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Highlights

- Applicability of six floor, wall, and ceiling radiant heating systems was studied
- Output, controllability, heat storage, building retrofit and costs were considered
- Wall with pipes in plaster performed best, had fast response but limited storage
- Classical floor heating performed consistently in all criteria
- Ceiling TABS was feasible only when long-term heat storage was needed

A comparative study on the applicability of six radiant floor, wall, and ceiling heating systems based on thermal performance analysis

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Abstract

Holistic comparisons of radiant heating systems that would help make an informed decision on the selection of the most convenient system for the specific application are lacking. The applicability of six representative radiant floor, wall, and ceiling heating systems was therefore compared in terms of thermal output and surface area required, controllability, short-term and long-term heat storage, suitability for building retrofit, and investments. Temperature and heat flux distribution in the structure, time constant τ_{63}, response time τ_{90}, and the number of operating cycles were computed by a custom-made and verified software tool using the finite volume method. Thermal energy stored was used to determine the ability of energy storage, whereas investment costs indicated affordability. Wall heating with pipes attached to a thermally insulating core had the highest thermal output, was easy to control, suitable for building retrofit, and most affordable while providing limited thermal storage. The performance of the wall system was retained when locating the pipes in plasterboard separated from the core by an air gap. Floor heating performed consistently in all the aspects evaluated. It was demonstrated that inserting a metal fin between pipes and the concrete spread layer improved thermal output, controllability, and storage capacity of the floor system with minor effect on investments. Ceiling with pipes insulated from the core performed well when thermal storage was not required. Ceiling with pipes embedded in the core was only feasible when long-term heat storage was needed.

Keywords: Radiant heating; thermal response; heat transfer; building retrofit; performance; thermally active building systems (TABS)

1. Introduction

Water-based radiant systems present a potentially viable solution for space heating because they are suitable for integration with renewable energy sources [1,2,3] and have the ability to create a comfortable thermal environment [4,5,6,7]. The applicability of the individual system types depends on their location (floor, wall, or ceiling), the configuration of material layers, and the level of thermal mass. These characteristics are crucial for the selection of the most suitable system for the specific situation such as the construction of new building vs. retrofitting of an existing building, thermal storage vs. fast thermal response, and traditional vs. low-temperature renewable heat source.

Several studies have compared the radiant systems assuming various locations and configurations of the material layers. The first category of studies involves calculations performed for a representative fragment of the heating or cooling element. Oxizidis and Papadopoulos [8] compared the energy performance and thermal comfort created by a floor cooling system, ceiling with pipes embedded in plaster, wall with pipes located in the plaster, and a generic thermally active building system (TABS). Although floor cooling consumed the least energy, it could not provide enough cooling output to attain thermal comfort. Ning et al. [9] calculated the response times of typical radiant systems as defined in ISO 11855 [10]. The systems were classified into three categories according to their thermal response: fast, medium, and slow. The difference in response times of ceiling/floor heating/cooling systems was explained by the different heat transfer coefficients between the radiant surface and indoor air. Krajčík and Šikula [11] compared four wall cooling systems. The system’s suitability depended on the requirements such as exploiting thermal storage, avoiding interventions on the interior surfaces, or attaining a rapid thermal response. The cooling output was sensitive to insulation thickness for the systems with pipes located in the thermal core and to pipe spacing for the systems having pipes underneath the surface.

The second category of studies involves experiments and computer simulations on a whole-room level. Mustakallio et al. [12] experimentally compared a chilled beam, chilled beam combined with a radiant panel, chilled ceiling combined with mixing ventilation, and four localized cooling panels combined with mixing ventilation. The differences in the thermal environment created by the systems were small. Le Dréau and Heiselberg [13] performed computer simulations of an active chilled beam, radiant floor, radiant ceiling, and radiant wall in an occupied room. Using radiant floor resulted in the lowest cooling demand but created least uniform thermal conditions. The most homogeneous thermal environment was attained by the radiant ceiling. Bojić et al. [14] numerically investigated the performance of ceiling, wall, floor, and floor-ceiling heating systems. The floor-ceiling system performed best in terms of energy and exergy saving, exergy destruction, CO_2 emissions, and operation costs whereas a single ceiling system was the least preferable option. Karabay et al. [15] recommended using wall over floor heating because it provided higher thermal performance and more favourable thermal conditions with lower water temperature, thus reducing energy consumption. On the other hand, the computer simulations by Myhren and Holmberg [16] showed that the vertical temperature gradient in the centre of the room was smaller for floor heating than for wall heating.

Thermal performance of radiant systems represented by, e.g., heating capacity, thermal resistance, and thermal response, is more relevant for design, testing and control of radiant systems compared to geometry and structure [9]. The thermal response is a decisive factor to determine the control strategy that is appropriate for the specific application. It is especially important for the design and operation of radiant systems with larger amounts of thermal mass. The related literature describes several approaches to assess the dynamic thermal performance of radiant systems. The literature survey has shown that response time τ_{63}, also referred to as the time constant, is frequently used as an indicator of thermal response. Time constant represents the time to reach 63% of the final value of the surface temperature, thermal output, or room temperature [17,18,19]. Alternatively, other percentages of the final value, e.g. 80% [20], 95% [9], may be used for the calculation. Depending on the radiant system used, the response time ranges from several minutes for, e.g., suspended ceiling panels up to several tens of hours for TABS [9]. A system with such a long response time requires using precise control strategy to provide comfortable conditions [21,22,23].

As reported by Ning et al. [9], a single number like τ_{63} or τ_{95} may not be fully representative of the thermal response of radiant systems. In such a case, calculating several response times, e.g., τ_{25}, τ_{50}, τ_{63}, or τ_{80} may be needed [9,24]. Other indicators of thermal response found in the related literature include the peak value of surface temperature [25], thermal admittance and transmittance [26,27], visual comparison of the step-up and decay curves of surface temperature [20,28,29,30], the required number of operation cycles to maintain thermal output between defined boundaries [11], and the heat storage efficiency that takes into account the evolution of surface temperature until reaching steady-state [31].

The literature review has shown that the existing studies usually compare the systems only from thermal comfort or energy efficiency point of view. Most of the research focuses on systems with massive thermal layers, whereas less attention is given to lightweight systems and their comparison with the massive systems. Moreover, the potential use of radiant systems for building retrofit has rarely been considered. It follows that holistic comparisons of radiant heating systems that would help make an informed decision on the selection of the most convenient system are lacking. The present study aims to facilitate the selection of the most suitable radiant heating system for both newly constructed and renovated buildings. The emphasize is on the thermal performance of the systems while also considering their price and suitability for building retrofit.

To accomplish this, numerical investigations of six representative radiant floor, wall, and ceiling heating systems (Figure 1) have been performed in terms of thermal output, area of the heating surface required, controllability, short-term and long-term energy storage, suitability for building retrofit, and investment costs. Thermal fields and heat flux distribution in the structure were computed to evaluate the thermal output. Time constant (τ_{63}), response time (τ_{90}), and the number of operating cycles were used to assess the thermal response and controllability. Thermal energy stored in the structure and investment costs were also determined to help evaluate the systems in a broader context.

2. Radiant heating systems investigated

Figure 1 shows the composition and configuration of the heating systems studied. The systems were designed to cover the design heat load which consisted of the design heat loss (744 W) and the heat-up capacity (10% of the design heat loss). The design and construction of the systems were supposed to closely reflect reality. Therefore, the differences between the systems in this respect were considered. The pipe spacing and diameter varied depending on the heating system used, which resulted in differences in the heating area, thermal output, and surface temperature. For example, the pipe spacing of the wall heating systems is denser than that of the floor heating systems, therefore the area is smaller and the surface temperature is higher for wall heating. All the heating systems were located between two conditioned rooms. The systems are described as follows:

(A) Floor – classical floor heating consisting of a bearing structure, system board made of a thermally insulating material, pipes embedded in the system board, a concrete spread layer, and wooden parquets. The outer and inner diameter of the pipes was 16 and 14 mm, respectively, and the pipe spacing was 150 mm.

(B) Floor with metal fins – like A, but a metal fin with high conductivity was inserted between the pipe and the concrete spread layer to enhance heat transfer.

(C) Wall (TABS) – pipes embedded in plaster underneath the surface of an internal wall. The outer and inner diameter of the pipes was 12 and 10 mm, respectively, and the pipe spacing was 120 mm.

(D) Wall (air gap) – pipes with an outer diameter of 10 mm and an inner diameter of 9 mm, spaced by 80 mm and embedded in a 15 mm plasterboard. The plasterboard panels can have various dimensions and be connected in series. The thermally active plasterboard is decoupled from the thermal core by an air gap, which may be filled with thermal insulation.

(E) Ceiling (TABS) – pipes embedded in the thermal core that is insulated to prevent heat loss to adjacent rooms. The outer and inner diameter of the pipes was 20 and 18 mm, respectively, and the pipe spacing was 300 mm.

(F) Ceiling (TI) – the heat was emitted through plasterboard connected with pipes located in an air gap. The thermally active layer was insulated from the main thermal mass to prevent heat storage and ensure a fast thermal response.

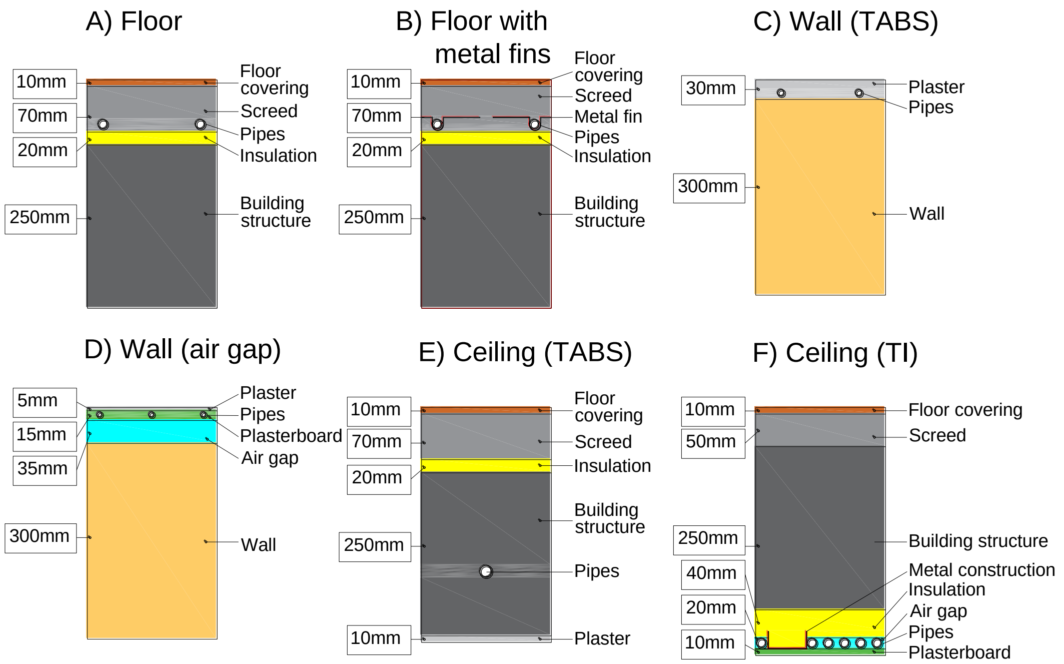

Figure 1. Radiant heating systems investigated: A – Floor, B – Floor with metal fins, C – Wall (TABS), D – Wall (air gap), E – Ceiling (TABS), F – Ceiling (TI).

3. Methodology

The heating system was located in the living room of a residential building. The dimensions of the room were 6 m x 4 m which can be considered usual in the region of Central Europe. The room design heat load was determined to be 818 W following EN 12831-1 [32]. This heat loss corresponds to an external temperature of -12°C and a room operative temperature of 20°C. These temperatures are representative of the design conditions in the humid continental climate typical of, e.g., Central and Eastern Europe or places like Maine and Michigan, USA. The mean temperature of the heating water was 35°C corresponding to the thermal gradient of 38/32°C.

In the present study, computer simulations were used as the research method because they allow to precisely formulate of the boundary conditions and retain the conditions for all the systems tested while being less costly than experimental measurements. A custom-made and verified software tool was employed to compute the temperature and heat flux distribution in the structure, time constant τ_{63}, response time τ_{90}, number of operating cycles, and thermal energy stored using the finite volume method.

3.1 Physical model and calculation principle of heat transfer

The heat transfer rate was calculated for a characteristic fragment of the structure. Figure 2 shows the configuration and composition of the material layers for each of the fragments as defined in the calculation model. The thermophysical properties of the individual materials are defined in Table 1. In the analysis, the thermophysical properties of materials were assumed to be isotropic, temperature independent, and constant. The heat was emitted to the environment through the surfaces facing the rooms.

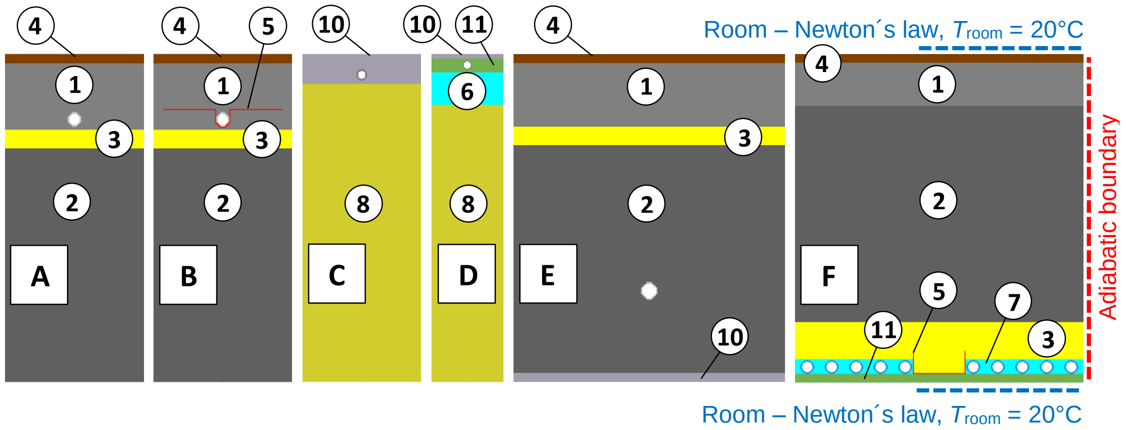

Figure 2. Radiant heating systems as defined in the calculation model: A – Floor, B – Floor with metal fins, C – Wall (TABS), D – Wall (air gap), E – Ceiling (TABS), F – Ceiling (TI). T_{room} – room temperature.

Table 1. Thermophysical properties of materials.

Material	Colour	Volumetric weight ρ (kg/m^3)	Thermal conductivity λ (W/(m·K))	Specific heat capacity c (J/(kg·K))
1 – Concrete		2100.0	1.23	1020
2 – Reinforced concrete		2400.0	1.58	1020
3 – Thermal insulation		25.0	0.04	1270
4 – Hardwood		600.0	0.22	2010
5 – Metal fin (steel) 1 mm		7850.0	58.00	440
6 – Air gap – system D		1.2	0.20	1010
7 – Air gap – system F		1.2	0.01	1010
8 – Aerated concrete		400.0	0.15	1000
9 – PE-Xa pipe		1270.0	0.45	1980
10 – Plaster		2000.0	0.99	790
11 – Plasterboard		750.0	0.22	1060

The heat flux and temperature distribution for stationary and dynamic analyses were obtained by CalA software [33,34]. The software was developed by one of the authors of this study to calculate two-dimensional heat and moisture transfer in building structures and was verified following the procedure as defined in ISO 11855 [10], Part 2, Annex D. For the verification of the software, the readers are referred to the supplementary material in Ref. [31]. The calculation principle of the software was previously described in Refs. [31, 35]. The governing equation describes the problem as two-dimensional transient heat conduction as follows:

$$\frac{\partial T}{\partial \tau} = \alpha \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) \tag{1}$$

where T, τ and α represent the temperature (K), time (s), and thermal diffusivity (m^2/s). Thermal diffusivity is the ratio of thermal conductivity of a substance to the product of its density and its specific heat capacity.

The models in Figure 2 consider both convection and radiation heat transfer in a single Robin-Newton boundary condition. The convective and radiative phenomena have been included in the total heat transfer coefficients on the surfaces. The total heat transfer coefficient on the inner surface (h_i) was 6, 7, and 8 W/(m^2.K) for the ceiling, floor, and wall, respectively. These coefficients can be considered realistic and are consistent with the values as defined in the related standards and literature [10,36,37,38,39]. The total heat transfer coefficient on the outer surface (h_e) was always 8 W/(m^2.K) representing an adjacent room. Although in reality, h_e may vary depending on the surface orientation, the effect of the difference in h_e between vertically (wall) and horizontally (floor, ceiling) oriented surfaces on the heat flux and temperature distribution is small and was therefore neglected. The heat transfer coefficient between pipe and water (h_{pipe}) ranged between 1170 and 1800 W/(m^2.K) depending on the system design. A sensitivity study has confirmed that in this range varying h_{pipe} has a negligible effect on the rate of heat transfer.

The boundary conditions on the surface of the computational domain were given by Eq. 2. The boundaries of the wall fragment were assumed adiabatic (Eq. 3):

$$-\lambda \left(\frac{\partial T}{\partial n} \right)_w = h\left(T_w - T_f\right) \tag{2}$$

$$-\lambda \left(\frac{\partial T}{\partial n} \right)_w = 0 \tag{3}$$

Here, the indexes n, w, and f denote the direction perpendicular to the surface, surface of an object, and the surrounding fluid, respectively; h is the combined heat transfer coefficient (W/(m^2.K)) which includes both the convective and radiative heat transfer from a radiant surface to the environment.

The heat transfer analysis was carried out using the Finite Volume Method. The implicit Euler scheme was used for the temporal discretization. The Gauss-Seidel iterative method with successive over-relaxation approach was employed to solve the resulting system of linear equations. The calculation was terminated when the convergence criterion given in Eq. 4 was satisfied:

$$\frac{\sum_{i=1}^{k} q_{,i}}{\sum_{i=1}^{k} |q_{,i}|} \leq 10^{-6} \qquad (4)$$

where q_i is the summation of all the heat fluxes entering and leaving the control volume.

3.2 Grid and timestep independence test

A uniformly structured mesh with rectangular elements was utilized to discretize the computational domain. The grid was progressively refined from 2×2 to 1×1 to 0.5×0.5 mm×mm to fulfil the cell size criteria as specified in ISO 10211 [40]. A similar test was done for the timestep, considering the timesteps of 15, 5, and 2.5 min. Both the grid and timestep independence were tested on system E – Ceiling (TABS). The simulation started by switching on the heating system, i.e. applying a step change in water temperature from 20 to 35°C and finished by turning the system off after 24 h. In grid independence tests, the mean surface temperature of the ceiling was monitored over 24 h with the time step of 15 min. In timestep independence tests, the mean surface temperature of the ceiling was computed over 24 h with the grid size of 2×2 mm×mm. The variation of the mean surface temperature for different mesh sizes and timesteps is plotted in Figure 3. As seen in the figure, all curves have the same trend.

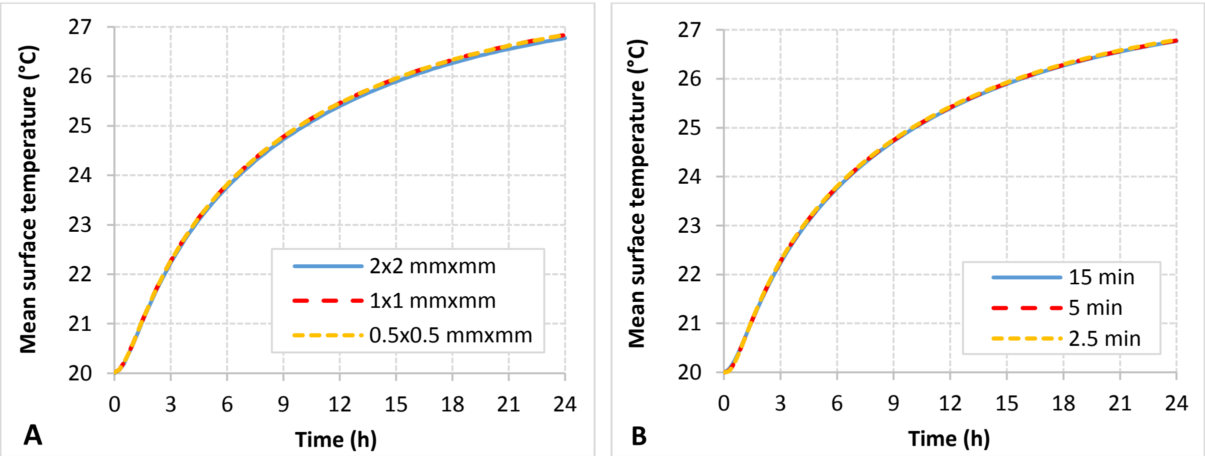

Figure 3. Temporal variation of mean surface temperature for various A) grid sizes, B) timesteps.

To quantify the influence of mesh size on the temperature profile, the absolute difference in the mean surface temperature of the actual grid size and the finest grid size of 0.5×0.5 mm×mm was averaged over 24 h (Table 2). Likewise, the absolute difference in the mean surface temperature of the actual time step and the finest time step of 2.5 min was averaged over 24 h. Based on these tests, a grid size of 1×1 mm×mm and the time step of 5 min were used in the simulations. The only exception was system E – Ceiling (TABS) where a grid size of 2×2 mm×mm was used as a suitable trade-off between the accuracy and computation costs.

Table 2. Grid and timestep independence test using the average surface temperature ($T_{surf,avg}$)

Grid size (mm)	2x2	1x1	0.5x0.5	**Timestep (min)**	15	5	2.5
$\Delta T_{surf,avg}$ (°C) over 24 h	$5.6*10^{-2}$	$4.0*10^{-7}$	-	**$\Delta T_{surf,avg}$ (°C) over 24 h**	0.032	0.011	-

$\Delta T_{surf,avg}$ – absolute difference in the mean surface temperature of the actual and finest grid size or timestep averaged over 24 h

3.3 Sensitivity analysis – effect of selected parameters on thermal output

A sensitivity analysis was conducted to observe the influence of various parameters on thermal output. The parameters studied were the heat transfer coefficient between pipe and water (h_{pipe}), heat transfer coefficient at the inner (h_i) and outer (h_e) surface of the structure, temperature of water in the pipes, and the room temperature. For each of the heating systems, simultaneously decreasing h_i and h_e by 1 W/(m^2.K) reduced the thermal output by at least 5% (Figure 4). This reduction was almost fully attributed to changing h_i, while the effect of h_e was small. Lowering the water temperature by 1°C in the interval 33 to 37°C reduced the output linearly by about 6% and lowering the room temperature by 1°C in the interval 18 to 22°C reduced the output by 5-7% regardless of the heating system.

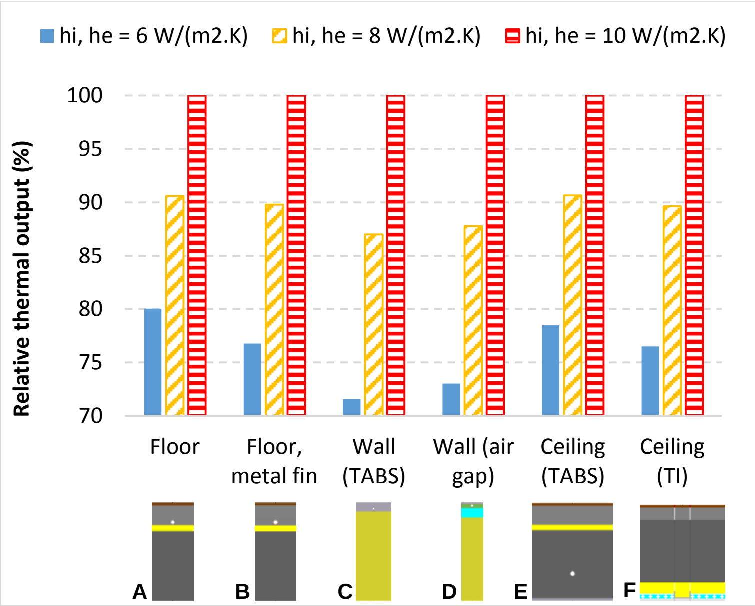

Figure 4. Relative effect of heat transfer coefficients at inner (h_i) and outer (h_e) surfaces of the structure on thermal output.

4. Results

Both the steady-state and dynamic thermal performance of the systems was evaluated. The steady-state performance was characterized by the temperature and heat flux distribution and thermal output. The dynamic thermal performance was analysed using time constant τ_{63} and response time τ_{95} as well as the number of operation cycles within 24 hours. Thermal energy stored over 24 hours and system costs were also elaborated to allow a complex comparison of the heating systems.

4.1 Temperature and heat flux distribution

Stationary simulations of heat transfer were used to visualize the temperature and heat flux distribution. The thermal fields at the design heat load are shown in Figure 5. The surface temperature is least homogeneous in the two wall heating systems (C, D) because the pipes are too close to the surface to create a uniform surface temperature. The surface temperature distribution is most uniform for floor heating (B) due to the positive effect of the conductive metal plate. In all the cases, the heat is directed to the interior and the heat loss is small (Figure 6). In the floor (A, B) and ceiling (E, F) systems this is attained by incorporating a thermal insulation layer in the structure. In the wall systems, the losses are prevented by using an insulating core material (C) and an air gap that separates the core from the pipes (D).

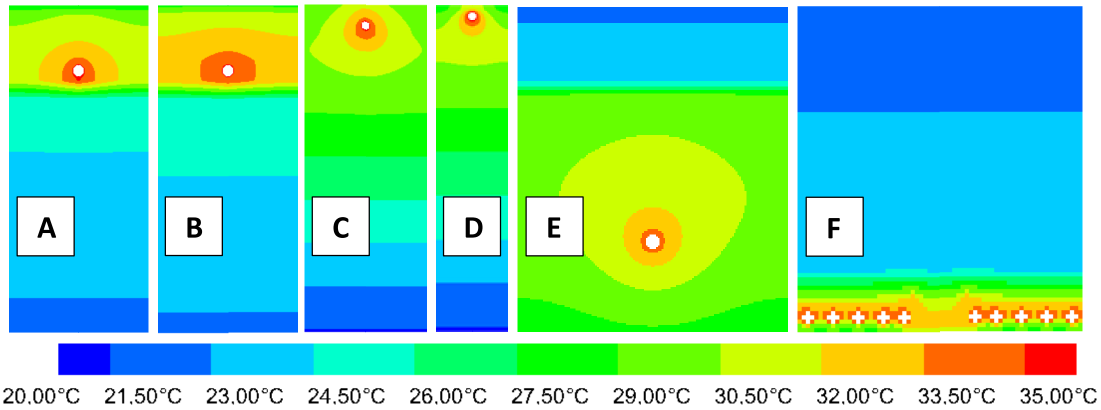

Figure 5. Thermal fields at design heating load: A – Floor, B – Floor with metal fins, C – Wall (TABS), D – Wall (air gap), E – Ceiling (TABS), F – Ceiling (TI).

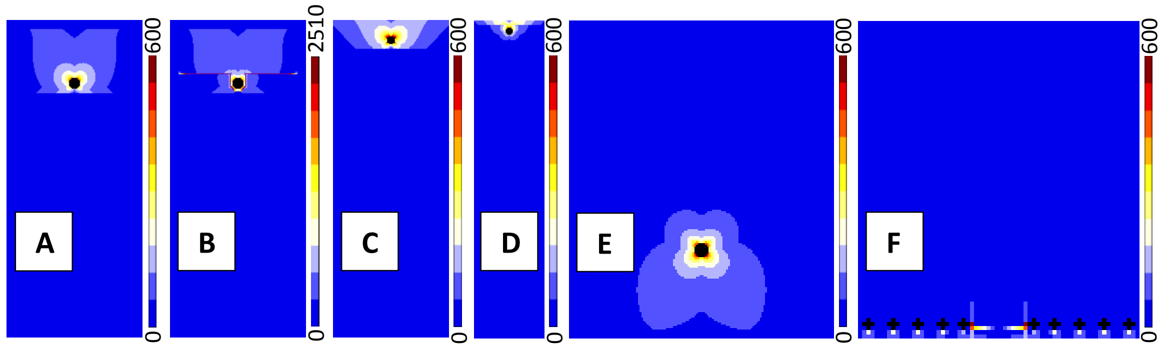

Figure 6. Heat flux distribution at design heating load in W/m^2: A – Floor, B – Floor with metal fins, C – Wall (TABS), D – Wall (air gap), E – Ceiling (TABS), F – Ceiling (TI).

Table 3 shows the sensitivity of thermal output to pipe spacing at the design heat load of 818 W and the corresponding areas of the heating surface. The values in bold correspond to the default design as presented in Figure 2. The thermal output is primarily a function of pipe location – the greatest output is attained for the two wall systems with pipes embedded underneath the surface. The output of wall C is higher than that of wall D since its thicker active layer and the pipes located further from the surface result in a more uniform surface temperature distribution (Figure 5). Consequently, the wall systems need the smallest area of the heating surface to cover the heat load (Table 3). It is shown that even when the spacing is equal for all the systems, e.g. 120 mm, the area of heating surface needed is lowest for wall TABS (C). For the other systems, the differences in the area of heating surface are relatively small except for system F (Ceiling TI) for which only very small pipe distances are meaningful. The maximum surface temperature was up to 29°C for the floor, 32°C for the wall, and 30°C for the ceiling systems which complies with the comfort limits as defined in ISO 7730 [41].

Table 3. Sensitivity of thermal output on spacing of pipes.

Spacing (mm)	A - Floor		B - Floor w. metal fin		C - Wall (TABS)		D - Wall (air gap)		E - Ceiling (TABS)		F - Ceiling (TI)	
	q_i (W/m^2)	A (m^2)	q_i (W/m^2)	A (m^2)	q_i (W/m^2)	A (m^2)	q_i (W/m^2)	A (m^2)	q_i (W/m^2)	A (m^2)	q_i (W/m^2)	A (m^2)
28	--	--	--	--	--	--	--	--	--	--	**52**	**16**
50	--	--	--	--	96	9	84	10	--	--	38	21
80	61	14	61	13	87	9	**71**	**12**	61	13	27	30
100	56	15	59	14	81	10	61	13	59	14	22	37
120	54	15	58	14	**76**	**11**	54	15	58	14	19	44
150	**51**	**16**	**55**	**15**	68	12	46	18	55	15	--	--
200	45	18	51	16	56	15	36	23	52	16	--	--
300	36	23	42	19	40	20	24	33	**45**	**18**	--	--

Key: q_i – thermal output, A – area of heating surface, -- spacing not meaningful/not considered

4.2 Thermal response

Three indicators have been utilized to assess the thermal response and controllability of the heating systems: time constant τ_{63}, response time τ_{90}, and the number of operation cycles. The time constant τ_{63} is commonly used to evaluate thermal response, but it may not fully describe the thermal response especially for radiant systems with complex thermal behaviour [9,24]. Response time τ_{90} was therefore calculated as well to help describe the thermal behaviour. The number of operating cycles provided valuable information about the preferable control strategy.

4.2.1 Time constant τ_{63} and response time τ_{90}

The response time was calculated from step-up curves obtained by switching on the heating system at t = 0 s. Two response times were calculated, τ_{63} and τ_{90}, defined as the time required for the surface temperature to reach 63% and 90% of its maximum value. Figure 7 shows a large variation in the response times of the six systems. The systems are ranked according to their thermal response from shortest to longest as follows: Ceiling (TI), Wall (air gap), Wall (TABS), Floor with metal fin, Floor, and Ceiling (TABS). Adding thermally conductive metal fins to a floor heating system (B) reduced the response time. i.e. enhanced the thermal response as compared to classical floor heating (A).

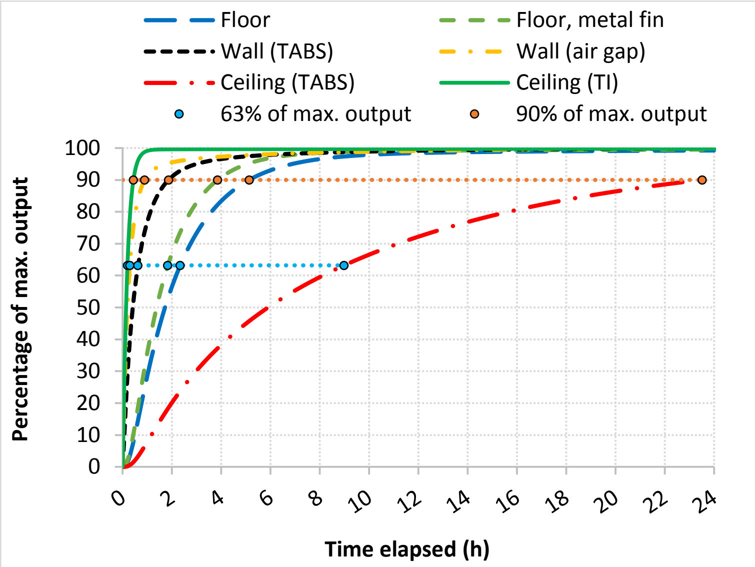

Figure 7. Time constant τ_{63} and response time τ_{90}.

4.2.2 Number of operating cycles

A simplified control strategy was employed to compare the controllability of the systems through the number of operating cycles. This control strategy involved turning the system on and off to keep the thermal output within a defined range over 24 h (Figure 8). The system was turned on at time $t = 0$ s and continuously operated at a constant water temperature of 35°C until the thermal output reached 90% of its maximum value. Then the heating system was turned off until the surface cooled down to the temperature corresponding to 66%, i.e. two-thirds of the maximum thermal output, after which it was turned on again. The number of operating cycles was counted to evaluate the controllability.

Adding metal fins to the floor heating system (A, B), increased the number of operating cycles thereby enhancing the system´s controllability (Figure 8). Wall TABS (C) was easy to control and had the highest thermal output, whereas the wall with an air gap (D) had even better controllability at a slightly lower output. The difference was most remarkable for the ceiling systems. Although their maximum output was similar, ceiling with pipes insulated from the core (F) was operated dynamically whereas ceiling TABS (E) gradually discharged heat providing no possibility of dynamic control.

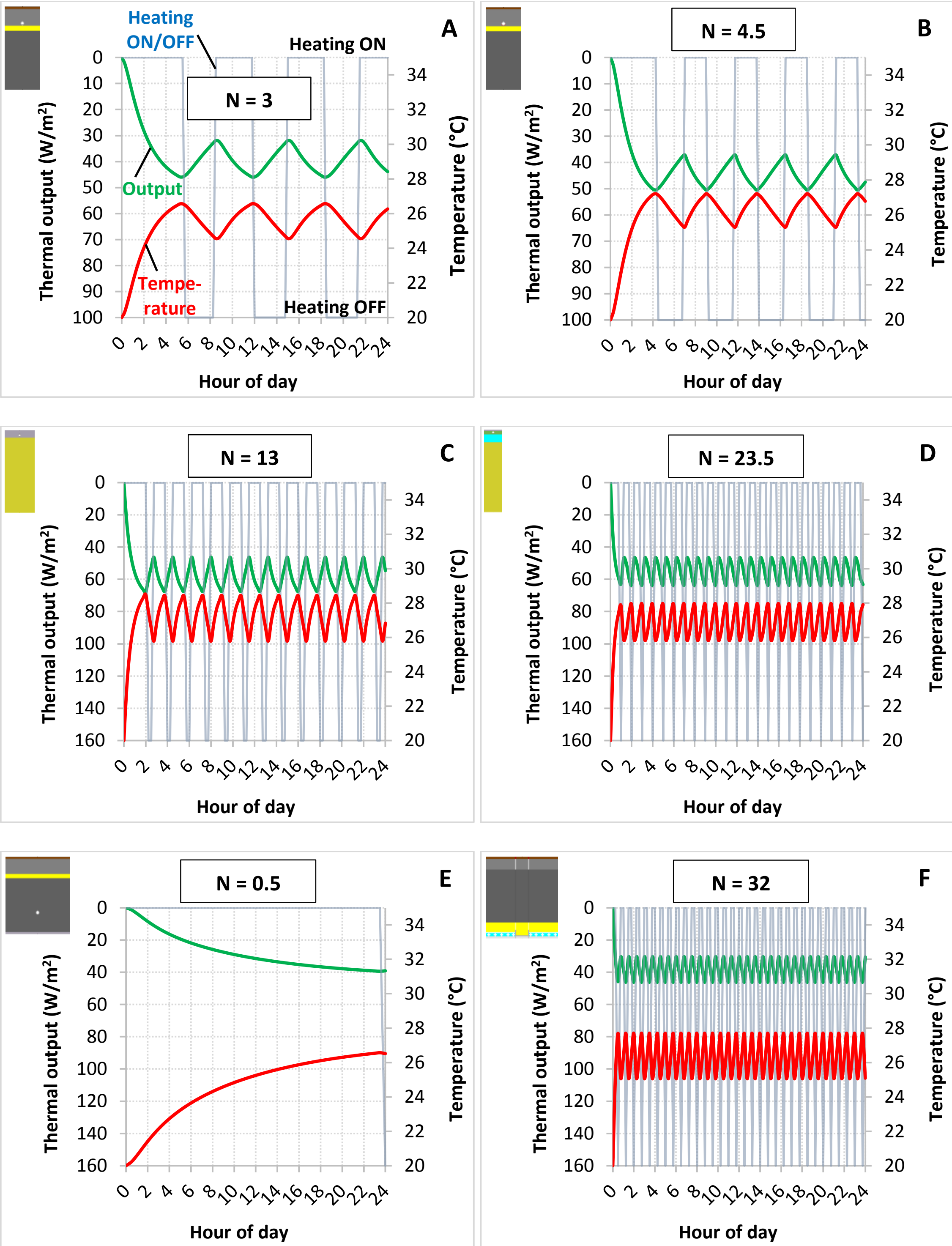

Figure 8. Number of operation cycles (N): A – Floor, B – Floor with metal fins, C – Wall (TABS), D – Wall (air gap), E – Ceiling (TABS), F – Ceiling (TI).

4.3 Thermal energy stored

Energy stored in the structure was computed to evaluate the systems' suitability for discontinuous heat supply. This information is useful when planning the control and operation strategy. The more energy a heating system accumulates, the fewer operating cycles of the heat source may be needed to keep the room temperature within a defined range. In the present study, two levels of thermal storage have been considered: short-term and long-term. The short-term storage was defined as the energy stored in the structure after three hours of operation. It shows the capability of a system to reduce the number of operating cycles of the heat source or cover short outages of the source. The long-term storage was defined as the energy stored in the structure after a 12-hour operation. It indicates the capability of a system to accumulate and discharge heat throughout the day. Such a system can be suitable for example for night heat accumulation.

Figure 9 shows the thermal energy stored in the structure over 24 hours. After three hours of operation, floor heating with metal fins contained most thermal energy, followed by ceiling TABS and floor heating. Most thermal energy was eventually stored in the ceiling TABS because of its great heat storage capacity. Thus, the floor systems are suitable for short-term heat storage, whereas ceiling TABS is suitable for both short- and long-term heat storage. The other systems are not suitable for heat storage and require a heat source with higher heating power and operated intermittently throughout the day. Alternatively, the storage capacity can be provided by a tank.

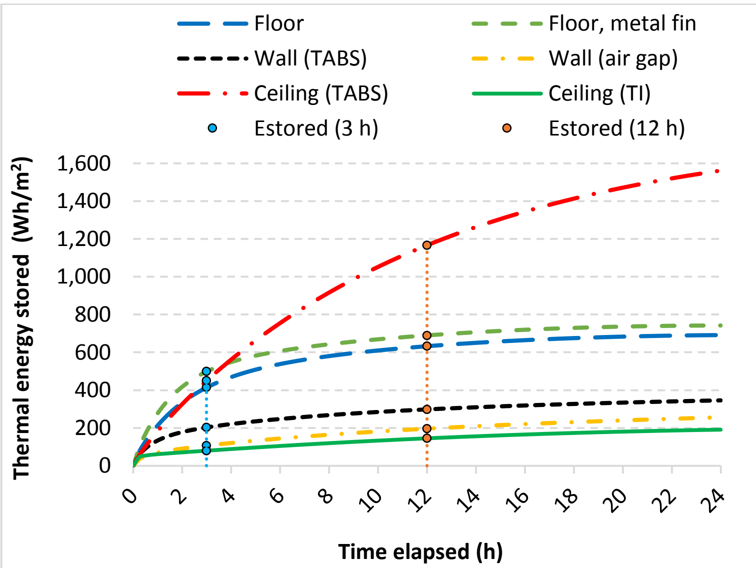

Figure 9. Thermal energy stored (E_{stored}) in the structure over 24 hours of continuous operation.

4.4 Construction costs

Construction costs have been compared using the overall construction costs (€/m^2) and price per thermal output (€/W) as shown in Figure 10. All construction costs refer to EUR (€) per m^2 of the floor area of the living room. The unit prices have been taken from a dedicated database which is routinely used by professionals to calculate construction prices in the Czech Republic [42]. All the costs reported represent standardized costs, not including VAT or any discounts. The prices of materials include the heating element, distribution pipes, control elements and actuators as well as surface finishing. The heat source was not included. The prices also include wages to install the equipment, supporting structures, transfer of materials, and functionality tests. Figure 11 shows standard hours representing a standardized amount of time needed to install the system per m^2 of floor area.

System A is the most common one from all the radiant heating systems considered. Systems B and C are from the technology point of view modifications of system A where the pipes are embedded in a thermally conductive material such as concrete layer or plaster. System F requires installations on the ceiling and the prices were increased correspondingly to reflect the additional effort. In system E, the heating element is an inherent part of the ceiling structure. The price for this system considers the space heating function of the slab, but not the statics.

Figures 10 and 11 show that Wall TABS (C) has the lowest price per thermal output despite taking the most time to construct per m^2. On the other hand, wall system D takes the least time to construct, has the best output per standard hour, but has relatively high overall construction costs because it contains many supporting structures and the components are expensive. The price of systems A, B, D, E per m^2 is similar, whereas ceiling system F is the most expensive. The reason for system F being the most expensive is the high cost of the prefabricated ceiling panels. It is also shown that adding a metal fin in a floor heating system is not too costly which can make this system (B) a feasible option.

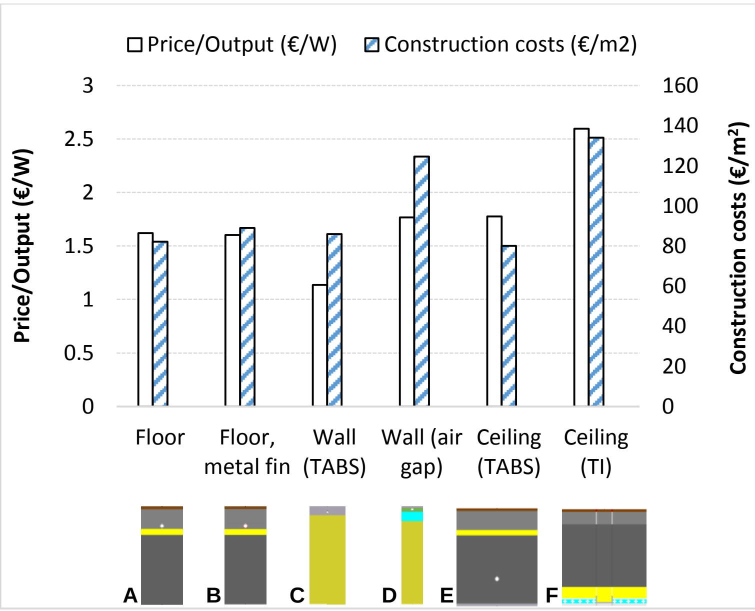

Figure 10. Construction costs of the radiant heating systems.

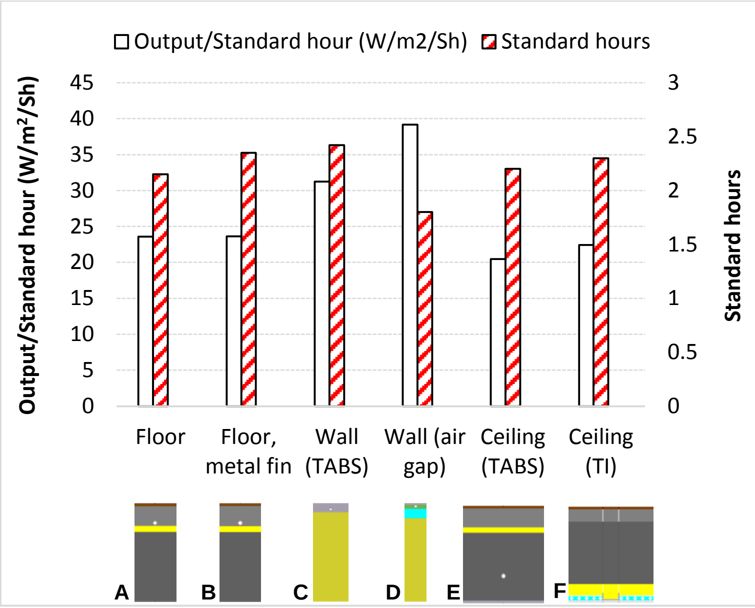

Figure 11. Standard hours needed to install 1 m² of the heating system.

Operating costs can differ for various systems with respect to the differences in thermal output and control strategy. The overall thermal output of the system depends on the area of the heating surface. Therefore, the surface area of each system needs to be adjusted to cover the room design heat loss. The differences in the costs between the systems due to the different thermal output per m² of surface area are therefore reflected in the price per output (Figure 10).

When each system provides the same overall thermal output to cover the room design heat loss, the difference in operating costs will be only caused by differences in the control strategy. In this respect, the advantage of the systems with fast thermal response is a more accurate control. The favourable features of the systems with a slow thermal response are the reduced number of operating cycles and lower peak power of the heat source. An accurate investigation of the effect of control strategy on operation costs would require additional complex calculations and is beyond the scope of this study.

5. Discussion

5.1 Sensitivity of thermal output and thermal response to system design

The parametric simulations have confirmed the sensitivity of thermal output to pipe spacing, h_i, water temperature, and room temperature. The thermal output is also sensitive to the distance of pipes from the surface (Table 3). The effect of concrete and insulation thickness has not been explicitly considered in the present study, but certain guidance is provided in the existing studies. The studies have shown that insulation thickness has a substantial effect on the thermal output for systems with pipes located in the thermal core, whereas the concrete thickness does not. The insulation thickness is of particular importance for the walls having a thermally conductive core [11,35]. Ning et al. [9] found that pipe spacing, concrete type, and concrete thickness have a considerable effect on the response time of TABS with pipes located in thermal core like system E in the present study, whereas room temperature, water temperature, water flow regime, and pipe diameter do not. Mosa et al. [43] emphasize that dendritic flow architecture and a compact geometry of the panels have a positive effect on the heating and cooling capacity.

5.2 Thermal performance of the heating systems

The thermal output is greatest for the wall systems (C, D) with pipes embedded underneath the surface (Table 3, Figure 5). Although the ceiling with pipes attached to plasterboard (F) also has the pipes located underneath the surface, its thermal output is lower because the heat transfer is hindered by locating the pipes in an air gap. Despite the thermal output of system F being lower than that of systems C and D, it has better controllability as indicated by its fast thermal response (Figure 7) and less operating cycles (Figure 8).

The floor systems (A, B) and ceiling TABS (E) tend to store heat as shown in Figure 9. Compared to ceiling TABS, the floor systems have better controllability (Figure 8) while still providing the potential for short-term heat storage. Using metal fin to enhance heat transfer in classical floor heating systems is encouraged because it improves thermal output (Table 3) and controllability (Figure 8) while increasing the heat storage capacity (Figure 11).

A very good performance was attained for wall TABS (C) which provided the greatest thermal output (Table 3) and was easy to control (Figures 7 and 8) despite having the pipes attached to the thermal core. This was caused by the low thermal conductivity of the thermal core made of aerated concrete. This makes Wall TABS (C) suitable for installation on existing walls made of aerated concrete as a part of building retrofit. Similar results can be attained in case of a conductive thermal core if thermal insulation is used to separate the pipes from the thermal core.

5.3 The effect of heat load on thermal performance of the heating systems

The results have been elaborated for the design heating conditions. In reality, the heat load is usually lower due to heat gains and milder climatic conditions. To investigate the effect of heat load reduction on the thermal performance of the heating systems, the simulations were also performed at a higher external temperature than the design value of -12°C. The value of 3.6°C was used, which corresponds to the average external temperature in Brno, Czech Republic in the heating season. This temperature can be considered usual in the region of Central Europe in winter as well as early spring and late autumn. The corresponding temperature of the heating water was determined from the heating curve (Figure 12). The solid line in the middle represents the mean temperature of the heating water, whereas the dashed and dash-dot lines represent the inlet and return water temperatures, respectively. The design temperature gradient at -12°C is 38/32°C and the range of water temperature is narrowing down as the external temperature is increasing and the water temperature is decreasing.

As expected, a lower external temperature lead to a reduction of the thermal output and thermal energy stored. The output dropped by 50±1 % for all the systems. Even though the relative drop in thermal output was about the same for all systems, in absolute values the drop was more rapid for the systems with higher design thermal output. For example, changing the external temperature from -12°C to 3.6 °C resulted in a reduction of the thermal output of the wall system including pipes attached to an insulating core (C) by 38 W/m^2 whereas the output of ceiling TABS (E) dropped by only 22 W/m^2. Decreasing the heat load had negligible influence on the thermal response of the systems as described by the time constant τ_{63} and response time τ_{90}.

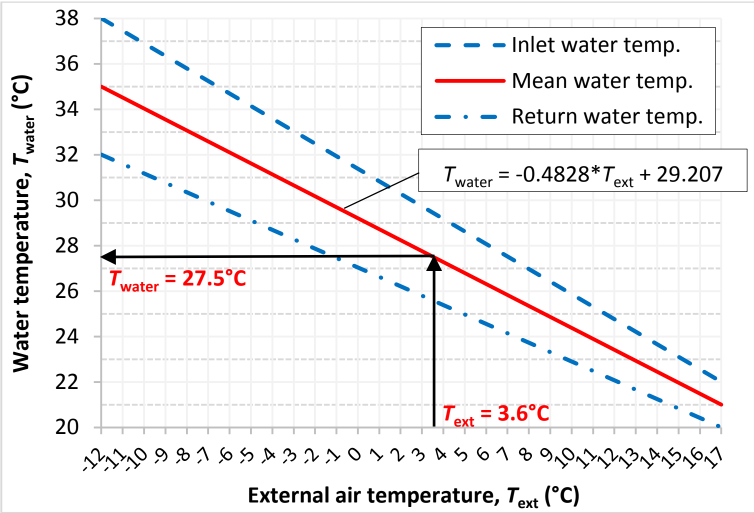

Figure 12. Determination of water temperature from the heating curve.

5.4 Suitability for building retrofit

The suitability of a radiant system for building retrofit depends on criteria such as the heating area required, ease of installation, and the extent of changes induced in the building structures. The advantage of walls over floor and ceiling systems is that they do not reduce the net story height and do not require substantial changes in building structures. They have a rapid thermal response (Figures 7), good controllability (Figure 8), and their thermal output is higher than that of radiant floors and ceilings owing to their construction and higher permissible surface temperatures (Table 3). The disadvantage could be the lower angle factor between the occupant and the wall, and that interventions in the wall need to be done with caution to prevent damaging the pipes.

Floor heating is a realistic solution for building retrofit. Besides creating a homogeneous thermal environment [4,5], its heating capacity is higher than that of ceilings [3], and the angle factor between the floor and the occupant is higher as compared to walls and ceilings. The limitation is that it either reduces the net story height or requires destructing the existing floor.

Ceiling with pipes attached to plasterboard (F) is potentially suitable for building retrofit. As compared to floor heating, no destruction of the floor is needed and it is less limited by obstacles such as the furniture. Its thermal response is fastest from all the heating systems investigated and it provides easy control of the room temperature (Figures 7 and 8). On the other hand, its heating capacity is lower than that of floors and walls [3] and it reduces the story height. Ceiling TABS (E) with pipes located in the core has limited applicability in building retrofit. A system with pipes attached to the surface could be an alternative because it is easy to install and responds faster.

5.5 Overall evaluation

The performance and applicability of the heating systems are summarized in Table 4. The summary given in the table is meant to facilitate the decision-making process to choose the most suitable system for the specific situation. The evaluation is qualitative to a certain extent and reflects the view of the authors which is based on the quantitative indicators elaborated in this study and professional experience. The evaluation is based on Table 3 and Figures 5 and 6 for thermal output, Figures 7 and 8 for controllability, Figure 9 for energy storage, Figures 10 and 11 for construction costs, and the discussion in Section 8.3 for building retrofit.

The summary in Table 4 shows that floor heating performs most consistently in all the aspects evaluated. Adding metal fin has only a minor effect on the investment but it enhances the output and energy storage of the floor heating system. Wall systems are preferable when good controllability and no thermal storage are required. In such a case, system C (wall TABS) is especially suitable for new buildings as well as building retrofit.

Table 4. Performance and applicability of the heating systems.

Heating system		Criterion					
		Thermal output	Controllability	Short-term en. storage	Long-term en. storage	Price/therm. output	Building retrofit
A	Floor	+	+	+	+	+	+
B	Floor w. metal fin	+	+	++	+	+	+
C	Wall (TABS)	++	++	o	o	++	++
D	Wall (TI)	++	++	-	-	+	++
E	Ceiling (TABS)	o	o	+	++	+	-
F	Ceiling (TI)	+	++	+	-	o	+

Key: **++** very good performance; **+** good performance; **o** mediocre performance; **-** not suitable

1) Applies to a thermally insulating thermal core. In case of a conductive core, the evaluation may be applicable if the pipes are thermally decoupled from the core by insulation.

2) Applies to pipes embedded in the structure. The system may be well applicable for retrofit if the pipes are attached to the core's surface.

6. Conclusion

At present, holistic comparisons of various types of radiant systems that would guide the selection of the most suitable heating system for the specific situation are lacking. The results elaborated in this study should therefore facilitate the process of selecting the most convenient radiant heating system for both newly constructed and renovated buildings. To accomplish this, a comparative study of six representative radiant wall, floor and ceiling systems has been conducted in terms of their thermal output, area of the heating surface required, controllability, short-term and long-term energy storage, suitability for installation in existing buildings, and construction costs. Temperature and heat flux distribution was calculated to determine the thermal output and area needed. Time constant (τ_{63}), response time (τ_{90}), and the number of operation cycles were used to evaluate the controllability. Thermal energy stored revealed the ability of short- and long-term heat storage, whereas construction costs indicated affordability of the systems. The conclusions are summarized as follows:

- The wall systems (C, D) were preferable when good controllability and no thermal storage were required. Due to their versatility and fast response, they are especially suitable for building retrofit. Wall TABS (C) with a thermally insulating thermal core also provided a certain potential for thermal storage while being the most affordable of all the systems.
- Floor heating (A) performed most consistently in all the aspects evaluated indicating its universal use. Adding metal fins between pipes and the concrete spread layer (B) had only a minor effect on the investment costs and it enhanced the thermal output and controllability while increasing storage capacity.
- Ceiling with pipes insulated from the thermal core (F) performed well when thermal storage was not required and is potentially suitable for building retrofit. Ceiling TABS (E) was the costliest system but might be feasible when long-term heat storage is needed. An alternative with pipes attached to the core’s surface could be preferable.
- Thermal response is a more important parameter to consider than thermal output. Maximum thermal output can be adjusted by variating the pipe spacing and active surface area. The thermal response is an inherent characteristic of the heating system, which has a significant effect on its controllability and the room’s thermal balance.

Future work should include CFD simulations of a room with radiant surfaces and realistic occupancy profiles. The simulations would provide additional information about temperature distribution and air velocity fields in the room. Besides, the calculations should be extended to cooling operation and the year-round operation of the systems should be discussed.

Acknowledgements

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