TRANSIENT MODEL OF A PANEL RADIATOR

Christian Brembilla, Mohsen Soleimani-Mohseni, Thomas Olofsson
Department of Applied Physics and Electronics, Umeå University, Umeå, Sweden

ABSTRACT

This paper shows a transient model of a hydronic panel radiator modelled as a system of multiple storage elements. The experiment´s results suggest the more suitable technique for modelling this technology. The panel radiator is modelled numerically with eight thermal capacitance connected in series by keeping a memory of the heat injected in the thermal unit. The comparison of the performance among lumped steady state models and transient model, in terms of heat emission and temperature of exhaust flow, shows the potential of the latter approach. To conclude, (1) the transient phase is essential for modelling stocky panels, and (2) this type of modelling has to be addressed for evaluating the performance of low energy buildings.

INTRODUCTION

General guidelines establish new challenges for the heating system (EPBD 2010). Heating needs obviously to decrease, thus an accurate model for predicting the heat emission of hydronic panel radiators is needed to fulfil the norm requirements. A question arises: what is the most suitable method for modelling the panel radiator to estimate its heat emission accurately? The answer to this question is not simple, because the detail level of modelling depends on the available information and the expected outcomes. Moreover, hydronic panel radiators vary in size and location of pipe connections. Firstly, the geometry of panel radiators, the ratio length/high (L/H) affects the charging time of the thermal unit. For instance, stocky panel radiators, with L/H>2, take more time for charging in comparison with panel radiator with L/H<2. Secondly, the location of supply and exhaust pipes can be positioned at the same or opposite side of the panel affecting the charging direction. For example, supply and exhaust lines on the same side make the charging process from left to right or vice-versa, whereas, from top toward bottom when the pipes are on opposite side of the panel. Lastly, the amount of mass flow rate ($\dot{m}_w$) injected into the panel increases the charging time of the thermal unit typically when ($\dot{m}_w$<0.02 $\dot{m}_{w,N}$).

These three details, geometry, pipes connection location and amount of mass flow rate are not considered by the norms (EN 15316-1 2007, EN 15316-2-3 2007) for calculating the *efficiency* for emission of radiators. The norms allow calculating the heat losses and efficiencies with dynamic simulation methods, taking into account the time history of variable values. These variables are the boundary values of the model such as the outside temperature. However, the norms do not specify the type of model needed for calculating the *efficiencies* The *efficiencies* of hydronic heating system are affected by the room temperature behaviour. The room temperature typically oscillates during period of low heat demand. A period of low heat demand occurs when the outside temperature is between 8–5°C and the free heat gains such as sun, occupancy, lighting significantly affect the room temperature trend (Tahersima et al 2013). A lumped steady state model of a panel radiator does not cope with such situation, therefore, it is not suitable to estimate the *efficiencies* of the hydronic system. Instead, a *transient* model predicts the time of heating/cooling of the thermal unit and consequently it defines the heat emitted during the transient phase by affecting the indoor temperature. Moreover, the *transient* model enables to encompass details such as location of pipes connections, amount of mass flow rate and the geometry.

This paper presents a detailed *transient model* of a panel radiator according to (Stephan) and (Holst 1996). The experiment with the thermal imaging shows the charging direction of this technology. Lastly, the comparison among lumped steady state models, and the transient model in terms of heat emission and temperature of exhaust flow emphasizes the advantages of the latter approach. All these types of modelling should be discussed among the research community to introduce approaches that guarantee an adequate estimate of hydronic systems performance.

Review of Panel Radiator Models

Panel radiators are water to air heat exchangers, designed to satisfy the requirements of heating demand typically for rooms. The radiator heats up the surrounding environment by following the mechanism of convection and radiation.

This mechanism is widely described by several authors. For instance, (Maivel et al. 2014) positions series radiators in a thermostatic room, which keeps the wall surfaces at the same temperature. The thermostatic room allows simplifying the heat exchanged by radiation among surfaces and it lets to develop and validate a lumped steady state model against the experimental results. In reality, the room is subjected to thermal gains from free sources by provoking oscillations of the room temperature. The lumped steady state model cannot deal effectively with such variations.

Other authors et al., present ventilation radiators for existing buildings (Myhren et al. 2009). The outside air passing through the channel made by a coupled panel radiators. This air works as ventilation for the indoor environment. The radiator model is a lumped steady state model working in static conditions.

(Maivel et al. 2015), in his second paper, estimates the heat losses coming from a heating system composed by tank, pipes, thermostatic valve and panel radiator. In this case, the radiator is modelled as lumped steady state model employed dynamic conditions. This model does not estimate the heat emission during the transient phase, consequently the model is not useful when room temperature oscillates to estimate the *efficicency for emission*.

(Tahersima et al. 2010) introduces a lumped steady state model of a panel radiator based on the energy balance between the heat injected and the emissivity towards the indoor environment. The lumped model divides the thermal unit in N elements, which exchange heat into the room only by convection neglecting the radiation. (Janćik et al. 2008) shows a transient model of a panel radiator resolved with a fourth order transfer function by using the technique of unknown system identification. Measured data of step response experiment validates the model developed in Matlab/Simulink programming. (Janćik et al. 2012) introduces a transient model of a panel radiator in which he implements the heat transfer by radiation and convection towards the room. (Tahersima et al. 2013) performs a horizontal transient modelling of the panel radiator divided in horizontal elements with the supply and exhaust connection positioned on the opposite sides of the panel. Tahersima resolves this problem with elegant analytical solution.

In all above-mentioned works, only Tahersima and Janćik together with Stephan and Holst (mentioned in the introduction section of this paper) have developed forefront models that are able to deal with dynamic boundary conditions. Those models are also suitable for controlling the thermal technology, by applying the TRV for adjusting the mass flow rate.

EXPERIMENT

The aim of the test is to investigate the *heating* of the panel radiator during the charging process. The name of the test is *step response* of the panel radiator. Qualitative measurements are the expected outcomes since the panel radiator is located in a room that does not follow the requirements listed in the EN 442 (EN 442-2 et al. 1996). A balancing valve, positioned on the return pipe of the system adjusts the pressure during the test. The thermal image takes note of the temperature patterns of the radiator surface. Fig 1 shows the panel radiator Lenhovda MP 25 500 used in the experiment. The panel radiator is located at the University laboratory.

The heating system is composed of four panel radiators, and for this experiment, the return valves of the others three thermal units are closed, thus the fluid cannot circulate in all system. The distribution system is composed of non-insulated copper pipes. Balancing valves (red spots in Figure 1) are positioned on return pipes of the system. The balancing valve on the lower part of Figure 1 is closed. Whereas, the valve in the middle is fully open and the last valve adjusts the mass flow rate according to *kv* coefficient. The manometer, positioned at the beginning of the supply line detects the flow pressure, thus the supply mass flow rate is calculated. The latter data is needed to set-up the k_v coefficient of the balancing valve for controlling the mass flow rate. The domestic hot water of the building injects the supply flow into the panel radiator. The water runs for 10min before reaching a constant temperature of 55°C. The supply line is then connected into the tap with a pressure of 5000Pa. The panel radiator has the supply and exhaust connection on the same side of the unit. The supply line is in the top right corner and the return connection is in the bottom right corner.

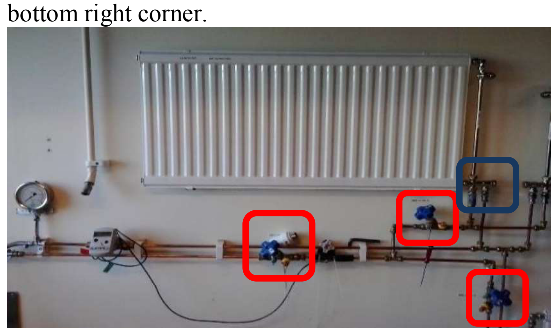


Figure 1: Panel radiator Lenhovda MP 25 500

The blue spot (in Figure 1) is the meeting point of supply and exhaust flows. The test was performed during a weekend of November 2014, when the building is empty to avoid possible pressure and temperature oscillations of the flow supplied.

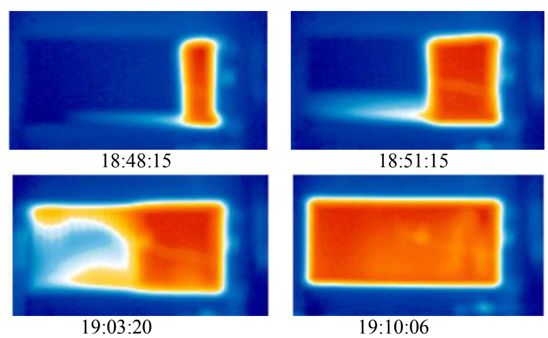


Figure 2: Heating up sequence

Figure 2 shows the process of heating up with the temperature field pattern of the radiator surface. At 18:44:00, the panel radiator is at the temperature of 20°C. The experiment starts at 18:45:00 and the radiator begins the charging phase. The thermal imaging clearly shows that the process of heating up is from right to left side with this type of connections. However, already at 18:48:15, it is visible a hot area in the lower part of the unit. This means that, a fraction of flow recirculates inside the panel. In the lower part of the second image at 18:51:15, the hot area is larger than before. The radiator is loaded backwards, and now, the process of heating up is from bottom toward top. This is because some residual air inside of the panel radiator does not allow a normal charging of the unit. By opening the vent valve, the charging process becomes from top toward bottom as shown in 19:03:20. At 19:10:06, the charging process is over. To conclude, this qualitative experiment clearly shows that the normal charging process of the panel radiator is from right toward left.

NUMERICAL MODEL OF PANEL RADIATOR

This section introduces the numerical model of the panel radiator. The first sub-section presents the model assumptions and the physical description of the thermal unit. The second sub-section describes the model constrains related to the amount of mass flow rate injected. The last subsection shows the model implementation.

Here, we first introduce the definition of transient, steady state model and dynamic method. A *transient model* occurs when the heat is stored in the system thermal capacitances (Mayer et al. 1998). This means that, the model keeps a memory of the heat injected and it takes time before the thermal unit is fully charged. Both temperature of fluid flow injected and heat emitted towards the indoor environment are time dependent and evolve during the charging/discharging process. Conversely, a steady state model performs the analysis without keeping a memory of the heat stored by showing, in solids, a linear temperature behaviour. Dynamic methods typically use steady state models by changing at each time step the value of input parameters or of boundary conditions. Dynamic methods are frequently used by commercial software for building simulation.

The author presents a panel radiator modelled according to the definition of transient system. Thus, the thermal unit is modelled according to the heat balance among the heat injected into the panel, the heat emitted towards the indoor environment and the heat stored into the thermal unit.

Assumptions and panel physical description

The model assumptions are the following:

- no hydraulic process of the fluid flow and relative hydraulic resistance of the panel are considered in the process,
- the panel radiator is divided into five equal elements connected in series, this means that, the temperature of the supply flow of the following element is the temperature of the exhaust flow of the previous one,
- the radiator thermal mass, sum of the water and metal mass, is concentrated at the exhaust point of each radiator element/capacitance,
- the surface temperature of panel radiator is equal to the flow temperature for each element,
- air temperature is assumed constant at 20ºC during the test,
- no additional time delay is considered in the simulation,
- the radiator can only heats the room, the cooling process is not considered.

Equation 1 shows the heat balance of the panel radiator.

$$\dot{Q}_{injected} = \dot{Q}_{stored} + \dot{Q}_{emitted} \qquad (1)$$

Equation 2 describes mathematically each term, formulating a first order partial non-linear differential equations with autonomous parameters. The PDE is evaluated in time and in the panel length, L, named as x direction. The number of differential equations is according to the number of thermal capacitances considered, in this case the capacitance are five.

$$\dot{m}_w \cdot c_w \cdot \frac{\partial T(x,\vartheta)}{\partial x} = C_{rad} \cdot \frac{\partial T(x,\vartheta)}{\partial \vartheta} + Q_N \left(\frac{\Delta T_{log}}{\Delta T_N}\right)^n \qquad (2)$$

C_{rad} is the total radiator capacitance calculated as sum of metal plus water capacitance. Equation 3 shows how to calculate the total radiator capacitance.

$$C_{rad} = M_w \cdot c_w + M_{met} \cdot c_{met} \qquad (3)$$

The water specific heat capacity depends on the temperature of the fluid in each element. This parameter is calculated according to Equation 4.

$$c_w = A_0 + A_1 \cdot T_{su} + A_2 \cdot T_{su}^2 + A_3 \cdot T_{su}^3 + A_4 \cdot T_{su}^4 + A_5 \cdot T_{su}^5 \qquad (4)$$

The coefficients $A_0, \ldots, A_5$ are constants listed in Table 1. The third term of Equation 2 is the logarithmic temperature difference between the temperature of supply flow, exhaust flow and indoor air.

$$\Delta T_{log} = \frac{T_{su} - T_{ex}}{\log \frac{T_{su} - T_{exh}}{T_{ex} - T_{air}}} \qquad (5)$$

ΔT_N is the logarithmic temperature difference at nominal conditions. This coefficient is usually found on technical catalog. Table 1 lists all parameters adopted in the transient model. The technical specifications are read from the catalog of the panel Lenhovda MP 25 500 (Lenhovda et al. 2014).

Table 1 Panel radiator data

Geometry		Metal specific capacity (c_{met})	897 $JKg^{-1}K^{-1}$
Height (H)	0.5 m	**Nominal conditions**	
Length (L)	1 m	Radiator exponent (n)	1.286
Metal mass per channel	0.41 kg/channel	Supply temperature ($T_{su,n}$)	55 °C
Water mass per channel	0.1243 l/channel	Exhaust temperature ($T_{exh,n}$)	45 °C
n° of channel	26	Air temperature ($T_{air,n}$)	20 °C
Total metal mass (M_{met})	10.71 kg	Nom. mass flow rate ($\dot{m}_{w,n}$)	0.0064 kgs^{-1}
Total water mass (M_w)	3.23 l	Nom. power ($\dot{Q}_n$)ΔT_n=30 °C	276 W
Adimensional coefficients of water specific capacity			
A_0	4.2169	A_3	$1.5590 \cdot 10^{-6}$
A_1	$3.2826 \cdot 10^{-3}$	A_4	$1.2034 \cdot 10^{-8}$
A_2	$1.0368 \cdot 10^{-4}$	A_5	$3.5411 \cdot 10^{-11}$

Figure 3 shows the modeling of the panel radiator divided into five equal elements or thermal capacitances. These elements are connected in series, this technique is also known as system with multiple storage elements (Siemens et al. 2015). Moreover, Figure 3 shows the terms of Equation 1, the power injected and emitted towards the indoor environment. The capacity to store thermal heat does not appear in the picture since it is hidden in each element.

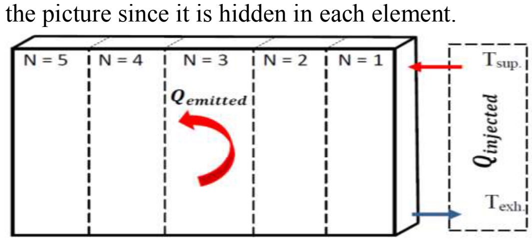


Figure 3: Power injected, emitted and stored

Mass flow rate

The panel radiator emits heat according to the amount of mass flow rate injected. Low mass flow rate, typically ($m_w \leq 0.02\ m_{w,N}$), can make the temperature of the exhaust flow greater or equal than the room temperature. When the temperature of exhaust flow is greater than the room temperature, a correction to ΔT_{log} needs to be introduced as shown in Equation 6.

$$\Delta T^*_{log} = \frac{\dot{m}_w}{\dot{m}_{w,n}} \cdot \Delta T_{log} + \left(1 - \frac{\dot{m}_w}{\dot{m}_{w,n}}\right) \cdot (T_{exh} - T_{air}) \quad (6)$$

The nominal mass flow rate is calculated according to the nominal conditions listed in Table 1 with Equation 7 (Stephan).

$$\dot{m}_{w,n} = \frac{Q_N}{c_w \cdot (T_{sup,N} - T_{exh,N})} \quad (7)$$

In case of the temperature of exhaust flow is equal to the temperature of indoor environment, the power emitted from the radiator is assumed equal to the power injected as described in Equation 8.

$$\dot{Q}_{emitted} = \dot{m}_w \cdot c_w \cdot (T_{sup} - T_{exh}) \quad (8)$$

On the other hand, when the mass flow rate is high, the temperature of supply flow could be equal to the temperature of exhaust flow. In this case, the logarithmic temperature difference in Equation 5 cannot be used if the following relation is fulfilled:

$$0.999 < \frac{T_{sup} - T_{air}}{T_{exh} - T_{air}} < 1.001 \quad (9)$$

The arithmetic temperature difference replaces the previous relation as read in Equation (10).

$$\Delta T = \frac{T_{su} + T_{ex}}{2} - T_{air} \quad (10)$$

Model implementation

Newthon-Rahpson (N-R) method solves the system of discretized equations. N-R is a second order method (quadratic) which finds successively approximations of a real-value function. The method starts with the guessing of the first value of T_{exh}. The new value of $T_{exh,new}$ is found by applying the N-R algorithm described in Equation 11.

$$T_{ex,new} = T_{ex,old} - \frac{F(T_{ex,old})}{F'(T_{ex,old})} \quad (11)$$

where:

$$F(T_{exh,old}) = \dot{m}_w \cdot c_w \cdot (T_{su} - T_{exh}) + C_{rad} \cdot \frac{T_{ex,new} - T_{ex,old}}{\Delta\vartheta} + \frac{Q_N}{N}\left(\frac{\Delta T_{log}}{\Delta T_n}\right)^n \quad (12)$$

$$F'(T_{ex,old}) = -\dot{m}_w \cdot c_w - \frac{C_{rad}}{d\vartheta} - \frac{Q_N}{N} \cdot \Delta T_{log,N}{}^{-n} \cdot n \cdot (\Delta T_{log})^{n-1} \frac{\left(\left(\frac{T_{su} - T_{ex}}{T_{exh} - T_{air}}\right) - \log A\right)}{(\log A)^2} \quad (13)$$

$$A = \frac{T_{su} - T_{air}}{T_{exh} - T_{air}} \quad (14)$$

The model is implemented by using Matlab programming.

RESULTS

This section presents the results of the numerical model. The first sub-section shows the main model outcomes: the temperature of exhaust flow, the heat stored and emitted by each capacitance. The second sub-section explains the time step adopted in the simulation and the last one the heat transfer towards the indoor environment. The simulation runs for 80 minutes, the time step is 5 seconds, the mass flow rate is 0.01 kgs^{-1} and the temperature of supply flow is 55 °C.

Temperature exhaust flow, heat stored and emitted

The temperature of exhaust flow is calculated for each thermal capacitance by emphasizing the temperature gradient on the panel surface. Figure 4a shows the temperature of exhaust flow at the step of supply flow. Each blue dash line represents the exhaust temperature of each thermal capacitance. These lines become parallel after 50 minutes of simulation. This means that, the steady state condition is reached. The temperature gradient, identified between the supply temperature (red line in Figure 4a) and the temperature of the last thermal capacitance (N=5), is about 6°C. The exhaust temperature shows a inflexion point and a dead time for the capacitances N=2,3,4 and 5.

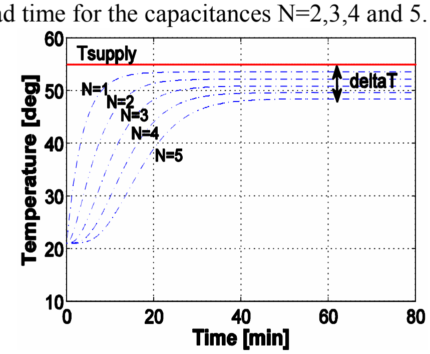
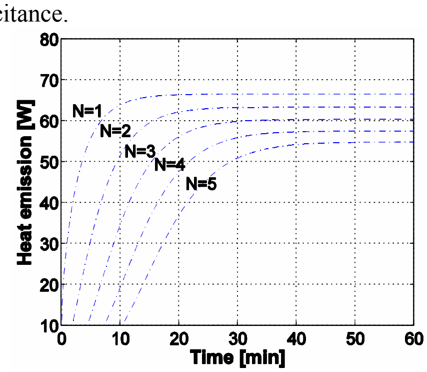


Figure 4: (a) Exhaust temperature of each thermal capacitance

Figure 4b shows the heat emission of each thermal capacitance.



(b)

Figure 4:(b)Heat emission of each thermal capacitance

Figure 5a and 5b show the heat injected and the heat stored into the panel radiator for each thermal capacitance. For the capacitance (N=1), the heat injected and stored is high due to the large difference of temperature between supply and exhaust. The thermal response of capacitances N=2,..,5 is delayed in time, since they are charged later in the process of heating up. Figure 5a and 5b are apparently the same unless the steady state condition is reached. After 50 minutes of simulation, the heat stored goes to zero.

This means that, all heat injected is equal to the heat emitted as it clearly showed in Figure 6.

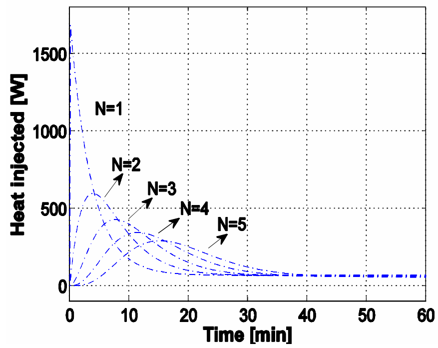


Figure 5: (a) Heat injected into each capacitance

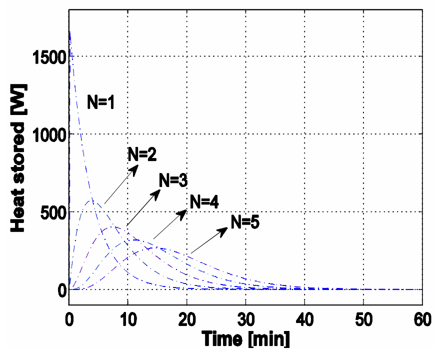


Figure 5: (b) Heat stored of each capacitance

A small delay between the heat injected and stored is not appreciable at the plot scale of the figure. The heat stored seams to appear at the same time of the heat injected. The modeling does not consider the heat conducted and stored in the cross section of the panel. On the other hand, the panel radiator is made by aluminum (Table 1, c_{met}), thus the heat is conducted really fast towards the indoor environment, by reducing the time delay between heat injected and stored.

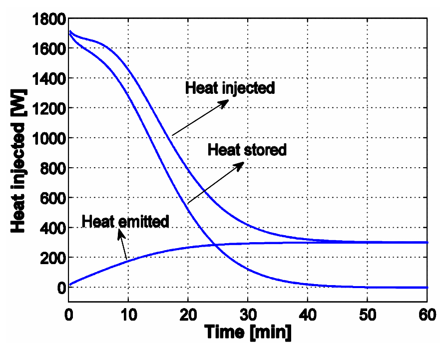


Figure 6: Heat emitted, injected and stored of the whole technology

Figure 7 shows the temperature distribution in the panel against the *x* variable (towards the panel length). *x* is discretized by subdividing the panel with chunks of the same length according to capacitances number. Here, the thermal unit is divided into 8 capacitance, hence *x=L/8*. It is possible to notice that, when the charging process starts, only the capacitances close to the supply line begin the heat process.

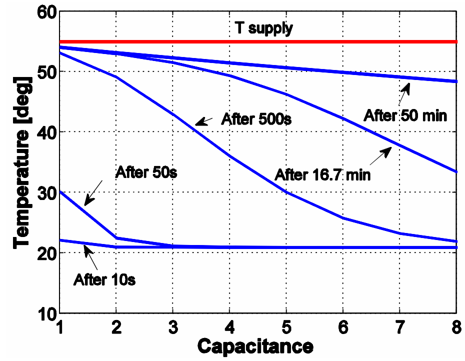


Figure 7: Spatial distribution of temperature

After 50 minutes the charging process is completely over. It is possible to appreciate a ΔT between the temperature supply and the latter blue dash line.

Time step

A short overview on the existing types of modelling explains how to choose the simulation time step. A thermal unit can be modelled without the capacity to store heat. For a system without storage elements, the output variable abruptly increases or decreases at each input value. Instead, systems with the capacity to store heat, know with the name of *transient models*, can be modelled with (i) one or (ii) multiple storage elements. When the unit is modelled with one capacity (i), the output variable increases gradually from the time x-axis until when the steady state condition is reached. The output variable trend, depends generally on the system thermal characteristics and on the input magnitude. The intersection point of the tangent curves at the beginning and ending of the output variable identifies on the time x-axis the thermal time constant of the system. Mathematically, and in thermodynamic terms, the thermal time constant is the ratio between the system thermal capacitance and the heat lost/transmitted in the unit of time. It is also identified numerically with the time taken from the systems to reach the 63.2% of its final asymptotic value (for increasing value of the input step).

On the other hand, (ii) for systems with multiple storage elements, the output variable slowly rises from the time x-axes, then it climbs with an increasing slope to the inflexion point and after that the rate of rise decreases again, by reaching the steady state condition. In this type of systems, the transient behavior is characterized by the delay time $T_{d(i)}$ and the balancing time $T_{b(i)}$. To determinate these times a tangent curve is plotted at the inflexion point of each thermal capacitance (Siemens et al. 2015). Figure 8 shows graphically the times ranked in Table 2. The balancing and dead time depend primarily on the amount of mass flow rate injected into the panel. This means that, the increase in the mass flow rate reduces the dead and balancing times or, summarizing, the transient phase. For all these reasons, the time step adopted in the simulation must be less of each dead or balancing time at high flow rate condition. Therefore, a time step of 5s is chosen for this simulation.

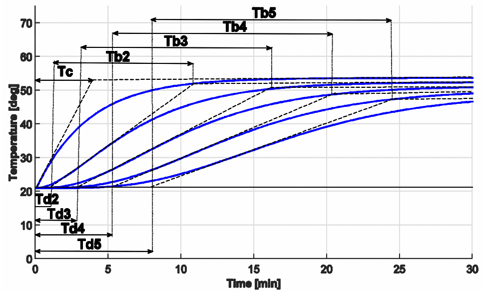


Figure 8: Temperature of exhaust flow of each thermal capacitance

Table 2 Dead, balancing time, time constant at $\dot{m}_{w,n}$

Balancing time		Dead time	
Tb_2	9 min 30sec	Td_2	1.5 min
Tb_3	13 min 30sec	Td_3	3 min
Tb_4	15 min	Td_4	5 min 20 sec
Tb_5	16 min 30sec	Td_5	8 min

Heat transfer towards indoor environment

The heat transfer from the panel radiator towards the indoor environment is mostly by convection. However, the panel also releases radiative heat together with the convective heat. Here, the radiative heat is calculated for the long wave radiation in Equation 15.

$$\dot{Q}_{radiative} = S_{rad.} \cdot \varepsilon \cdot \sigma \cdot \left(\overline{T_m}^4 - T_{wall}^4\right) \qquad (15)$$

Equation 15 assumes the wall temperature fixed at 20°C and the variable T_m is the mean radiator temperature of the panel calculated as the average temperature between the temperature supply and exhaust. Secondly, the radiative heat is accounted as a fraction of the total heat emitted as shown in Equation 16.

$$s_{rad,N} = \dot{Q}_{rad}/\dot{Q}_N \qquad (16)$$

Figure 9 shows the heat emitted into the room as function of the mean radiator temperature. The blue line is the total heat emitted and the red line is the radiative heat. For this type of panel radiator only the front surface is considered for the heat exchanged by radiation. The percentage of heat transmitted by radiation varies between about 27-35% of the total heat.

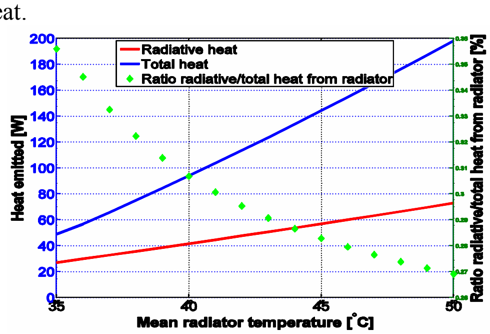


Figure 9: Total and radiative heat emitted

COMPARISON OF PERFORMACE AMONG MODELS

The first sub-section explains briefly the lumped steady state model for *water radiator* used in IDA ICE software. The second sub-section compares the performance among *transient* model, its *lumped steady state model* and *IDA ICE* model in terms of i) temperature of exhaust flow and ii) total heat emission.

IDA ICE model

IDA ICE presents, in the library, a lumped steady state model of a *water radiator*. The method applied for simulating the hydronic unit is known with the name of *dynamic method*. A *dynamic method* occurs when, the boundary values ($\dot{m}_w, T_{liq}$) change independently at each time step of the simulation. Equation 17 describes the model of *water radiator*, it is a first order non-linear ordinary differential equation (ODE) with autonomous parameters. The equation represents the heat balance between the heat injected and emitted.

$$\dot{m}_w \cdot c_w \cdot \frac{dT_{liq}(x,\vartheta)}{dx} = k \cdot (T_{liq} - T_{air})^n \qquad (17)$$

The flow temperature (T_{liq}) is only function of the variable x accounted as the panel length (L). The model does not considers the heat stored, hence, the variable time (ϑ) does not appear in the heat balance. The model calculates an average temperature of the liquid inside the panel, by applying a second heat balance at the boundary between the panel surface and the surrounding environment. Figure 10 and Equation 18 show the second heat balance.

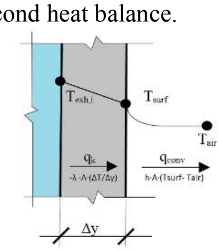


Figure 10: Heat balance at the boundary between radiator surface and surrounding environment

$$T_{liq} = T_{surf} + \left(T_{surf} - T_{air}\right) \cdot \frac{\Delta y \cdot h}{\lambda} \qquad (18)$$

The term Δy·h/λ is the thermal resistance which explain the mechanism of heat transfer between radiator and indoor air simplified into the coefficient k. The outlet temperature of the panel radiator is now calculated by replacing T_{liq} in Equation 17.

$$T_{out} = T_{air} + (T_{in} - T_{air}) \cdot e^{-\frac{k \cdot L}{\dot{m}_w \cdot c_w \cdot (T_{liq} - T_{air})^{n-1}}} \qquad (19)$$

T_{in} is the inlet temperature to the panel radiator set exactly as T_{sup} (Table 1).

Performance Comparison among Transient model, its lumped Steady State Model and IDA ICE Model

The outputs of transient model are compared among its lumped steady state model and IDA ICE model in terms of i) exhaust temperature and ii) heat emission. The lumped steady state comes from the transient model just by eliminating the storing term from

Equation 2, thus, $\dot{Q}_{injected} = \dot{Q}_{emitted}$. IDA ICE model is, as explain before, a lumped steady state model, which differs from the other steady state model only for the mechanism of heat transfer towards the indoor environment. Figure 11 compares the temperature of the exhaust flow, when the panel radiator is modelled with steady state model and transient model with one, two, four and eight storage elements. Dead and balancing time are computed numerically with T_d of 6 minutes and 30 seconds and T_b of 9 minutes and 30 seconds. Figure 12 compares the total heat emitted by the models. The blue dash lines represent the total heat emitted from the panel radiator modelled according to N=1,2,4,8 thermal capacitances and the red line in steady state condition. The grey area is the amount of energy overestimated from the steady state model. This area is about 50 Wh.

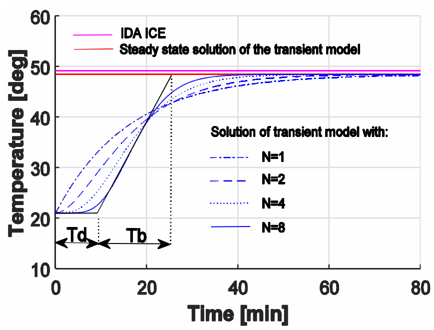


Figure 11: Temperature of exhaust flow for lumped steady state and transient models

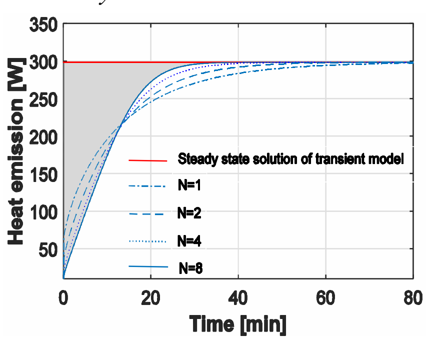


Figure 12: Heat emission of lumped steady state and transient model with 1,2,4 and 8 capacitances

UNCERTAINTY OF MODEL VALIDATION AND DISCUSSION

The validation of the numerical model is performed by tracking the temperature of mass flow rate. Three thermocouple TT are positioned on the supply, exhaust pipes and in the center of the panel radiator. The thermocouples are connected to a data tracker which records the temperature every 15s. The thermocouples are covered with insulation material for minimizing the disturbances from the indoor air. Figure 13 shows the temperature profile of supply, exhaust and on the middle of panel surface. It is possible to notice that, the temperatures arise almost abruptly at the input of supply heat flow. This is because of the amount mass flow rate injected was set too high at 0.04kgs^{-1}. The experiment was repeated with $\dot{m}_w$= 0.01, 0.06 kgs^{-1}. In the latter cases, the experimental results were strongly affected by the flow recirculation not providing any useful outcomes. Moreover, in the chart below, it is possible to notice some bumps due to the opening of the panel vent valve.

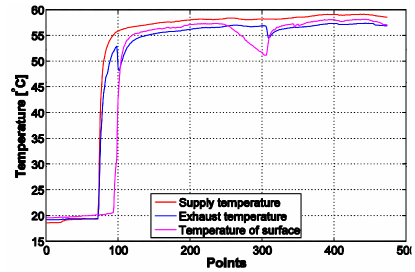


Figure 13: Temperature profile

The experimental results just give an idea of the temperature trend. The uncertainties in the experiment does not give a direct numerical validation between measured and simulated data. The heating process is also underlined by Figure 7, where the temperature of thermal capacitances arises one at a time. This phenomena is evident at 500s. The first capacitance is almost charged when the last one is still empty. The temperature of exhaust flow, in Figure 11, between steady state models (red and magenta lines) are similar in the trend, but it occurs a discrepancy between them magnitude. This is because the models employ different heat transfer mechanism between panel and indoor air. The steady state model uses the logarithmic temperature difference ΔT_{log}, instead, IDA ICE the so called power low relation. The results in Figure 12 are the amount of energy delivered into the environment. The lumped steady state model overestimates the heat emission during the charging phase. This result is essential for assessing the panel performance and efficiency for low energy buildings as state into EN 15316. The normative presents a gap to estimate the efficiency of radiators. The technical norm does not clarify which type of model should be employ for estimating the efficiency emission of sub-system for low energy buildings. Further investigations must be carried out to obtain new tabulated values of efficiency for hydronic heating systems by using sophisticated transient models.

CONCLUSION

The paper shows the transient model of a panel radiator modelled as a series of heat storage capacitances. The heating up phase, recorded with thermal imaging, clearly shows the charging direction of the thermal unit. The thermal imaging suggests the type of modelling that best suits for the case studied. The transient model keeps a memory of the heat injected into the panel. This essential characteristic is important for stocky panels, which need longer time for the charging phase. In addition, the total heat emitted towards the indoor environment is defined in the transient phase. The lumped steady state model overestimates the total heat emitted of 50Wh during this phase. The transient model is suitable for evaluating the efficiencies of low energy buildings. Moreover, this model allows controlling effectively the heat emitted towards the room when the indoor temperature oscillates. Lastly, EN 15316 should be re-discussed for calculating the efficiency for emission of hydronic sub-system. The norm does not give any clarification for dealing with such hydronic systems.

NOMENCLATURE

T	= temperature	[ºC]
$\dot{Q}$	= heat/power	[W]
M	= mass	[kg]
C	= capacitance	[$J^{o}C^{-1}$]
c	= specific heat capacity	[$J^{o}\ kg^{-1}\ {}^{o}C^{-1}$]
ΔT	= temperature difference	[ºC]
$\dot{m}$	= mass flow rate	[$kg\ s^{-1}$]
ϑ	= time step	[s]
S	= surface	[m^2]
σ	= S. - B. constant	[$Wm^{-2}K^{-4}$]
ϵ	= emissivity	
n	= radiator exponent	
s	= ratio	
k	= power law coefficient	[$Wm^{-1}K^{-n}$]
λ	= thermal conductivity	[$Wm^{-1}K^{-1}$]
Δy	= radiator thickness	[m]
x	= radiator length	[m]

subscript

$_{exh}$	= exhaust	$_{c}$	= time constant
$_{su}$	= supply	$_{d}$	= dead time
$_{w}$	= water	$_{b}$	= balancing time
$_{met}$	= metal	$_{wall}$	= wall
$_{rad}$	= radiator	$_{m}$	= mean temperature
$_{log}$	= logaritmic	$_{liq}$	= water flow
$_{N}$	= nominal		

ACKNOWLEDGEMENT

The authors are thankful to *Anders Strömberg* for his assistance and support during the experiment set-up. The first author would like to thank *Michael Würth* from the Fraunhofer Institute for Building Physics IBP for his valuable suggestions and *Muhammad Sikandar Lal Khan* for reviewing this paper.

REFERENCES

Directive 2010/31/EU of the European Parliament and of the Council of 19 May 2010 on the energy performance of buildings.

EN 15316-1 2007 Heating systems in buildings – Method for calculation of system energy requirements and system efficiencies –Part 1

EN 15316-2-3 2007 Heating systems in buildings – Method for calculation of system energy requirements and system efficiencies –Part 2-3: Space heating distribution systems

EN 15316-2-3:2007 Heating systems in buildings – Method for calculation of system energy requirements and system efficiencies –Part 2-3: Space heating distribution systems

EN 442-2:1996/A2:2003, Radiators and Convectors Part 2: Test Methods andRating, CEN, Brussels

European Standard EN. 15316-1, Heating Systems in Buildings: Method for Calculation of System Energy Requirements and System Efficiencies. Part 1: General, 2007.

Janćik L., Bašta, J., Dynamic behavior of panel radiators – further investigation, Proceedings of experimental Fluid Mechanics, 2009, TU Liberec, 2009, p.83

Janćik L., Bašta, J., Dynamic behavior of panel radiators – Thermographic investigation and computer modelling of heat-up, EPJ Web of Conferences 25, 2012

Holst S., Models for Radiator Heating System, TRNSYS TYPE 161-162, April 1996, Germany

IDA ICE software and user manual, Version 4.5, accessed Febraury 2015

Lenhovda Värmer, Panel radiator MP-R.O.T., accessed the 10th November 2014

Maivel, M. Kurnitski J., Low temperature radiator heating distribution and emission efficiency in residential buildings, Energy and Buildings, 2013, vol 69, pp. 224–236

Maivel, M., Konzelmann, M., Kurnitzki, J., Energy performance of radiators with parallel and serial connected panels, Energy and Buildings, 2015, vol. 86, pp. 745-753

Myers, G., Analytical methods in conduction heat transfer, McGraw-Hill, 2nd Edition, 1998

Mikk, M., Kurnitski, J., Low temperature radiator heating distribution and emission efficiency in residential buildings, Energy and Buildings, 2014, vol. 69, pp. 224-236

Myhren, J., Holmberg, S., Design considerations with ventilation-radiators: comparison to traditional two-panel radiators, Energy and Buildings, 2009, vol. 41, pp. 92-100

Siemens Switzerland, Guidebook Control Technology, Building Technologies Division, accessed March 2015

Stephan, W., System simulation, Specification: radiator International Energy Agency, Annex 10, University of Stuttgart

Tahersima, F., Stoustrup, J., Rasmussen, H., An analytical solution for stability-performance dilemma of hydronic radiators, Energy and Buildings, 2013, vol. 64, p.439-446

Tahersima, F., Stoustrup, J., Rasmussen, H., Nielsem, P. G., Thermal analysis of an HVAC system with TRV controlled Hydronic Radiator, 6th annual IEEE Conference on Automation Science and Engineering, August 2010, Toronto, Canada