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SPECIFYING BOUNDARY CONDITIONS FOR THE OPERATION OF PIPE HEATING SYSTEMS WITH IMPACT ON THE BUILDING ENERGY BALANCE

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Abstract. Unaddressed boundary conditions in the design of heating systems affect the energy balance of buildings, especially in buildings with very low energy consumption. Buildings with very low energy consumption are very sensitive to any heat flow and neglecting realistic heating water parameters affects their energy balance. Two simulation models of the end part of the heating system have been developed. The first simulation model shows the effect of the inaccuracy of the designed heating element on the calculated room temperature. The second model shows the hydraulic behaviour of the connection pipes of the designed heating element.

Keywords: Radiator, heating system, energy balance, low energy building heating system.

1. Introduction

In buildings with low energy consumption, we expect high efficiency of energy systems [1]. The use of hot water systems is most common due to the worst environmental impacts of direct electricity use. Therefore, it is important not only to fit new efficient heat sources and use modern control valves but also to focus on the most accurate calculations for their correct design. There is a lot of simplification of individual boundary conditions in the usual heating system design procedures, as they were fully suitable for older and simpler systems.

2. Heating system effect

The output of the heating elements that provide the indoor ambient temperature should be the same as the heat loss of the heated space. In this case, the heating system operates at its highest efficiency. There are several reasons why this condition may not be met. These include incorrect determination of room heat loss, oversized heating element performance, neglect of heat gain from visible or built-in ductwork, or incorrect heating water parameters – temperature or flow rate. At the same time, a different internal room temperature can also cause heat flows within an energy-efficient building that did not occur previously due to temperature equality. The dependence of the heating performance on the indoor temperature can be determined by calculation.

3. Model of the panel radiator

The room temperature indicates the calculated internal room temperature and is based on the rules specified in the CSN EN 12831-1 standard [2]. The standard considers radiators to have the same air temperature as the room temperature. The calculation is solved for steady state in accordance with the standard, and therefore the accumulation does not affect the resulting parameters. Heat losses and heat gains are solved in the calculation using their own control elements. The results of the behaviour of control elements are always considered after stabilization of the changed parameters. The following iteration formulas (1), (2) are used to calculate the operating room temperature.

$$Ti_{\mathrm{OP}} = \frac{m_{\mathrm{OP}} \cdot c \cdot (Tp_{\mathrm{OP}} - Tz_{\mathrm{OP}})}{H_{\mathrm{OP}}} + Te. \tag{1}$$

$$Tz_{\mathrm{OP}} = Tp_{\mathrm{OP}} - \frac{m_{\mathrm{N}} \cdot (Tp_{\mathrm{N}} - Tz_{\mathrm{N}})}{m_{\mathrm{OP}}} \cdot \left(\frac{\frac{Tp_{\mathrm{OP}}+Tz_{\mathrm{OP}}}{2} - \frac{m_{\mathrm{SK}} \cdot c \cdot (Tp_{\mathrm{OP}} - Tz_{\mathrm{OP}})}{H_{\mathrm{OP}}} - Te}{\frac{Tp_{\mathrm{N}}+Tz_{\mathrm{N}}}{2} - Ti_{\mathrm{N}}} \right)^n. \tag{2}$$

Subscripts are used in Equations (1) and (2):

OP operating,

N nominal.

These subscripts are used for the physical parameters:

Q radiator heating power [W],

m mass flow [$\mathrm{kg\,s^{-1}}$],

c specific heat capacity water [$\mathrm{J\,kg^{-1}K^{-1}}$],

n heating element exponent [-],

Tp supply water temperature [°C],

Tz return water temperature [°C],

Ti room temperature [°C],

Te external temperature [°C],

H specific heat flux through building structures [$\mathrm{W\,K^{-1}}$].

	Var. 1	Var. 2	Var. 3	Var. 4	Var. 5	Var. 6	Var. 7	Var. 8
Supply water temperature Tp_N [°C]	75	75	75	75	55	55	55	55
Return water temperature Tz_N [°C]	65	65	60	60	44	45	40	40
Real mass flow m_{OP} [$\mathrm{kg\,s^{-1}}$] =	m_N	m_P	m_N	m_P	m_N	m_P	m_N	m_P

TABLE 1. Boundary conditions to calculate the internal temperature of the room (m_N = mass flow for requested power output, m_P = mass flow for room heat loss).

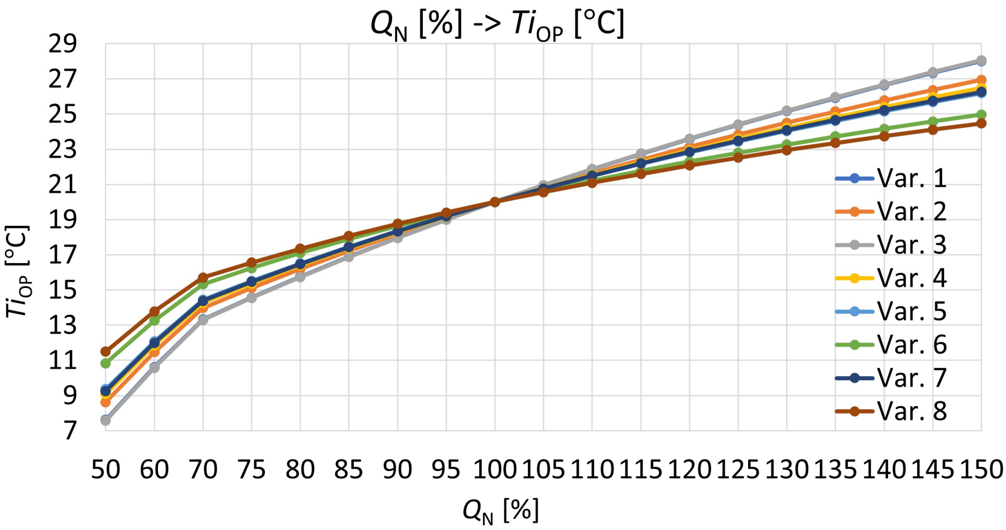

FIGURE 1. Calculated value Q_N [%] and Ti_{OP} [°C].

The panel radiator model contains 8 variants of different heating water temperatures and the resulting mass flow rates (see in Table 1).

Design room temperature $Ti_N = 20\,°C$, external temperature $Te = -15\,°C$, heating element exponent $n = 1.3$ (typical value for flat radiator), room heat loss $Q_P = 350\,W$.

The actual room temperature Ti_{OP} [°C] and the actual output of the heating body Q_N [%] were calculated.

3.1. THEORETICAL CASE STUDY OF THE PANEL RADIATOR

The results of modelling the given variants are presented in graphical form in the following figure for clarity.

From the results in Figure 1, it can be seen that the variants with real mass flow $m_{OP} = m_N$ (nominal value) have larger room temperature deviations Ti_{OP} than the variants with real mass flow $m_{OP} = m_P$ when the same % deviation of Q_N from Q_P is specified. Hence, it appears to be more advantageous to regulate the balancing at $m_{OP} = m_P$, as the effect of underrated Q_N is compensated here, and conversely, overrated Q_N does not overheat as much again.

It can also be seen that the larger deviations of the room temperature Ti_{OP} are at $Q_N < 100\,\%$ than at $Q_N > 100\,\%$, hence more problematic when undersizing the panel radiator.

However, in general the variations in room temperature are quite significant for individual Q_N <,> 100 % with respect to thermal comfort of the occupants in the heated room and the control range of the local heating element control. For Q_N = 110–120 % the pressure (according to variants Tp_N and Tz_N) is out of the range of the thermostatic valve with head. This implies that the valve closes already by design error and is therefore no longer able to react to random heat gains, which must thus legitimately start to overheat the heated space. For the calculated underpressure at $Q_N < 100\,\%$ (according to the variants Tp_N and Tz_N) it is important whether the valve on the heating element has the possibility of a so-called overflow compared to the balanced Kv value e. g. at $Xp = 2\,K$ and whether this overflow is able to raise the underpressure to the desired state. However, even in this case, the overflow when opening the valve reduces the effectiveness of the subsequent control of the random heat gains.

4. MODEL OF THE RADIATOR CONNECTION PIPE

Today, however, the requirements for systems are more complicated with the growth of many variant solutions, and at the same time computer programs no longer pose obstacles to more demanding mathematical models. One of the boundary conditions addressed is the effect of heating water temperature on the pressure loss in piping systems commonly known calculation for the total pressure loss is the sum of the pressure loss through the inserted resistances and the frictional

T_M [°C]	ρ [kg/m^3]	ρ [%]	ν [m^2/s]	ν [%]
75	974,97	100	3,75E-07	100
70	977,92	100,3	4,03E-07	107,4
65	980,69	100,6	4,34E-07	115,7
60	983,28	100,9	4,69E-07	125,0
55	985,69	101,1	5,08E-07	135,5
50	987,92	101,3	5,53E-07	147,4
45	989,97	101,5	6,04E-07	161,0
40	991,84	101,7	6,63E-07	176,6
35	993,53	101,9	7,31E-07	194,7
30	995,04	102,1	8,10E-07	215,9

TABLE 2. Comparison of kinematic viscosity ν and water density ρ as a function of mean water temperature T_M.

pressure loss according to the Darcy-Weisbach Equation (3) [3–5].

Table 2 shows a comparison of the deviations of the kinematic viscosity ν and density ρ as a function of the mean heating water temperature T_M. From Table 2 it can be seen that the influence of ρ is very small within units of %, but the influence of ν is significant.

Heating systems often operate at variable water temperatures during operation, whether it is the designed temperature gradient, the effect of qualitative regulation [5] or, for example, the temperature drop due to cooling of the heating water in the pipe route [6].

Therefore, a comparison of the pressure losses at heating water temperatures from 75 °C to 30 °C at a step of 5 °C was made. Other variable inputs for the determination of the pressure loss are the internal pipe diameter d, the wall roughness k and the mass flow rate m with respect to the type of flow. In the following section it is shown how ν and ρ affect the pressure drop through local resistances Z and the frictional pressure drop R in the different flow types.

Pressure loss through the local resistances Z depending on the heating water temperature depends only inversely on ρ according to Table 2. Thus, at $T_M = 30$ °C it is about 98 % compared to $T_M = 75$ °C. For this reason, we will only consider the frictional pressure drop R (3) in the following, where the problem is more complex.

$$R = \frac{\lambda \cdot w^2 \cdot \rho}{d \cdot 2}. \tag{3}$$

In this paper, the flow is considered for analysis in the laminar region according to Poiseuille (4).

$$\lambda_{\mathrm{lam}} = \frac{64}{Re}. \tag{4}$$

Substituting (4) into (3) we get the modified formula for the basic parameters (5).

$$R = \frac{0.04 \cdot m \cdot \nu}{\pi \cdot d^4}. \tag{5}$$

And in the smooth region of flow according to Blasius (6).

$$\lambda_{\mathrm{smooth}} = \frac{0.3164}{Re^{0.25}}. \tag{6}$$

Substituting (6) into (3) we get the modified formula for the basic parameters (7).

$$R = \frac{0.0005 \cdot m^{\frac{7}{4}} \cdot \nu^{\frac{1}{4}}}{\pi^{\frac{7}{4}} \cdot \rho^{\frac{3}{4}} \cdot d^3}, \tag{7}$$

where:

R friction pressure loss [Pa m^{-1}],

m mass flow [kg s^{-1}],

d piping diameter [m],

ρ water density [kg m^{-3}],

λ friction coefficient [-],

ν kinematic viscosity [m^2 s^{-1}],

Re Reynolds number [-],

w water speed [m s^{-1}].

4.1. Theoretical case study of the radiator connection pipe

The following example shows the flow in the laminar region for a Cu 15 × 1 pipe up to mass flow $m = 31$ kg s^{-1} for variable water temperatures. The connection of a heating element with a power of 550 W at $\Delta T = 15$ °C and 350 W at $\Delta T = 10$ °C is solved. For larger pipe dimensions where mass flows are also larger, laminar flow does not occur. As can be seen from Figure 2, the deviation increases at $T_M = 30$ °C up to 216 % compared to $T_M = 75$ °C, which is a lot, but it is necessary to look at the absolute values of the pressure drop. These values are at extremes of $R = 4.5$–9.6 Pa m^{-1} for $m = 30$ kg s^{-1}, which are very low for heating systems in energy efficient buildings and the difference does not have a significant effect on the resulting hydraulics.

In Figure 3 we can see that in the smooth flow zone, the change of friction loss depending on the temperature of heating water is caused by both ν and ρ with corresponding exponents. The deviation of friction loss at $T_M = 30$ °C is 119 % in comparison with $T_M = 75$ °C, which is not negligible. This behaviour

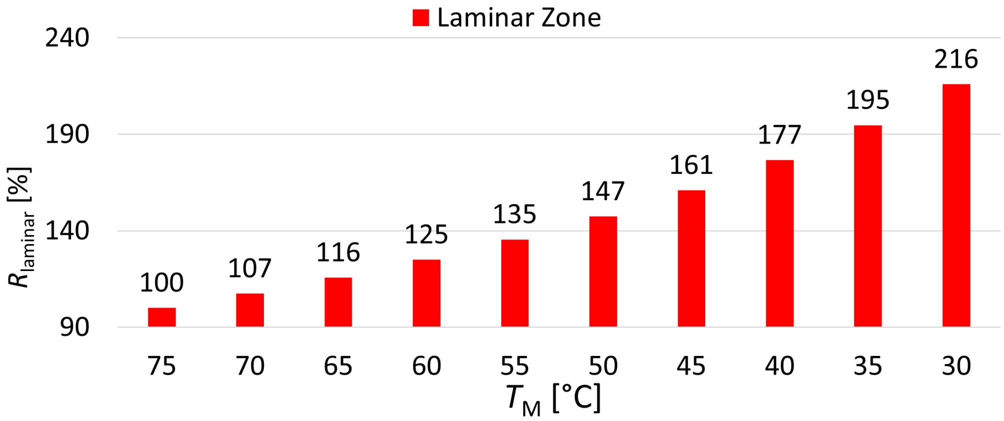

Figure 2. Relative pressure drop R for different mean heating element temperatures T_{M} (laminar flow).

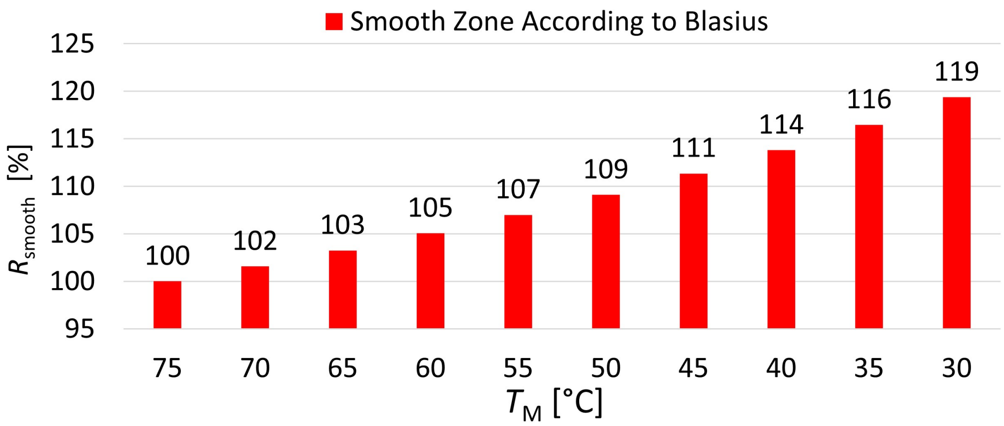

Figure 3. Relative pressure drop R for different mean heating element temperatures T_{M} (smooth zone).

of the deviance in the smooth zone is according to Blasius constant for all m, before the flow transforms from smooth to transition zone.

In general, for the solved variants with boundary conditions of conventional heating systems, the transition from laminar flow to turbulent flow behaves hydraulically smooth at first, even for values with piping roughness $k = 0.2\,\mathrm{mm}$. Only by further increasing mass flow does the smooth region variant change to a transition region. The rough hydraulic region is never reached in the boundary conditions for normal mass flow in the pipeline.

5. Conclusion

For all of the above reasons, it is recommended that the heat loss is calculated accurately, and the heating surface is designed for this loss correctly. If we avoid problems with an incorrect design of heating surface performance, we will have a heating system ready for operation with effective control efficiency and a positive effect on thermal comfort.

The results also showed that it is preferable to balance the design of the heating surfaces to the flow rate given by the design output, as it compensates for the problem of under- and overheating compared to heaters balanced to the flow rate given by the design output.

It was also observed that the internal temperature deviation increases faster for underbalanced heaters than for balanced heaters. An important piece of information is that oversizing a heater by about 15 % above the requirement will already overheat the room beyond the normal local control range, leading to cycling.

Furthermore, the effect of the heating water temperature was pointed out. In laminar flow, which occurs in the heating surface connections in modern heating systems, a large percentage deviation of R was shown with decreasing heating water temperature. It was also found that in the smooth region of turbulent flow, the pressure drop deviations with decreasing temperature go up to 119 %. This flow region is most commonly represented in today's heating system. This implies that for today's heating systems in modern low energy buildings, which are typically low temperature due to the small required heating element capacities and the requirements of heat sources such as heat pumps or gas condensing boilers, the pressure drops are proportionally higher for the same flow rate than for older buildings with high heating water temperatures. This has implications for the design and power consumption of circulators and the design of control and balancing valves.

Acknowledgements

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References

[1] European Union. Directive (EU) 2018/844 of the European Parliament and of the Council of 30 May 2018 amending Directive 2010/31/EU on the energy performance of buildings and Directive 2012/27/EU on energy efficiency. *Official Journal of the European Union* **61**(L156):75–91, 2018. `http://data.europa.eu/eli/dir/2018/844/oj`

[2] Standard CSN EN 12831-1 Energy performance of buildings – Method for calculation of the design heat load – Part 1: Space heating load, Module M3-3, 2021.

[3] A. Avci, I. Karagoz. A new explicit friction factor formula for laminar, transition and turbulent flows in smooth and rough pipes. *European Journal of Mechanics - B/Fluids* **78**:182–187, 2019. `https://doi.org/10.1016/j.euromechflu.2019.07.007`

[4] C. A. Dorao, T. Langeland, M. Fernandino. Effect of heating profile on the characteristics of pressure drop oscillations. *Chemical Engineering Science* **158**:453–461, 2017. `https://doi.org/10.1016/j.ces.2016.10.009`

[5] R. Petitjean. *Total Hydronic Balancing: A Handbook for Design and Troubleshooting of Hydronic HVAC Systems*. Tour & Andersson AB, 1994. ISBN 9163026260.

[6] J. Spurný, M. Kabrhel. Effect of heat distribution losses and cooling of heating water on design of heating system [In Czech]. (Vliv tepelných ztrát rozvodů a ochlazování otopné vody na návrh otopné soustavy). *Vytápění, větrání, instalace* **26**(1):2–5, 2017.