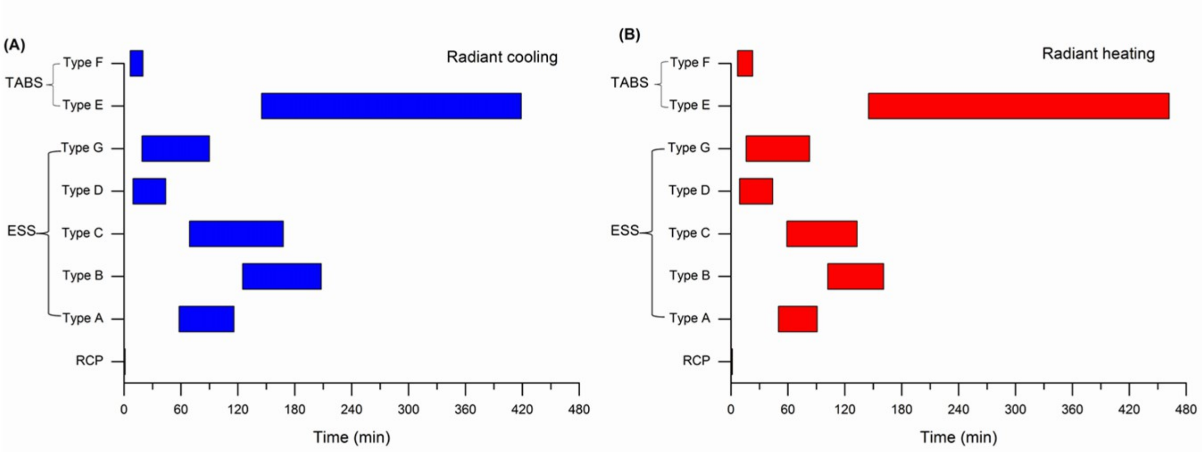

Figure 3. Time constant for different radiant system types (A) in cooling and (B) heating.

From Figure 3, we can see the time constants for the radiant system types described in the current radiant system design standards and guidebooks vary from 30 s to almost 8 h.

For radiant ceiling panels systems the time constant are about 31-72 s for ceiling cooling, and 38-91 s for heating. This is because the heat transfer coefficients between floor surface to the space is larger in the heating mode in comparison with cooling mode.

For embedded surface systems floor systems like Type A, Type B, Type C and Type G, the time constant range is 16-161 min in heating mode, and 19-208 min in cooling mode. For the capillary tube mats, type D, the time constants vary between 9.5 to 44 min for cooling mode, 9-32 min for heating.

For thermally activated building systems like type E, the time constants are 146-419 min for cooling mode, 145-462 min for heating. For Type F, the time constants are 6.5-20 min for cooling mode, and 7.5-22 min for heating.

It should be noted that this study gives an approximate result of time constant based on the calculated cases, for radiant walls and other types of radiant systems, it's not covered here. The feasibility of developing a detailed classification of radiant systems based on time constant and the design approaches, testing methods, and operational strategies will be investigated in a future study.

CONCLUSIONS

Pipe spacing, thickness and material of pipe embedded layer material, and floor covering material, have significant impact on time constant of radiant systems, while room operative temperature, average water temperature and water flow regime have insignificant impact.

The time constants for 66 radiant system cases have been calculated. We can see radiant ceiling panels are quick response radiant systems, as the time constant varies between 30-91 s. TABS like Type E is are slow response radiant system types, as the time constants are between 2.4 to 7.7 h. Embedded surface systems and thermally activated building systems like Type F can be regarded as medium response radiant system types; the time constant is between 0.1 to 3.5 h. By using the time constant, thermal behaviors of radiant systems can be better described.

ACKNOWLEDGEMENT

This study was supported by the China Scholarship Council (NO. 201406130019).

REFERENCES

ASHRAE. 2012. *ASHRAE Handbook: HVAC Systems and Equipment*, Panel heating and cooling. Atlanta: American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc.

ASHRAE. 2012. *ASHRAE Handbook: Fundamentals*, Chapter 26, Heat, Air, and Moisture Control in Buildings Assemblies-Material Properties. Atlanta: American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc.

Babiak J., Olesen B., and Petras D. 2007. *Low temperature heating and high temperature cooling*. Brussels: REHVA Guidebook, Federation of European Heating and Air-Conditioning Associations.

BEKA. 2014. *Capillary Tube Mats*. http://www.beka-klima.de/en

Feng J., Schiavon S., and Bauman F. 2013. Cooling load differences between radiant and air systems. *Energy and Buildings*, 65, 358-367.

ISO. 2012. *ISO 11855:2012*, Building Environment Design-Design, Dimensioning, Installation and Control of Embedded Radiant Heating and Cooling Systems. Geneva: International Organization for Standardization.

ISO. 2007. *ISO 10456:2007*, Building materials and products-Hygrothermal properties-Tabulated design values and procedures for determining declared and design thermal values, International Organization for Standardization, 2007.

Weitzmann P. 2004: Modelling building integrated heating and cooling systems. *Ph.D. Thesis*. Technical University of Denmark, Denmark

Price. 2011. *Price Engineer's HVAC Handbook*. Winnipeg: Price Industries.

Larsen S.F., Filippín C., and Lesino G. 2010. Transient simulation of a storage floor with a heating/cooling parallel pipe system. *Building Simulation*, 3, 105–115.

Thomas, S., P.Y. Franck., and P. André. 2011. Model validation of a dynamic embedded water base surface heat emitting system for buildings. *Building Simulation*, 4, 41–48.

Uponor. 2013. *Radiant Cooling Design Manual, Embedded Systems for Commercial Applications*. Vantaa. Uponor Corporation.

Viega. 2013. *Viega ProRadiant*. http://www.viega.us/xchg/en-us/hs.xsl/5876.htm

Zhao K., Liu X. H., and Jiang, Y. 2014. Dynamic performance of water-based radiant floors during start-up and high-intensity solar radiation. *Solar Energy*, 101, 232-244.