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Insulation panels for active control of heat transfer in walls operated as space heating or as a thermal barrier: numerical simulations and experiments

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Highlights

- Thermally active wall with pipes arranged in thermal insulation was investigated.
- The wall’s function can be alternated between space heating and a thermal barrier.
- The system has the potential to reduce heat loss when used as a thermal barrier.
- Heating capacity is reduced compared to walls with pipes outside thermal insulation.
- Embedding pipes in thermally conductive material is crucial for heating capacity.

Abstract

Numerical simulations and experiments were performed for a thermally active wall with pipes arranged in milled channels in the thermal insulation. The advantage of this system is its suitability for installation in both new and existing buildings in the form of precast heat insulation panels attached to their facades. The study shows that by active control of the supply water temperature, it is possible to alternate the wall’s function between space heating and a thermal barrier. The wall system has the potential to significantly reduce heat loss when used as a thermal barrier. When operated as space heating, embedding the pipes in thermal insulation reduced the heating capacity by 50% as compared to systems with pipes arranged in a concrete core and by 63% for pipes arranged in a layer underneath the surface. It is crucial that pipes arranged in channels are embedded in a thermally
conductive material. Failing to do so can substantially diminish the heating capacity due to the imperfect contact between the pipes and radiant surface and also due to the air gap that may form around the pipes. The thickness of the thermal insulation, spacing of the pipes, and supply water temperature also have a substantial effect on the heating capacity, whereas the thickness of the concrete core does not.

**Keywords**

Radiant heating system; Wall heating; Thermally activated building systems (TABS); Thermal barrier; Thermal insulation; Heat flux

**Nomenclature**

**Abbreviations**

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>E</td>
<td>exterior</td>
</tr>
<tr>
<td>I</td>
<td>interior</td>
</tr>
<tr>
<td>SH</td>
<td>space heating</td>
</tr>
<tr>
<td>T1, T2</td>
<td>temperature sensors for supply and return water temperature, respectively</td>
</tr>
<tr>
<td>TABS</td>
<td>thermally active building system(s)</td>
</tr>
<tr>
<td>TB</td>
<td>thermal barrier</td>
</tr>
</tbody>
</table>

**Symbols**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>c</td>
<td>specific heat capacity at constant pressure (J/(kg.K))</td>
</tr>
<tr>
<td>d</td>
<td>thickness of layer (m)</td>
</tr>
<tr>
<td>f</td>
<td>index denoting surrounding fluid</td>
</tr>
<tr>
<td>h</td>
<td>overall heat transfer coefficient between radiant surface and environment (W/(m².K))</td>
</tr>
<tr>
<td>l</td>
<td>characteristic dimension (m)</td>
</tr>
<tr>
<td>n</td>
<td>index denoting a line perpendicular to surface</td>
</tr>
<tr>
<td>Nu</td>
<td>Nusselt number (-)</td>
</tr>
</tbody>
</table>
\( q \)  
heat flux to the interior (W/m\(^2\))

\( q_e \)  
heat flux to the exterior (W/m\(^2\))

\( q \)  
overall amount of energy transferred from pipes to wall (W/m\(^2\))

\( S \)  
internal heat source (W/m\(^3\))

\( T \)  
temperature (K)

\( U_{\text{wall}} \)  
heat transfer coefficient of wall (W/m\(^2\).K)

\( w \)  
index denoting surface of an object

\( \alpha \)  
convective heat transfer coefficient for the water and pipe surface (W/(m\(^2\).K))

\( \Delta \theta \)  
difference between the surface temperature of the heating surface and the room air temperature (K)

\( \theta_{\text{amb}} \)  
ambient air temperature (°C)

\( \theta_i \)  
room air temperature (°C)

\( \theta_{\text{op}} \)  
operative temperature (°C)

\( \theta_{\text{sup}} \)  
supply water temperature (°C)

\( \theta_w \)  
temperature of water in heating pipes (°C)

\( \lambda \)  
thermal conductivity (W/(m.K))

\( \lambda_L \)  
thermal conductivity of the fluid (W/(m.K))

\( \rho \)  
bulk density (kg/m\(^3\))

\( \rho_w \)  
volumetric weight (kg/m\(^3\))

\( \tau \)  
time (s)

1. Introduction

Current trends in the design and operation of heating, ventilation and air conditioning include the increasingly frequent use of low-exergy water-based radiant systems. As opposed to all-air systems, the task of providing thermal comfort is limited to radiant systems, while the provision of ventilation is separate, i.e., the radiant system covers heat losses in the winter and eliminates heat gains in the summer, whereas fresh air is supplied by a separate air system [1,2,3].

Although research on radiant surfaces has mostly focused on structural floors and ceilings, evidence from several research studies suggests that radiant walls also present a potentially feasible
solution for space heating (SH). Karabay et al. (2013) [4] recommend considering wall heating rather than floor heating since a better thermal performance and comfort conditions can be achieved with a lower water temperature, thereby reducing fuel consumption. Myhren and Holmberg (2008) [5] conclude that floor heating, radiators and wall heating are all capable of creating a comfortable indoor environment in a well-insulated room, although one problem with radiant systems can be the weaker counteraction of cold downflow from the air supply units. For a detached house powered by natural gas, Bojic et al. (2013) [6] indicate a preference for wall heating over both floor and ceiling heating in terms of energy and exergy consumption, destroyed exergy, CO₂ emissions, and operating costs as well as the nominal power of the boiler. Radiant walls are more efficient in terms of heat and cool emission than heated ceilings and cooled floors, respectively, and they have higher heating capacity per surface area than floor heating due to a wider range of permissible surface temperatures [7,8].

Contemporary research is focused on traditional wall heating solutions with pipes that are thermally insulated from the main building structure. The present study is aimed at radiant walls that work with emitting elements that are thermally coupled to the building structure and that represent a thermally active building structure (TABS) [9,10]. The major advantage of these systems is their large energy storage capacity, which permits reducing peak heating or cooling power by shifting peak loads to periods in which the system works more efficiently [11,12]. Typical examples of TABS are systems with pipes embedded in a massive concrete core (Fig. 1a) and systems with capillary mats embedded in a layer on the inner surface (Fig. 1b) [13]. In this study an alternative design is investigated, where a system of pipes arranged in milled channels in thermal insulation is attached to the bearing structure of a building in the form of precast insulation panels (Fig. 1c) according to a patent [14]. The advantage of this solution is its potential suitability for installation in existing buildings as a part of retrofitting. If the bearing structure has enough storage capacity, the system will behave as a TABS.
Besides SH, it is also possible to use a wall system as a thermal barrier (TB) to reduce transmission heat losses. An example of a wall system designed to serve as a TB is shown in Fig. 1d [15]. The system acts as a TB when the temperature of the fluid in the pipes is lower than necessary for heating, but is still high enough to reduce the heat loss. As an option, an additional absorption layer can be arranged on the insulating layer to collect thermal energy during the summer months (Fig. 1e) [15]. Doležel [16] estimates that in a passive house [17], the heat loss is lower for a wall with a TB than for a conventional external wall when the ambient temperature drops below -2.5°C for pipes arranged in a concrete layer (Fig. 1f), below 6.3°C for pipes embedded between bricks and thermal insulation (Fig. 1g), and below 9.6°C for pipes between an OSB plate and thermal insulation (Fig. 1h). Despite the high thickness of the thermal insulation used in [16], the main advantage of TB is a reduction of the thickness of the outer insulating layer, which should permit the total thickness of the wall to be less than the insulation of a conventional highly insulated exterior wall [15]. Thus, together with aerogel [18,19] and vacuum insulation panels [20,21], TB presents an alternative to conventional insulations made of, e.g., mineral wool or polystyrene, which allows for a lower thickness of a wall due to its ability to actively control the heat transfer.

Compared to traditional insulation methods, TB requires a fluid of a certain temperature to circulate in the pipes. To ensure that the energy to heat the water in the pipes and the auxiliary energy does not exceed the energy-saving benefits, it is recommended that the energy is harvested by
renewable energy sources such as a solar roof or collectors and that the excess is fed to an underground reservoir as soon as the outdoor temperature exceeds the indoor temperature. During the winter, the building can be fed from this reservoir [15,22]. Krzaczek and Kowalczuk [22] showed that combining TB (Fig. 1i) with geothermal energy storage and fixing its temperature close to 17°C all year round can help reduce the heat losses through external walls to one third as compared to traditional insulated walls. Xie et al. [23] showed that for a building envelope structure consisting of two brick layers (Fig. 1j) supplied from a low-grade energy source, a TB in a hot climate can decrease the heat transfer from outside to the interior space to almost zero and thereby reduce the energy consumption significantly.

Existing research has indicated the insulation potential of TB. However, the heat transfer in a TB has not been fully explored, and a detailed comparison with SH is lacking as well. Furthermore, the existing studies focus on systems specifically designed to serve as TB intended for new buildings. Systems that can serve both as SH and a TB and are intended for use both in new and in existing installations have not been previously investigated.

This paper studies the possibilities of the active control of heat transfer by insulation panels in walls operated as SH or TB as shown in Fig. 1c. The main contributions are summarized as follows:

(1) The originality of the proposed system is its potentially universal use as either TB or SH, depending on the system’s configuration and operating conditions. Unlike previously investigated systems, this solution is suitable for installation in both new and existing buildings.

(2) The proposed TABS wall presents an alternative to the more common floor and ceiling TABS. It is also an alternative to the more typical TABS solutions shown in Figs. 1a and 1b. The heat transfer in the proposed TABS wall is explored and compared to a typical TABS wall.

(3) To facilitate the design of this system, the effect of different parameters, such as the supply temperature, the thickness of the concrete and insulation, the spacing of pipes, and the thermal conductivity of the material surrounding the pipes, on the heating capacity is investigated.

(4) Published scientific studies on TB are scarce. This study adds to the existing knowledge and explores the insulation potential of a particular TB design. The TB is researched and compared with a traditionally insulated wall and with a SH in terms of heat transfer.

2. Cases investigated and methodology
The heat flux, surface temperatures, and other parameters necessary to describe the heat accumulation and discharge process of a wall were investigated by numerical simulations using dedicated software and by experiments performed on a wall fragment exposed to ambient conditions simulated in climate chambers. An overview of the intended boundary conditions and parameters investigated is shown in Tab. 1, together with indications of the research method used.

All the simulations and experiments were performed at a room air temperature of 20°C. The ambient temperature and the supply water temperature were varied as shown in Tab. 1. The room air temperature of 20°C was selected as it is frequently used by designers as the lowest acceptable temperature limit in the winter. As pointed out by Kalmár and Kalmár (2012) [24], the mean radiant temperature in rooms with low-temperature radiant systems can be lower than the room air temperature, depending on the room’s geometry. At a room air temperature of 20°C, this might result in an operative temperature (θ_{op}) slightly less than that required for new and reconstructed buildings [25,26]. This θ_{op} does not affect the applicability of the results to spaces with an optimum θ_{op}, which may be 3-4°C higher.

The ambient temperature of 4.2°C represents the mean outdoor air temperature in the winter and is typical of the temperate Central European climate of Bratislava, Slovakia. The ambient temperature of -11°C represents extreme conditions that are used when designing a heating system [27]. The supply temperature was equal to the room air temperature in cases with a TB and was above 20°C when the system was operated as SH.

The supply water temperature of 30°C used in the experiments can be considered typical for water-based radiant heating systems operated in thermally insulated buildings under average winter conditions (θ_{amb} = 4.2°C). It is also realistic in thermally insulated buildings under design winter conditions (θ_{amb} = -11°C). Water with this temperature can usually be supplied by a renewable energy source [12,28,29]. The supply temperature of 25°C in stationary simulations is used to study and emphasize the effect of even a relatively small increase in temperature on the heat transfer (SH versus TB). It should also emphasize the effect of the piping arrangement on the heat transfer for a surface temperature close to the room temperature. To observe the effect of variations in the supply temperature on the heating capacity, dynamic simulations are performed under periodically changing ambient temperatures to account for dynamic changes in weather conditions.
The effect of the spacing of pipes and the thickness of the concrete and insulation was studied under design ambient conditions to provide recommendations regarding the optimum system design. The effect of the material surrounding the pipes was additionally studied after the experiments had been performed. The aim was to explain and emphasize the important role of the material surrounding the pipes with respect to the experimental results.

<table>
<thead>
<tr>
<th>Operation mode</th>
<th>Ambient temp. $\theta_{amb}$ (°C)</th>
<th>Room air temp. $\theta_{i}$ (°C)</th>
<th>Supply temp. $\theta_{w}$ (°C)</th>
<th>Simulation</th>
<th>Heat flux (W/m²)</th>
<th>Wall thickness (mm)</th>
<th>Spacing of pipes (mm)</th>
<th>$\lambda$ of material surrounding pipes (W/(m.K))</th>
</tr>
</thead>
<tbody>
<tr>
<td>TB</td>
<td>(-11)</td>
<td>20</td>
<td>20</td>
<td>stationary</td>
<td>+</td>
<td>-</td>
<td>-</td>
<td>+</td>
</tr>
<tr>
<td>TB</td>
<td>(-14) - 0</td>
<td>20</td>
<td>20</td>
<td>dynamic</td>
<td>+</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>SH</td>
<td>(-11)</td>
<td>20</td>
<td>25</td>
<td>stationary</td>
<td>+</td>
<td>+</td>
<td>+</td>
<td>-</td>
</tr>
<tr>
<td>SH</td>
<td>(-14) - 0</td>
<td>20</td>
<td>25 - 40</td>
<td>dynamic</td>
<td>+</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>SH</td>
<td>4.2</td>
<td>20</td>
<td>30</td>
<td>stationary</td>
<td>+</td>
<td>+</td>
<td>-</td>
<td>+</td>
</tr>
</tbody>
</table>

Key: + experiment performed / parameter investigated; - experiment not performed / parameter not investigated;

TB – thermal barrier, SH – space heating

Tab. 1 Intended boundary conditions and parameters investigated

3. Numerical simulations of thermally active walls

The calculations relating to heat accumulation and discharge, heat flux, and surface temperatures were computed by stationary and dynamic numerical simulations, using CalA 3.0 [30,31] software, which has been validated in accordance with EN ISO 10211 [32].

3.1 Calculation principle

CalA 3.0 software was primarily developed to simulate stationary and dynamic 2D heat transfer by conduction [33]:

$$\frac{\partial}{\partial x} \left( \lambda \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left( \lambda \frac{\partial T}{\partial y} \right) + S = \rho \cdot c \cdot \frac{\partial T}{\partial \tau}$$

(1)
where $T$ is the temperature (K); $S$ is an internal heat source (W/m$^3$); $t$ is time (s); $\lambda$ is thermal conductivity (W/(m.K)); $\rho$ is bulk density (kg/m$^3$); and $c$ is the specific heat capacity at a constant pressure (J/(kg.K)).

The thermophysical properties of materials are considered to be constant, isotropic, and temperature independent in all the simulations. The boundary conditions defining the specific heat flux on the surface of a computational domain are calculated according to Newton's law of cooling (2), assuming the adiabatic wall boundaries (3) as shown in Fig. 2 [33]:

\[
-\lambda \left( \frac{\partial T}{\partial n} \right)_{w} = h(T_w - T_f) 
\]

\[
-\lambda \left( \frac{\partial T}{\partial n} \right)_{w} = 0 
\]

where $w$ is an index denoting the surface of an object; $f$ is an index denoting the surrounding fluid; $n$ is an index denoting the direction perpendicular to the surface; and $h$ is the heat transfer coefficient (W/(m$^2$.K)), including both the convection and thermal radiation heat transfer from a radiant surface to the surrounding environment.

![Diagram of boundary conditions defining specific heat flux on a wall surface](image)

**Fig. 2 Boundary conditions defining specific heat flux on a wall surface**

The convective heat transfer coefficient for the water and the pipe surface is calculated by [33]:
\[ \alpha = Nu \cdot \frac{\lambda}{l} \]

where \( \lambda \) is the thermal conductivity of the fluid (W/(m.K)); \( l \) is a characteristic dimension (m). \( Nu \) is the Nusselt number, which represents the ratio of the convective to the conductive transfer, which is determined as a function of the Grashof, Prandtl, and Reynolds numbers [33,34].

The heat transfer coefficient for the water and pipe surface was determined to be 1218 W/(m\(^2\).K) according to eq. (4). The total heat transfer coefficient \( h \) (combined convection and radiation) between the radiant surface and the space recommended by the standard is 8 W/(m\(^2\).K) for a heated wall [13]. In an experimental room (6 m x 4 m x 3 m) with one window, radiant heating panels that covered one of the walls, and no mechanical ventilation, Koca et al. (2014) [35] measured a total heat transfer coefficient of 8.20 at a surface temperature equal to 24.58°C and an air temperature equal to 20.06°C. In our simulations, the total heat transfer coefficient between the radiant surface and space is assumed to be 8 W/(m\(^2\).K). The external convective heat transfer coefficient is 25 W/(m\(^2\).K), which corresponds to a wind speed of about 5 m/s calculated according to the simplified method in EN ISO 6946 [36], 2 m/s according to the model by Hagishima & Tanimoto [37], and 2.2 m/s using Sturrock’s model [38,39]. The wind speed represents a light to gentle breeze on the Beaufort scale [40]. For insulated buildings, such a wind speed should have little or no impact on the heat transfer coefficient on the inner surface [41].

### 3.2 Physical model of the wall

Fig. 3 shows a physical model of the wall fragment as defined by the software. The pipes (6), which represent the active facade element, are embedded in the thermal insulation (4) attached to the concrete core (2) by adhesive mortar (3). If not stated otherwise, the space between a pipe and the thermal insulation is filled by cement mortar (7). The thermophysical characteristics of the materials in Fig. 3 are shown in Tab. 2.
Fig. 3 Physical model of the wall fragment used in the numerical simulations

<table>
<thead>
<tr>
<th>No.</th>
<th>Material</th>
<th>Thickness</th>
<th>Volumetric weight</th>
<th>Thermal conductivity</th>
<th>Specific heat capacity</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Inner plaster</td>
<td>0.01</td>
<td>1600</td>
<td>0.88</td>
<td>840</td>
</tr>
<tr>
<td>2</td>
<td>Reinforced concrete</td>
<td>0.1 - 0.2</td>
<td>2400</td>
<td>1.58</td>
<td>1020</td>
</tr>
<tr>
<td>3</td>
<td>Adhesive mortar</td>
<td>0.01</td>
<td>1300</td>
<td>0.8</td>
<td>840</td>
</tr>
<tr>
<td>4</td>
<td>Insulation EPS F</td>
<td>0.1 - 0.2</td>
<td>17</td>
<td>0.04</td>
<td>1270</td>
</tr>
<tr>
<td>5</td>
<td>Outer plaster</td>
<td>0.01 - 0.03</td>
<td>1300</td>
<td>0.8</td>
<td>840</td>
</tr>
<tr>
<td>6</td>
<td>Plastic pipe ø20 mm</td>
<td></td>
<td>1200</td>
<td>0.35</td>
<td>1000</td>
</tr>
<tr>
<td>7</td>
<td>Cement mortar</td>
<td></td>
<td>2000</td>
<td>1.16</td>
<td>840</td>
</tr>
</tbody>
</table>

Tab. 2 Thermophysical characteristics of the simulated wall

3.3 Results of the numerical simulations

The effect of the piping arrangement, the thickness of the wall, and the spacing of the pipes on the heat flux was investigated through the stationary parametric simulations. The dynamic simulations were used to investigate the processes related to the thermal inertia of the wall.

3.3.1 Heat transfer for the various piping arrangements

Fig. 4 compares the heat flux and temperature distribution within the wall fragment that was used to operate as wall heating for different piping arrangements. From the total amount of energy transferred to the wall \(q_t\), the part transferred to the interior \(q_i\) increases, while the heat lost to the ambient environment \(q_e\) drops as the pipes are gradually moved from the insulation (a) to the inner surface layer (c). The same trend applies to the ratio of the usable heat flux \(q_i\) to the heat loss \(q_e\) as shown in Fig. 5.
All the heat flux values refer to a square meter of the wall surface area and not a square meter of the pipe area. The supply water temperature is 25°C; the pipe spacing is 150 mm; and the thickness of the concrete core is 200 mm as is the thickness of the thermal insulation. The internal plaster is 30 mm thick for case (c) with the heating pipes embedded in the plaster and 10 mm thick in the other two cases (a, b).

![Fig. 4 Heat flux and temperature distribution for a heating wall according to different piping arrangements. Boundary conditions: $\theta_{amb} = -11^\circ C$, $\theta_i = 20^\circ C$, $\theta_w = 25^\circ C$](image)

![Fig. 5 Percentages of usable heat flux ($q_i$) and heat loss ($q_e$) for different piping arrangements](image)

### 3.3.2 Space heating versus a thermal barrier
Fig. 6 shows how the heat loss through an insulated wall (a) diminishes to almost zero through active control of the heat transfer by TB (b). The heat flux from the pipes visualized by the arrows counterbalances the heat flux from the interior to the outside, but is not strong enough to heat the space. The wall acts as SH by increasing the supply temperature above the room’s air temperature (c); there is no heat loss through the wall from the space; it is only from the piping.

As stated above, the pipe spacing is 150 mm, and the thickness of the concrete core is 200 mm as is the thickness of the insulation. The pipe surface in case (a) with no active element is considered adiabatic.

<table>
<thead>
<tr>
<th>Case</th>
<th>Heat Flux at Interior</th>
<th>Heat Flux at Exterior</th>
</tr>
</thead>
<tbody>
<tr>
<td>a) no active element</td>
<td>$q_i = 0 \text{ W/m}^2$</td>
<td>$q_i = 0 \text{ W/m}^2$</td>
</tr>
<tr>
<td>b) thermal barrier ($\theta_w = 20^\circ\text{C}$)</td>
<td>$q_i = 5.9 \text{ W/m}^2$</td>
<td>$q_i = 5.8 \text{ W/m}^2$</td>
</tr>
<tr>
<td>c) space heating ($\theta_w = 25^\circ\text{C}$)</td>
<td>$q_i = 17.9 \text{ W/m}^2$</td>
<td>$q_i = 17.9 \text{ W/m}^2$</td>
</tr>
</tbody>
</table>

Fig. 6 Heat flux and temperature distribution for a) no active element, b) TB and c) SH. Boundary conditions: $\theta_{\text{amb}} = -11^\circ\text{C}$, $\theta_i = 20^\circ\text{C}$

### 3.3.3 Supply water temperature variations

The delay between the maximum ambient temperature ($\theta_{\text{amb}}$) and the maximum heat flux to the interior ($q_i$) is about six hours (Fig. 7). Fig. 7 shows that the supply temperature ($\theta_w$) clearly affects the magnitude of the heat flux to the interior but has little effect on the delay between $\theta_{\text{amb}}$ and $q_i$. The graphs present a fragment taken from a 7-day simulation, where the ambient temperature was periodically varied from -14°C at night up to 0°C at midday. The wall system was always turned on at
5:00 a.m. and turned off at 10:00 p.m. The thickness of the concrete core of the simulated wall was 100 mm, and the thickness of the insulation was 150 mm as is the pipe spacing. The room air temperature was kept constant at 20°C.

![Graph](image)

Fig. 7 Heat flux during a winter day a) to the interior and b) to the exterior

### 3.3.4 Thickness of the insulation and spacing of the pipes

Fig. 8 shows the substantial effect of the thickness of the insulation on the heat flux both to the interior and exterior under design climatic conditions ($\theta_{\text{amb}} = -11^\circ\text{C}$), especially for the thickness of the insulation between 50 and 100 mm. On the other hand, the effect of the thickness of the concrete, which is represented by different lines, is relatively small.

The effect of the spacing of the pipes on the heat flux, as shown in Fig. 9, is mainly significant for the lowest amount of insulation. The results refer to a constant thickness of the concrete of 100 mm with variable thicknesses of the thermal insulation, as represented by the different lines.
Fig. 8 Effect of thickness of wall on the heat flux a) to the interior, b) to the exterior. Boundary conditions: $\theta_{\text{amb}} = -11^\circ\text{C}, \theta_i = 20^\circ\text{C}, \theta_w = 25^\circ\text{C}$

![Graph showing effect of thickness on heat flux to interior and exterior](image)

Fig. 9 Effect of spacing of pipes on the heat flux a) to the interior and b) to the exterior. Boundary conditions: $\theta_{\text{amb}} = -11^\circ\text{C}, \theta_i = 20^\circ\text{C}, \theta_w = 25^\circ\text{C}$

![Graph showing effect of pipe spacing on heat flux to interior and exterior](image)

4. Experimental measurements of the thermally active wall

Experimental measurements were carried out on a thermally-insulated concrete fragment, which represented the external wall of, e.g., a house. The experiments were performed to extend and to also partially confirm the results of the numerical simulations.

4.1 Experimental design

The dimensions of the fragment were 1140 mm x 1360 mm (Fig. 10). The wall was located between two climate chambers with controlled air temperatures and humidity: one chamber simulated the indoor environment, while the other simulated the outdoor climatic conditions. During the experiments, the desired air temperature in both climate chambers was achieved by a fan, which circulated the air in a downward direction. The heat transfer coefficients were therefore higher than those corresponding to natural air convection. The thermally active element was represented by 20-mm pipes made of aluminium-plastic, which were pressed into channels milled in expanded polystyrene as suggested in the patent [14]. The calculated heat transfer coefficient of the wall ($U_{\text{wall}}$) was 0.35 W/(m².K). Solar radiation, wind, and rain were not considered in this study. With respect to
the low levels of solar radiation in the winter and the sufficient thickness of the insulation, ignoring climatic conditions was assumed to have only a small impact on the applicability of the results. The temperature of the concrete was monitored by PT100 platinum resistance thermometers located at selected points along the panel (Fig. 10, points A, B, C, D) at several depths (Fig. 10, points 1 to 5), together with the supply and return water temperatures. The heat flux was monitored by a thermopile sensor for studies of the radiative and convective heat flux with a level of accuracy variable in the range ±5% of the value measured. The sensor was located underneath the surface in the center of the fragment as recommended by [18] (Fig. 10).

4.2 Results of the experimental measurements

One experiment was performed for TB under design ambient conditions, and two experiments were done for SH, i.e., one under design and one under average ambient conditions.

4.2.1 Space heating at the design ambient temperature
This measurement involved the operation of the wall fragment as SH ($\theta_i = 20^\circ$C, $\theta_w = 30^\circ$C) under design ambient conditions ($\theta_{amb} = -11^\circ$C). The wall heating system was turned on after the boundary conditions had reached a steady state, and it was turned off shortly after the heat flux reached a constant value of about 4.5 W/m$^2$ (Fig. 11). The time constant, expressed as the time for the heat flux to reach 63.2% of its final value, was about four hours. In Fig. 11, point B1 represents the contact between the piping and the concrete core, and points A5 and B5 represent the wall fragment’s inner surface. The heat transfer coefficient between the radiant surface and the space ($h$) is likely to be higher than the 8 W/(m$^2$.K) listed in [13] because of the air circulating in the experimental chamber. Due to the very small difference between the temperature of the heating surface and the room air temperature ($\Delta \theta$), the uncertainty in $h$ is high, and its true value is estimated to be in the interval $18.2\pm9$ W/(m$^2$.K).

The heat flux measurements were also performed for the supply water temperature of 35°C, with a resulting heat flux of 13 W/m$^2$, thus exceeding the former value by 8 W/m$^2$.

![Fig. 11 Heat flux and temperatures for a thermally active wall operated as SH under design ambient conditions](image)

4.2.2 Space heating at average ambient temperature

In a subsequent measurement the wall fragment was operated as SH ($\theta_i = 20^\circ$C, $\theta_w = 30^\circ$C) under an average ambient temperature of 4.2°C. The heating system was started on 6 August at 9.00 a.m.
and powered off after about 24 hours, when the heat flux was in a steady state (Fig. 12). The heat flux measured was 8.5 W/m², and the time constant calculated from the gradually increasing heat flux curve was about four hours. The true value of \( h \) is estimated to be in the interval 29.5±13 W/(m².K).

![Graph showing heat flux and temperatures for thermally active wall operated as SH at average ambient conditions](image)

**Fig. 12** Heat flux and temperatures for thermally active wall operated as SH at average ambient conditions

### 4.2.3 Thermal barrier at design ambient temperature

Fig. 13 presents the heat flux and associated temperatures for the wall fragment operated as a TB (\( \theta_i = 20^\circ C, \theta_w = 20^\circ C \)) under design ambient conditions (\( \theta_{amb} = -11^\circ C \)). Shortly after the boundary conditions reached a steady state and the TB was turned on, a drift in supply temperature occurred due to the short distance of the heat generator from the fragment. After action had been taken to restore the balance, the supply water temperature was stabilized at 20°C. Based on the trend in Fig. 13, the heat flux in a steady-state condition is estimated to have a value very close to -5 W/m².
4.3 Heat conduction between the pipes and concrete core

Fig. 14a shows the simulated heat flux and temperature distribution based on the boundary conditions measured during the experiment for SH at an average ambient temperature (Fig. 12). The heat transfer coefficient between the radiant wall and experimental chambers was estimated to be 25 W/(m².K) on both sides of the wall. The position of the pipes in the insulation was simplified by assuming a uniform distribution of the air gap around the pipes. The thermal resistance of the air gap was adjusted so that the results of the experiments corresponded to the simulation.

The useful heat flux \(q\) is rather low due to the high thermal resistance of 0.12 (m².K)/W between the pipes and the wall structure caused by an air gap that formed around the pipes arranged in the channels (Fig. 14a). The thermal resistance between the pipes and the wall structure could be reduced to 0.003 (m².K)/W by embedding the pipes in a thermally conductive material such as adhesive mortar (Fig. 14b), assuming a thermal conductivity of 1.0 W/(m.K) for reinforced concrete and 0.8 W/(m.K) for adhesive mortar.

Numerical simulations for the same wall with the pipes embedded in adhesive mortar, indicate a heating capacity of 24.9 W/m² and a heat loss of 16.6 W/m² at design ambient conditions (\(\theta_{\text{amb}} = -11^\circ\text{C}\)), and a heat transfer coefficient of 8 W/(m².K) between the wall and room and 25 W/(m².K) between the wall and the exterior.
Fig. 14 Heat flux and temperature distribution for a radiant wall with pipes embedded in a) air gap and b) adhesive mortar. Boundary conditions: $\theta_{amb} = 4.1^\circ C$, $\theta_i = 20^\circ C$, $\theta_w = 29.7^\circ C$, $h = 25$ W/(m².K)

5. Discussion

The accumulation capacity, which is expressed as a time constant, can vary from about 2.5 to almost 8 h for a system with pipes embedded in a massive concrete core [42]. The time constant of the wall system tested, which is calculated using the course of the heat flux as shown in Figs. 11 and 12, is slightly above 4 hours, which corresponds to a TABS.

5.1 Heating capacity and the effect of the material surrounding the pipes

At a supply water temperature ($\theta_w$) of 30°C, the heating capacity measured was only 5 W/m² (Fig. 11) during design and 8.5 W/m² during average winter conditions (Fig. 12). The difference in temperature between the supply and return water was small, and the surface temperature was only slightly above the room air temperature. The heating capacity was lower, despite the $h$ being higher than in other studies that involve wall heating systems with pipes embedded in a layer between the thermal insulation and the room. For example, the heating capacity was 18.7 W/m² at the supply water
temperature ($\theta_{sup}$) of 30°C, $\Delta \theta$ of 1.94K, and $h$ equal to 9.62 W/(m².K) as reported in [4], and it was 32.98 W/m² at $\theta_{sup}$ of 31.89°C, $\Delta \theta$ of 4.2K and $h$ equal to 8.16 W/(m².K) as reported in [35]. The overestimation of the heating capacity measured in this study due to the elevated $h$ as compared to the standard value of 8 W/(m².K) [13] is estimated to be within 10% of the value measured.

One reason for the low heating capacity obtained is shown in Fig. 14. Although ideal conditions regarding the heat transfer between pipes and a radiant surface are assumed in the design process, these may be difficult to achieve in practice. This is due to the pipes having been pressed into channels milled in the polystyrene thermal insulation [14]. When the pipes and insulation are attached to the concrete core in the form of precast panels, even a small discrepancy between the insulation’s surface plane and the pipes’ surface plane results in an imperfect contact between the pipes and radiant surface; consequently, the heating capacity is lower than expected. Moreover, an air gap may form around the pipes that hinders the heat transfer due to its low thermal conductivity, as shown in Fig. 14a.

It is therefore important that the pipes embedded in the channels are surrounded by a thermally conductive material such as adhesive mortar to distribute the heat more efficiently and thereby enhance the heating capacity (Fig. 14b). Failure to take this step can result in a heating capacity of only one third of the capacity for a system with pipes embedded in adhesive mortar. This value refers to conditions that correspond to an average ambient temperature ($\theta_{amb} = 4.2^\circ$C) in the winter and a heat transfer coefficient of 25 W/(m².K) on both wall surfaces. Numerical simulations indicate a reduction in heating capacity by 10.3 W/m², but the heat loss increases by 6.7 W/m² if the wall is exposed to a design winter temperature ($\theta_{amb} = -11^\circ$C), assuming a standard heat transfer coefficient between the radiant surface and the environment of 8 W/m² and an outdoor heat transfer coefficient of 25 W/(m².K).

5.2 The effect of the piping arrangement

The results in Fig. 4 show an important effect of the piping arrangement: the closer the pipes are located to the exterior, the less efficient the heat transfer is to the room. On the other hand, arranging the pipes closer to the interior improves the heating capacity by: a) increasing the heat transfer from the pipes to the radiant wall (Fig. 4), and b) raising the usable heat flux ($q_i$) to the heat loss ($q_e$) ratio
(Fig. 5). As compared to pipes embedded in thermal insulation, about 45% more heat is transferred to the wall when pipes are positioned in the inner plaster and 35% more heat is transferred when pipes are positioned directly in a concrete core (Fig. 4). Only 60% of the heat transferred to the wall is transferred to the interior for pipes embedded in thermal insulation as opposed to 75% for pipes embedded directly in a concrete core and 80% for pipes in the surface layer (Fig. 5). This means that all the other conditions being equal, moving the heating pipes from the thermal insulation to the concrete core causes an up to 50% overall increase in the heat flux to the interior. Moving the pipes further to a layer on the inner surface results in an additional 13% increase.

5.3 The effect of the supply water temperature

The thermal fields in Fig. 6 demonstrate that the operating mode (SH or TB) of a thermally active wall significantly affects the direction of the heat flux within a wall. The flux is directed towards the interior, and the system acts as SH when the supply water temperature is higher than the room air temperature (Fig. 6c); on the other hand, it acts as TB and diminishes the heat loss to almost zero when the supply water temperature equals the room air temperature (Fig. 6b). Numerical simulations have shown the potential of TB to reduce the heat loss when the heat loss dropped from 5.9 W/m² to 0.6 W/m² compared to a wall that does not contain any thermally active elements (Fig. 6a). This result is consistent with [22], where fixing the water temperature in TB at about 17°C led to heat loss variations within a small range of −1.104 to −0.651 W/m². The ability of the TB to decrease the loads also corresponds with the results presented in [23], where fixing the water temperature in a hot climate in TB at 23°C, i.e., close to the room temperature, diminished the heat transfer to only 9.1% and the maximum heat transfer to the space to about one third as compared to a conventional wall.

This positive effect of TB was also confirmed by the experiment (Fig. 13). The expected heat loss of a wall identical to the one used in the experiments but with no TB, and which is exposed to design ambient conditions (\(\theta_{\text{amb}} = -11^\circ\text{C}\)), is about 11 W/m² based on the numerical simulations. Thus, the heat flux of -5 W/m² measured during the experiment means a heat loss reduction of about 50% as compared to a wall without any thermally active element. The discrepancy between the actual heat loss and the zero heat loss expected for TB can be explained by the imperfect contact between the
pipes and radiant surface. Consequently, less heat was transferred from the pipes to the wall and the TB was performing below expectations.

Fig. 7a further emphasizes the substantial effect that the supply water temperature has on the heat flux to the interior. Under dynamically changing ambient winter conditions, the heat flux is 24 W/m² at a supply water temperature of 40°C, whereas it is slightly negative when an active wall system is operated as TB ($\theta_w = 20^\circ$C). Depending on the supply water and ambient temperatures, the heat flux for SH ranges between 2 and 24 W/m², i.e., raising by 0.7 to 1.2 W/m² per °C of the supply water temperature. These results refer to an $h$ equal to the standard value of 8 W/(m².K). The increase in heat flux would be more pronounced if a rise in $h$ by increasing $\Delta\theta$ as suggested by [43] would have been considered.

The heat flux to the exterior primarily depends on the ambient temperature, while the effect of the supply water temperature is limited (Fig. 7b).

### 5.4 The effect of the thickness of the wall and the spacing of the pipes

The heat flux to the interior ($q_i$) increases (Fig. 8a), while the heat loss ($q_e$) drops rapidly (Fig. 8b) for insulation in a range of 50 to 100 mm thick. Beyond 100 mm, the effect of adding more insulation becomes relatively small. This is true regardless of the thickness of the concrete core, which has almost no effect on the heat flux. On the other hand, the effect of the spacing of the pipes on the heating capacity is obvious (Fig. 9).

Despite its small effect on the heat flux to the interior, increasing the thickness of the concrete, and thereby the wall’s inertia, can reduce the energy demand for heating. For example, for the climate of Milan, Italy, the energy demand may be up to 10% lower for high inertia walls than for low inertia ones [44]. Calculations for the climate of Riyadh, Saudi Arabia, show a decrease in energy needs when increasing the thickness of walls and energy savings for heating up to 35% as a result of optimizing the thermal mass [45]. The thickness of the concrete is important for the thermal dynamics of buildings and has to be considered when designing a new building, reconstruction or a control system [46,47,48].

### 6. Conclusion
A thermally active wall with pipes arranged in milled channels in the thermal insulation was investigated. This system can be installed in both new and existing buildings in the form of precast insulation panels attached to their facades. The conclusions that may be drawn from this study are:

- By actively controlling the supply water temperature, it is possible to alternate a wall’s function between SH and TB, depending on the user’s preferences or the actual conditions.
- The wall system has the potential to significantly reduce heat loss when used as TB and thereby reduce the need for thick insulation compared to conventional insulation.
- The investigated system presents an alternative solution for SH that can be considered for installation in existing buildings as a part of retrofitting. In new buildings a more efficient system with heating pipes located in the concrete or the internal plaster is preferable.
- Pressing the pipes into channels milled in thermal insulation as suggested in the patent [14] can diminish the heat transfer from the pipes to the structure. To facilitate the heat transfer, the pipes arranged in the channels should be embedded in a thermally conductive material.
- The thickness of the insulation, the spacing of the pipes, and the supply water temperature have a substantial effect on the heating capacity, whereas the thickness of the concrete does not.

It is suggested that future research should investigate the possibility of (1) quantifying the feasibility of the proposed solution in a complex building system as compared to other insulation methods; (2) alternating the operation of the system between TB and SH, depending on the actual operating conditions, the configuration of the system, and availability of energy sources; (3) applying the insulation panels to walls with low heat accumulation capacity; and (4) embedding the pipes in insulation materials other than expanded polystyrene due to their fire resistance and their potential ecological and other benefits.

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