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Dynamic Simulation and Experimental Validation of Unsteady State Operation of Floor Heating Systems

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Abstract

Due to the rising auxiliary energy demand for the heat distribution in buildings alternative control strategies become necessary. The Unsteady State Floor Heating Project conducted at the Hermann-Rietschel-Institute at the Technical University of Berlin aimed at developing a control strategy to reduce the pump energy demand for heat distribution while maintaining comfortable thermal conditions and short response times of floor heating systems. The project idea was to transfer the heat unsteadily by operating the circulation pump in a pulsed manner. By increasing the supply temperature of the heating system the median mass flow rate could be reduced.

Two typical floor heating systems are modelled in Modelica - a conventional system with heating tubes installed in a counter flow spiral pattern and a state of the art capillary tube system installed in a parallel tube pattern with a reduced concrete layer thickness. Different supply temperatures and mass flow rates were investigated concerning their applicability for an unsteady operation.

Both systems were built up at the Institute's thermal test facility to generate experimental results for the validation of the simulation model.

With rising supply temperatures and decreasing mass flow rates the specific energy demand in both systems could be reduced by around 82 % at a supply temperature of 55 °C. The heat-up time to increase the operative temperature from 18 to 22 °C could be reduced by a factor of up to 4.6 using the capillary tube system and up to 2.6 using the conventional tube system.

Keywords – Floor Heating System, Unsteady State Operation, Dynamic Simulation, Experimental Validation, Capillary Tube

1. Introduction

The project with the short title "Unsteady State Operation of Surface Heating Systems" conducted at the Hermann-Rietschel-Institute at the Technical University of Berlin was initiated in response to the rising auxiliary energy demand for heat distribution in buildings. The project idea was based on an intermittent heat supply which has not been investigated thoroughly yet. Previous investigations did not cover the unsteady heat supply using on-off-controllers.

E.g. Glück [1] and Schnieders [2] conducted studies under unsteady outer conditions using steadily controlled heat transfer to vary supply temperatures and mass flow rates. By increasing the supply temperature and the temperature difference across the heating surface the hot water mass flow rate and thus the pump energy demand can be reduced. The room's heat up time after a cool down period could also be reduced.

This paper aims at presenting the results regarding achieved electrical energy savings, reduced heat up time and the validation of the simulation models.

2. Experimental Setup

A thin layer system based on capillary tubes installed in a parallel pattern shown in Figure 1 and a conventional system based on heat tubes installed in a counter flow spiral pattern given in Figure 2 are built up and investigated at the institute's thermal test facility. The thermal test facility consists of a climate chamber and a test chamber. The climate chamber can be cooled to around -9 °C to produce transmission and ventilation heat losses. Inside the test chamber two office working environments are set up in order to provide realistic internal heat sources (shown in Figure 3).

For an unsteady state operation of the surface heating systems applicable supply temperatures in the range of 40 to 55 °C and relative mass flow rates in the range of 0.55 to 1 (relative to the maximum mass flow rate of 330 kg/h) are identified. An electrically powered water heater with an internal supply temperature control and a constant pressure drop is used as a heat source. The heating water is conveyed by a frequency controlled pump. There is no return flow control installed. Thus the total heating systems' pressure drop is dependent on the mass flow rate only.

The heat supply is controlled using an on-off-controller maintaining an operative temperature of 22 ± 0.2 °C inside the test chamber. The operative room temperature is measured in head height (1.1 m) between the working environment and the outer wall adjacent to the climate chamber.

The floor temperature is measured with thermocouples arranged in a three times three matrix along the floor surface on top of the structured fleece carpet.

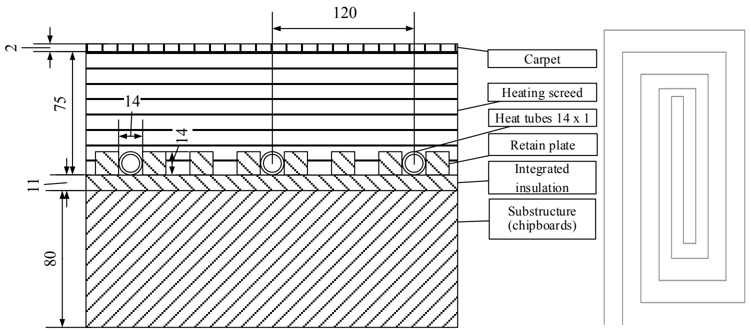

Fig. 1 Cross section of conventional surface heating system and laying pattern (not to scale)

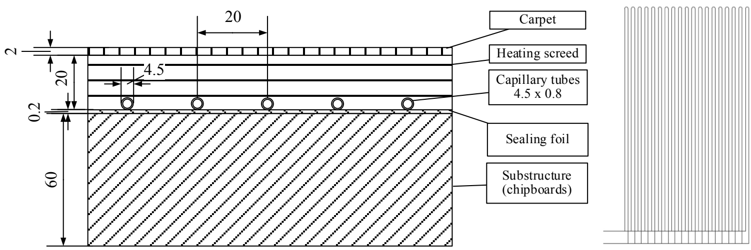

Fig. 2 Cross section of thin layer surface heating system and laying pattern (not to scale)

Fig. 3 Test chamber with two office working places and globe thermometers

3. Experimental Results

The experimental results are compared to the steady state operation of each system separately.

For the evaluation of the energy savings the specific energy demand e_{spec} is determined. It is the quotient of electrical E_{el} and thermal Q_{th} energy consumption as determined in (1) and allows the comparison among heating systems.

$$e_{spec} = E_{el} \cdot Q_{th}^{-1} \tag{1}$$

The reduction of the specific energy demand can be achieved for both examined systems. Increasing the supply temperature from 35 °C (steady state) to 50 °C (unsteady state) and decreasing the relative mass flow rate from 1 to 0.7 leads to a reduction of the specific energy demand of approx. 88 % for the thin layer system and approx. 86 % for the thick layer system (calculated values based on experimental measurements).

The room's heating energy demand remains the same through the application of the unsteady state pump operation, whereas the total exergetic efficiency ε_{total} of the heating system calculated according to (2) decreases by 26.6 % for an operation at a supply temperature of 50 °C compared to the operation at a supply temperature of 35 °C.

$$\varepsilon_{total} = (1 - T_{amb}/T_{op}) \cdot (1 - T_{amb}/T_{supply})^{-1} \tag{2}$$

Wherein T_{amb} denotes the ambient temperature of 273 K (0 °C), T_{op} the operative temperature of 295 K (22 °C) and T_{supply} the heating water supply temperature (varying between 308 and 323 K respectively 35 to 50 °C).

For the evaluation of heat up time reduction the time to increase the operative temperature from 18 to 22 °C is measured. The temperature inside the test chamber was maintained at a value of 16 °C for two hours before the measurements were started to provide constant initial conditions. As expected, the response time of the thin layer system is lower than the response time of the thick layer system as depicted in Fig. 4. An increase in the supply temperature from 35 to 55 °C leads to a reduction of the room's heat up time by a factor of 4.6 for the thin layer system and by a factor of 2.6 for the conventional system.

The floor temperature temporarily exceeds 29 °C (maximum permitted temperature according to DIN EN 1264 [4]) using the thin layer system. The median floor temperature does not exceed the temperature limit – neither during the thin layer nor the conventional system operation.

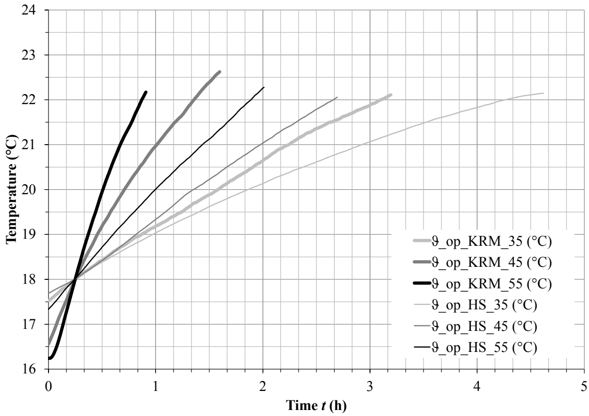

Fig. 4 Trend of operative temperature after cool down period at different supply temperatures for the thin layer system (KRM) and the conventional sytem (HS)

4. Simulation Models

In order to simulate the unsteady operation of a floor heating system and evaluate its effect on the dynamic thermal behavior of the room a Modelica model was created. It consists of a room model and a floor heating system based on the model presented by Gilani in 2014 [3].

The room model represents the test chamber of the experimental setup. It consists of heat transfer components for the enclosing surfaces. The ceiling, walls, windows and the door have been modeled to calculate the heat conduction and heat storage for different material layers. In addition there are components included to calculate the convective heat transfer on both sides of the surfaces. The inner sides of the surfaces possess two connections, one connects the surface with the room air and the other determines the surface temperature for the radiation exchange component. In the simulated room there are six surfaces with different temperatures in radiative exchange with each other. The radiative exchange is of special importance in this study, since a considerable part of the heat transfer from the heat floor to the room is being carried out by radiation. In the model component for radiation exchange it has been assumed that the surfaces are black bodies with an emission coefficient of ε = 0.95. The radiation exchange rates are calculated as given in (3).

$$\dot{Q}_r = 5.67 \cdot 10^{-8}\, F_{ij}\epsilon_1\epsilon_2 A(T_1^4 - T_2^4) \qquad (3)$$

Where F_{ij} is the shape factor between the two surfaces which is calculated according to Çengel et al. [5]. Moreover, a model component was developed for the calculation of the mean radiant temperature of the room [6] discerning between seated (4) and standing persons (5):

$$\bar{T}_{\mathrm{r}} = \frac{1}{1.4} \cdot \begin{bmatrix} 0.18 \cdot \left(T_{\mathrm{pr}}[top] + T_{\mathrm{pr}}[down]\right) + 0.22\left(T_{\mathrm{pr}}[right] + T_{\mathrm{pr}}[left]\right) \\ +0.30\left(T_{\mathrm{pr}}[front] + T_{\mathrm{pr}}[back]\right) \end{bmatrix} \quad (4)$$

$$\bar{T}_{\mathrm{r}} = \frac{1}{1.28} \cdot \begin{bmatrix} 0.08 \cdot \left(T_{\mathrm{pr}}[top] + T_{\mathrm{pr}}[down]\right) + 0.23 \cdot \left(T_{\mathrm{pr}}[right] + T_{\mathrm{pr}}[left]\right) \\ +0.35 \cdot \left(T_{\mathrm{pr}}[front] + T_{\mathrm{pr}}[back]\right) \end{bmatrix} \quad (5)$$

Wherein denoting:

$\bar{T}_{\mathrm{r}}$ - Mean radiative temperature

T_{pr} - Surface radiation temperature

The room air in this model has been considered as a heat storage, which represents the heat storage of the air as well as the furniture, computers and the heating dolls. The operative temperature is calculated as the average of the air temperature and the mean radiant temperature.

The floor model was developed separately and connected to the room. The heat transfer process in floor body comprises of thermal conduction and heat storage between at least three temperature levels, which are the temperatures of the upper and the lower surface and the screed portion directly in contact with the pipes. For the development of the radiant slab model, modifications to the component *SingleCircuitSlab* of the Modelica Buildings Library were applied to match the component with the characteristics of the installed floor heating systems. The model is analogue to a network of thermal resistors and capacitors as shown in Figure 5. The temperature of the floor body that contains the pipes is calculated by a fictitious resistance for an arbitrary number of segments along the pipe. The model for the convective heat transfer in a straight pipe with circular cross section calculates a heat transfer coefficient as a function of the mass flow rate. For this purpose a distinction is made between:

a) A uniform wall temperature (UWT) or a uniform heat flux (UHF)

b) A full developed (DFF) or an undeveloped hydrodynamic flow (UFF)

c) Neglecting or considering the pressure loss.

The maximum Reynolds number for the laminar flow in this model is 2.200. Reynolds numbers over 10,000 indicate a completely turbulent flow.

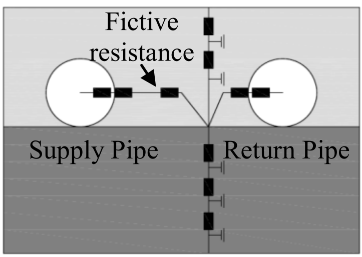

Fig. 5 Schematic of the floor model with resistors and capacitors

Both construction variants where simulated with a similar model structure. In both cases the fundamental model component consists of a supply pipe and its neighboring return pipe as shown in Figure 5. The model is able to discretize the pipe and floor in arbitrary number of segments. The number and shape of the segmentations as well as the geometric and thermal parameters distinguish the two construction models. Figure 6 displays the segmentation of the floor in the studied cases.

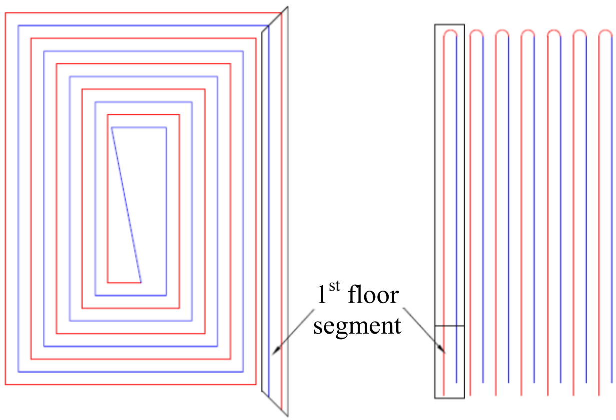

Fig. 6 Schematic of heating tube segmentation for conventional system (left) and thin layer system (right)

5. Validation of Simulation Models

The simulations models are validated with experimental data from the measurements of operative room temperature and the median floor surface temperature. For the validation all experimental measurements under steady state outer and inner conditions are included. In Figure 7 the temperature trends for the thin layer system are shown and in Figure 8 for the conventional system respectively (for both systems during the operation at a supply temperature of 50 °C and a relative mass flow rate of 1).

For the thin layer system the root mean square deviation (RMS) between simulated and measured operative temperature ranges between 0.11 and 0.39 K (with a mean value of 0.26 K), the RMS between simulated and measured mean floor surface temperature ranges between 0.46 and 0.92 K (with a mean value of 0.66 K). For the conventional system the RMS between simulated and measured operative temperature ranges between 0.56 and 0.95 K (with a mean value of 0.81 K), the RMS of between simulated and measured mean floor surface temperature ranges between 0.37 and 097 K (with a mean value of 0.66 K).

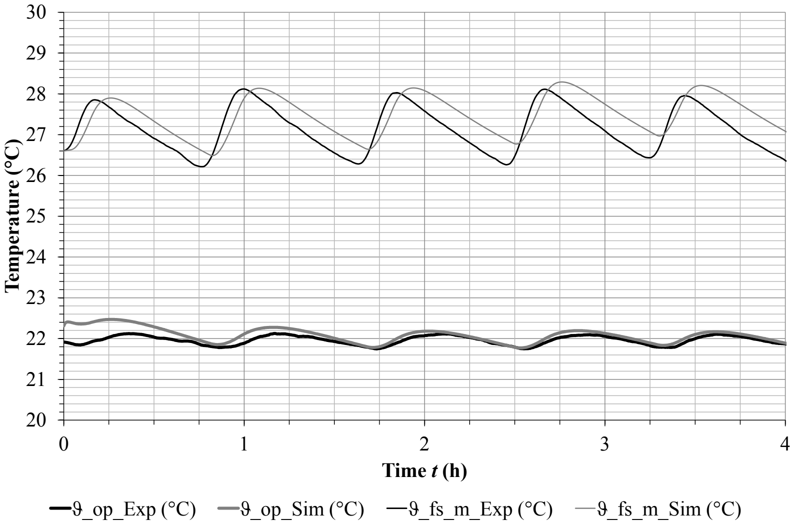

Fig. 7 Trend of operative (op) and median floor surface (fs) temperature at a supply temperature of 50 °C and a relative mass flow rate of 1 for the thin layer system during experiment (Exp) and simulation (Sim)

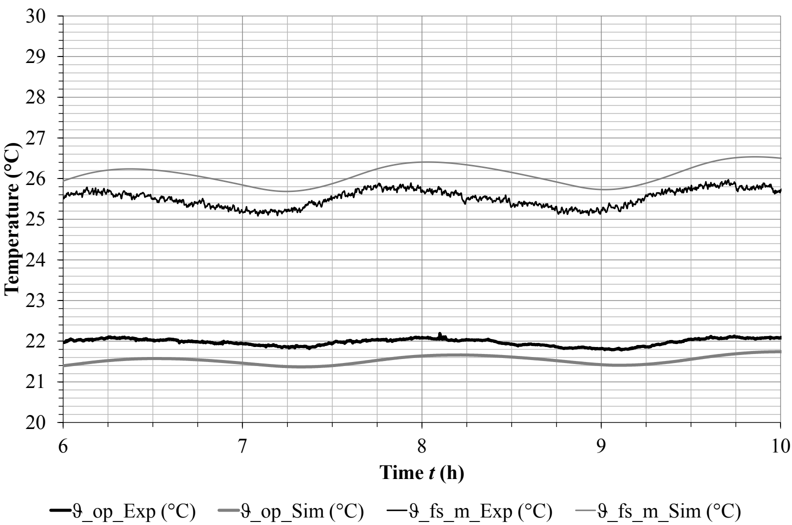

Fig. 8 Trend of operative and median floor surface temperature at a supply temperature of 50 °C and a relative mass flow rate of 1 for the conventional system during experiment (Exp) and simulation (Sim)

6. Discussion

The uncertainty of all calculated values is determined using the maximum error propagation method. The uncertainty of the specific energy demand is about ± 1.6 %. The median floor surface temperature's uncertainty is approx. ± 0.2 % referred to a reference temperature of 27 °C. The uncertainty of the measured operative temperature can be neglected.The thin layer system model calculates accurate results regarding the maximal and minimal mean floor surface temperatures. There is only a small time shift between the simulated and measured temperature maxima and minima. The time shift is a result of the parametric optimization to adapt the overall model behavior to the observed real system's behavior. Due to the model's higher heat capacity and lower heat conduction the predicted surface temperature evolves slower than the real system's surface temperature. Nevertheless, the model can be used for parametric studies without further revision regarding a supply temperature between 40 and 55 °C.

The conventional system model tends to slightly underestimate the operative room temperature and to overestimate the floor surface temperature in comparison to the empirical data. The deviation is a result of the uncertainty concerning the floor structure material's properties and the simplified discretization of the conventional heat system's layout.

The limited controllability of the temperature inside the climate chamber due to the large time hysteresis of the compressor obstructed the model validation under dynamic outer conditions. This does not have negative effects on the validity of the validation mentioned before. For the implementation of different controller types the dynamic trend of the temperature inside the climate chamber has to be used as an input for the simulation.

7. Conclusion and Outlook

The experimental results show promising applicability in systems demanding short response times or systems providing supply temperatures higher than those usually applied for floor heating systems, e.g. during renovation of old buildings. If the existing or planned heating generator provides supply temperatures above 35 °C, the auxiliary energy demand of the distribution pump can be significantly reduced. The replacement of return flow addition to control the supply temperature is generally recommended in combination with the unsteady state operation. However, the maximum floor temperature has to be limited. Extending the investigation to the impact of the floor construction and the variation of the control strategy to achieve this goal will be part of the further investigations.

The created Modelica models can be easily adapted to different constructive and operational requirements regarding mass flow and supply temperature set points and hence provide a powerful tool for broader parameter studies. Nevertheless studies with alternative controller types require an extended validation under dynamic environmental and internal conditions.

Acknowledgement

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