Energy analysis of low temperature heating systems

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Abstract: The assurance of the heat demand for millions of buildings equipped with centralized heating systems imposes equipment with high performances not only in the heat generation but also in the distribution of thermal energy. One way to obtain higher efficiency of the heating systems is to work on reduced temperature, which means that the temperatures throughout the system are lowered as much as possible. In this system the heat losses will decrease, the consumers will benefit a higher thermal comfort and it is possible to use with higher efficiency the renewable energy sources. The system must be controlled and optimized in correspondence with the ever-changed heat demand. In this paper are analysed the energy savings in heating systems with reduced forward temperature, for different types of radiators taking into account the thermal insulation of the distribution pipes. In this purpose are developed some calculation models and the effects of thermal agent temperature variation on the energy saving are illustrated by a numerical example.

Key-Words: Central heating systems, Low forward temperature, System control, Radiator heating, Floor heating, Energetical analysis, Energy saving, Computation models.

1 Introduction

Buildings are an important part of European culture and heritage, and they play an important role in the energy policy of Europe. Economical strategy of a sustainable development imposes certainly to promote efficiency and a rational energy use in buildings as the major energy consumer in Romania and the other member states of the European Union. Thus, conform the structure of energy consumption at world level buildings are the greatest energy consumer with about 45%, followed by industry and transports with 20%. From the total energy consumption of a building, about 54% represents heating and to cover this energy demand great quantities of fossil fuel are burned, which means considerable CO_2 emissions.

Due to the reduction of the fossil fuel reserves of the world and the strict environmental protection standards, one of the main research direction on the construction field is the reduction of the energy consumption, which suppose materials, technology and conception of buildings with lower specific enery need on one hand and equiupment with high performances on the other hand.

In central heating systems the hot-water forward temperature could have different values. In the recent past the most used value in Romania, as well as in other EU countries, was 90 °C with 20 °C temperature drop but nowadays the forward temperature usually is lower than 90 °C.

The assurance of the heat demand for buildings in Romania equipped with central heating imposes systems with high efficiency not only in the heat generation process but also in the distribution of the thermal energy. One way to obtain higher efficiency of the heating systems is to work on reduced temperature [3, 12]. In these systems the heat losses will decrease, the consumers will benefit a higher thermal comfort and it possibile to use renewable energy source with higher efficiency. The system must be controlled and optimized in corespondence with the ever-changed heat demand.

It is known that the energy and exergy efficiency of the central heating systems is higher at lower temperature of the hot-water [21], but based on [22] it have to be stated that this is valid only for total balanced systems. For the same value of the temperature drop, when the hot-water flow differs from the design value, as higher is the heat carrier temperature as stable the heating system is. Means that obtained indoor temperature is closer to the set-point value even if the flow is lower or higher than the design value, which results in better thermal comfort and lower energy losses. At the same time the low temperature central heating system stability could be improved decreasing the temperature drop. Thus we can obtain heating systems with a higher stability and energy efficiency decreasing in the same time the forward temperature and the temperature drop.

In this paper are analysed the energy savings in heating systems with reduced forward temperature, for different types of hot-water radiators taking into account the thermal insulation of the distribution pipes.

2 Hot water radiator heating

In centralized heating systems the transfer of thermal energy at the consumers is realized, in most of the cases, by static radiators. The heat is transferred by convention and radiation. The convection supposes circulation of the indoor air in the room. The place of the emitter and the channelling of the airflow could have an important influence on the heat transfered to indoor air. If the warm-air is guided along a low temperature surfaces (e.g. window) part of the transferred heat does not participate in the heating up the air in the occupancy zone. Thus the heat losses of the room will increase. At the same time the convective heat transfer will lead to a lower relative humidity of the air and, at high surface temperature of the radiator the dust particles could be burned, leading to a lower indoor air quality. Thus, emitters should be realized with a radiation factor as high as possible [26]. For the usual radiators, the values of the radiation factor are presented in table 1.

Table 1. Radiation factor of the usual emitters

Radiator type	Heat transferred by radiation		
	room-wards	wall-wards	total
0	1	2	3
Steel column radiator	0.28	0.10	0.38
Cast-iron column radiator	0.26	0.10	0.36
Panel radiator: 1/0*)	0.38	0.18	0.57
1/1	0.25	0.11	0.36
2/0	0.23	0.10	0.33
2/1	0.20	0.08	0.28
2/2	0.17	0.07	0.23
3/3	0.14	0.04	0.18

*) first number represents the number of radiative elements and the second the number of convective elements

The vertical temperature distribution in the room is more uniform at the radiators with lower carrier temperature, but it depends also on the wall surface temperature behind the heater and its geometrical characteristics. The high temperature of the carrier (hot-water) can lead to a lower thermal comfort level due to the asymmetric radiation.

The specific thermal energy transferred by a radiator depends on the carrier temperature:

$$q = q_0 \left(\frac{\Delta t}{\Delta t_0} \right)^{\alpha} \tag{1}$$

in which: q_0 is the specific thermal energy transfered by the radiator at nominal conditions (Δt_0 = 60 °C); α – an exponent with the value 1.24...1.36 for usual radiators; Δt – average logarithmical temperature difference, given by:

$$\Delta t = T_R = \frac{t_d - t_r}{\ln \frac{t_d - t_i}{t_r - t_i}} \tag{2}$$

where: t_d is the forward temperature of the carrier; t_r – return temperature; t_i – indoor air set-point temperature.

The variation of the radiator heat transfer coefficient k_R, depending on the carrier temperature, is given by [18]:

$$k_R = k_{R0} \left(\frac{\Delta t}{60} \right)^{\alpha - 1} \tag{3}$$

where k_{R0} is the heat transfer coefficient at the nominal conditions (t_d = 90 °C, t_r = 70 °C, t_i = 20 °C).

Figure 1 presents the variation curves for the k_R coefficient in function of the average logarithmical temperature difference $\Delta t = T_R$, for different values of α exponent.

For heating systems with nominal temperature lower than 90/70 °C the necessary radiator surface A_R will increase. Figure 2 presents the variation of radiator surface in function of the average logarithmical temperature difference for different α values. It can be observed a growing of the radiator surface (A_R/A_{R0}) while the values of the α exponent decrease for the same temperature difference Δt value.

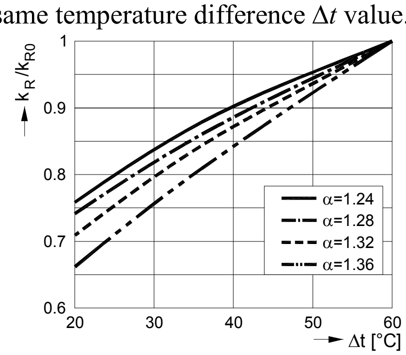

Fig. 1 Variation of heat transfer coefficient for the radiators

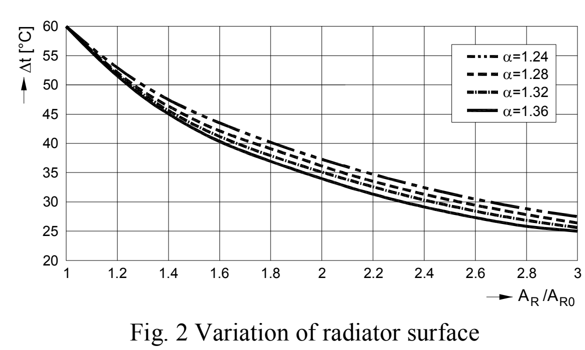

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Fig. 2 Variation of radiator surface

3 Control of central heating systems

To ensure the ever-changing heat demand are used sophisticated control systems. Depending on the controlled parameter the adjustment can be qualitative, quantitative or mixed.

To simplify the calculation the followed notations will be used:

$$T_a = t_i - t_e \tag{4}$$

$$T_d = t_d - t_i \tag{5}$$

$$T_r = t_r - t_i \tag{6}$$

According to the relation (5) and (6) the expression (2) for the radiators average logarithmical temperature difference ($\Delta t = T_R$) will be:

$$T_R = \frac{T_d - T_r}{\ln \dfrac{T_d}{T_r}} \tag{7}$$

where t_e is the outdoor air temperature.

Thus, the thermal balance equation will be:

$$Q_t = Q_R = Q_w \tag{8}$$

where:

$$Q_t = T_a \sum_j A_j k_j \tag{9}$$

$$Q_R = A_R k_R T_R \tag{10}$$

$$Q_w = m c_p \left(T_d - T_r\right) \tag{11}$$

in which: Q_t is the heat demand of the room; Q_R – heat transfered by the radiator; Q_w – heat tranfered by the hot-water; A_j – area of external building element j; k_j – heat transfer coefficient of external building element j; m – mass flow of the hot-water; c_p – specific heat of the hot-water.

The variation of thermal heat transfer coefficient k_R in function of the carrier temperature, according to relation (3) will be:

$$k_R = k_{R0}\left(\frac{T_R}{60}\right)^{\alpha - 1} \tag{12}$$

3.1 Qualitative control

In this case the controlled parameter is the carrier temperature and the flow rate is constant (m = const.) during the operation time. If the multiple forms of the heat balance equation (4) are used with explain parameters, for the nominal value of the controlled parameters (T_{a0}, T_{R0}) and for current values (T_a, T_R) lower than the design values, the following expression will be obtained:

– average logarithmical temperature difference:

$$T_R = T_{R0}\left(\frac{T_a}{T_{a0}}\right)^{\frac{1}{\alpha}} \tag{13}$$

– forward temperature of the carrier:

$$T_d = \frac{\dfrac{T_a}{T_{a0}}\left(T_{d0} - T_{r0}\right)\exp\left[\left(\dfrac{T_a}{T_{a0}}\right)^{\frac{\alpha-1}{\alpha}} \ln\dfrac{T_{d0}}{T_{r0}}\right]}{\exp\left[\left(\dfrac{T_a}{T_{a0}}\right)^{\frac{\alpha-1}{\alpha}} \ln\dfrac{T_{d0}}{T_{r0}}\right] - 1} \tag{14}$$

– return temperature of the carrier:

$$T_r = \frac{\dfrac{T_a}{T_{a0}}\left(T_{d0} - T_{r0}\right)}{\exp\left[\left(\dfrac{T_a}{T_{a0}}\right)^{\frac{\alpha-1}{\alpha}} \ln\dfrac{T_{d0}}{T_{r0}}\right] - 1} \tag{15}$$

It can be noticed that the forward and return temperature difference at the boiler, at different values of the outdoor air temperature is the same for any radiator type and any value of the forward carrier temperature.

3.2 Quantitative control

In this case the controlled parameter is the flow rate, the forward temperature remaining constant (t_d = const.) throughout the whole operation period. Due to the reduced carrier discharge, at higher oudoor air temperatures and at the constant forward temperature, the return temperature will be lower.

Using the equations (2) and (13) for the initial value of the outdoor air temperature T_{a0} and for any else one T_a, will be obtain:

$$\frac{T_a}{T_{a0}} = \left(\frac{T_d - T_r}{\ln\dfrac{T_d}{T_r}}\,\frac{\ln\dfrac{T_d}{T_{r0}}}{T_d - T_{r0}}\right)^{\alpha} \tag{16}$$

In figure 3 the variation of the return carrier temperature t_r depending on the outdoor air temperature t_e for different values of the forward temperature t_d and radiator exponent α is presented.

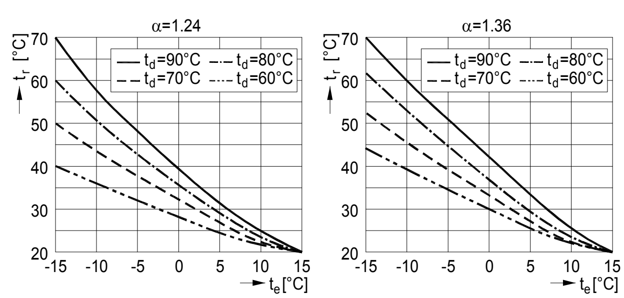

Fig. 3 Return temperature variation in function of outdoor air temperature

At the same time, writing the equation (8) for the initial value of the outdoor temperature T_{a0} and for any else T_a, the relations (9) and (11) results in:

$$\frac{m}{m_0}\frac{T_d - T_r}{T_d - T_{r0}} = \frac{T_a}{T_{a0}} \tag{17}$$

Thus, based on the relation (16) results:

$$\frac{m}{m_0} = \left(\frac{T_d - T_r}{T_d - T_{r0}}\right)^{\alpha-1}\left(\frac{\ln\frac{T_d}{T_{r0}}}{\ln\frac{T_d}{T_r}}\right)^{\alpha} \tag{18}$$

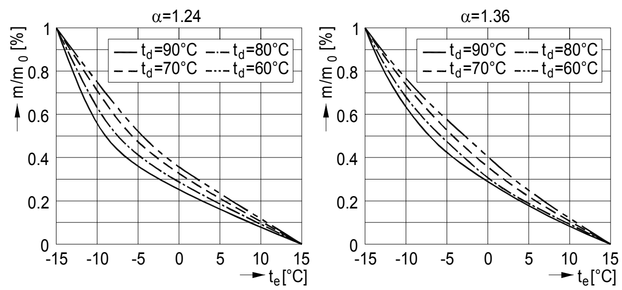

Fig. 4 Carrier flow rate variation

In figure 4 the variation curves of carrier mass flow m/m_0 for different values of the carrier forward temperature t_d and radiator exponent α are presented.

4 Thermal insulation of distribution pipes

In the case of a pipe without thermal insulation, the heat transfer coefficient k in W/(m·K) is determined with the well-known formula:

$$k = \frac{1}{\frac{1}{\alpha_i \pi D_i} + \frac{1}{2\pi\lambda}\ln\frac{D_e}{D_i} + \frac{1}{\alpha_e \pi D_e}} \tag{19}$$

where: α_i, α_e are the convective heat transfer coefficients at the internal and external surface of the pipe respectively, in W/(m^2·K); D_i, D_e – internal and external pipe diameter respectively , in m; λ – heat conductivity of the pipe material, in W/(m·K).

For metal pipes the first two terms of the denominator are much lower than the third and in followings will be neglected. Thus, the equation (19) become:

$$k = \alpha_e \pi D_e \tag{20}$$

Considering a unit length pipe the specific heat loss q, in W/m, will be:

$$q = k\Delta t = \alpha_e \pi \Delta t \tag{21}$$

where:

$$\Delta t = \frac{t_{wi} + t_{we}}{2} - t_e \tag{22}$$

in which: t_{wi}, t_{we} are the hot-water temperatures in the input and output section of the pipe respectively; t_e – air temperature around the pipe.

If the pipe is insulated, the efficiency of the thermal insulation layer η_{iz} can be defined as:

$$\eta_{iz} = \frac{q - q_i}{q} = 1 - \frac{q_i}{q} \tag{23}$$

where q_i is the specific heat loss of the insulated pipe, in W/m.

From the relations (20) and (23) results:

$$q_i = \alpha_e \pi D_e \Delta t (1 - \eta_i) \tag{24}$$

At the same time:

$$q_i = mc_p \Delta t_w = mc_p\left(t_{wi} - t_{we}\right) \tag{25}$$

The temperature difference from the equation (26) can be written as:

$$\Delta t = \frac{t_{wi} + t_{we}}{2} - t_e = t_{wi} - \frac{\Delta t_w}{2} - t_e \tag{26}$$

Using the relations (24) and (26) and taking into consideration that $mc_p \gg \alpha_e \pi D_e(1-\eta_{iz})/2$ results:

$$\Delta t_w = \frac{\alpha_e \pi D_e \left(t_{wi} - t_e\right)\left(1 - \eta_{iz}\right)}{mc_p} \tag{27}$$

The external surface conductance α_e depends on the difference between the pipe surface temperature t_s and outdoor air temperature t_e [27]:

$$\alpha_e = 8{,}1 + 0{,}045\left(t_s - t_e\right) \tag{28}$$

Taking into consideration the thermal insulation efficiency, in the case of metal pipes the equation (27) becomes:

$$\Delta t_w = \frac{\left[8{,}1 + 0{,}045\left(t_{wi} - t_e\right)\left(1 - \eta_{iz}\right)\right]\left(t_{wi} - t_e\right)\left(1 - \eta_{iz}\right)\pi D_e}{mc_p} \tag{29}$$

The ratio between the carrier temperature drops $\Delta t_w/\Delta t_{w0}$ at different water temperatures is given by:

$$\frac{\Delta t_w}{\Delta t_{w0}} = \frac{\left[8{,}1 + 0{,}045\left(t_{wi} - t_e\right)\left(1 - \eta_{iz}\right)\right]\left(t_{wi} - t_e\right)\rho_0}{\left[8{,}1 + 0{,}045\left(t_{wi0} - t_e\right)\left(1 - \eta_{iz}\right)\right]\left(t_{wi0} - t_e\right)\rho} \tag{30}$$

where ρ and ρ_0 are the hot-water density at t_{wi} and t_{wi0} temperature.

5 Numerical application

It is considered a heating system with a thermal load of 40 kW that assures the heat demand for a family house. The distribution pipe with a diameter of 40/32 mm is 10 m length, and the air temperature around the pipe is 10 ^oC. The velocity of water flow in the pipe is 0.48 m/s, at nominal parameters. It is illustrated the effect of hot-water temperature variation on the energy consumption for a heating system, using different control methods of the delivered heat quantity and different insulation level of the distribution pipes.

The carrier temperature drop, in these conditions, using the geometrical interpolation method [7], can be written in simplified form:

– for η_{iz}=0: $\Delta t_w = \left(-5 + 0.32t^{1.17}\right)\cdot 10^{-3}$ (31)

– for η_{iz}=0.7: $\Delta t_w = \left(-1 + 0.05t^{1.28}\right)\cdot 10^{-3}$ (32)

Using the relations (31) and (32) can be seen that the real values of the carrier temperature drop at the boiler are higher than the theoretical ones. The differences between the real and theoretical values are lower when the carrier temperatures are much lower and the differences are higher when the values of the radiator exponent are higher.

In table 2 the values of the real temperature drop at the boiler are presented, for qualitative control, depending on the outdoor temperature at different values of the carrier forward temperature and for different values of the thermal insulation layer efficiency.

In case of quantitative control the variation of the real temperature drop Δt_w, depending on the outdoor temperature t_e at different values of the hot-water forward temperature t_d and thermal insulation efficiency η_{iz} are presented in figure 5.

In figure 6 are presented the percentage of energy saving e_s for hot-water forward temperatures t_d lower then 90 °C. It can be observed that the energy saving increases for higher values of the radi-ator exponent α and for lower values of the carrier temperature, and the energy saving decreases when the thermal insulation level of the pipes is higher.

In figure 7 the energy saving e_s at different values of the carrier temperature t_d is presented, improved thermal insulation level (η_{iz} = 0.7) of the distribution pipe. It can be observed that the energy saving decreases at lower values of the carrier temperature increases with the radiator exponent α in the case of qualitative control and decreases at higher values of the radiator exponent in the case of quantitative control.

Table 2. Real values of the temperature drop for qualitative control

t_d [°C]	t_e [°C]	η_{iz}=0		η_{iz}=0.7		t_d [°C]	t_e [°C]	η_{iz}=0		η_{iz}=0.7	
		α=1.24	α=1.36	α=1.24	α=1.36			α=1.24	α=1.36	α=1.24	α=1.36
90	−12.8	19.676	19.678	18.988	18.989	70	−12.8	19.384	19.386	18.908	18.908
	−8.3	17.003	17.014	16.388	16.392		−8.3	16.748	16.756	16.319	16.321
	−5.0	15.043	15.059	14.482	14.487		−5.0	14.815	14.825	14.420	14.423
	0	12.071	12.094	11.594	11.600		0	11.884	11.899	11.543	11.547
	5.0	9.097	9.124	8.705	8.712		5.0	8.952	8.969	8.666	8.671
	10.2	5.999	6.028	5.699	5.707		10.2	5.899	5.917	5.673	5.678
80	−12.8	19.527	19.531	18.947	18.948	60	−12.8	19.244	19.246	18.870	18.870
	−8.3	16.874	16.884	16.353	16.355		−8.3	16.626	16.631	16.286	16.287
	−5.0	14.928	14.941	14.450	14.454		−5.0	14.705	14.712	14.391	14.393
	0	11.977	11.996	11.568	11.573		0	11.794	11.804	11.519	11.522
	5.0	9.024	9.046	8.685	8.691		5.0	8.881	8.894	8.648	8.651
	10.2	5.949	5.972	5.686	5.692		10.2	5.850	5.863	5.661	5.664

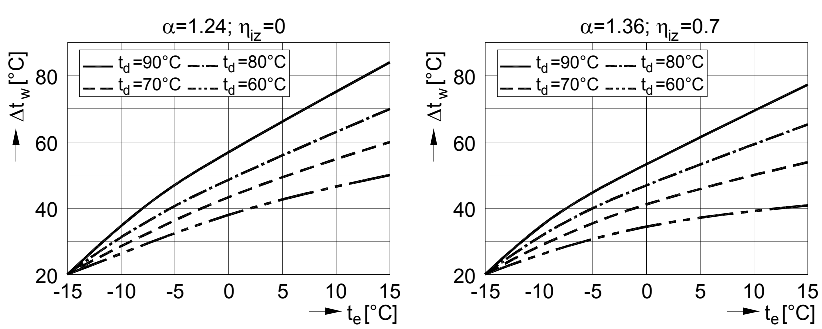

Fig. 5 Variation of the real temperature drop values for quantitative control

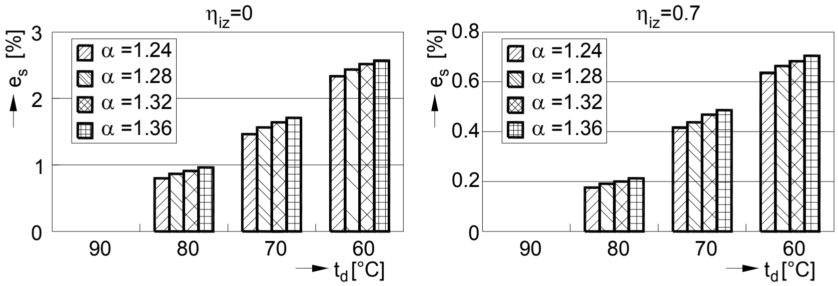

Fig. 6 Energy saving in function of forward temperature

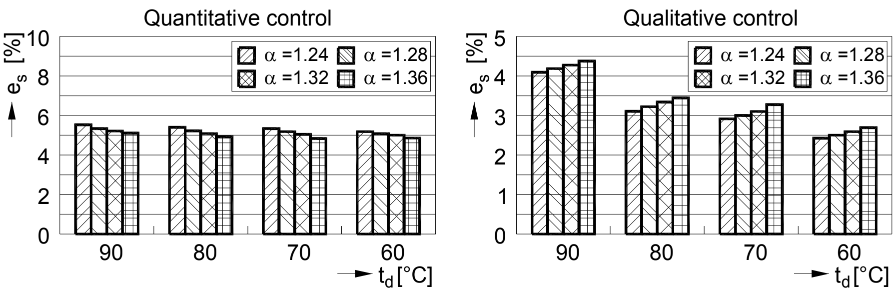

Fig. 7 Energy saving by pipe insulation in function of control system

6 Low temperature radiator heating

In order to facilitate the discussion, a concrete reference site was selectected, in the event a German house located in Munich with thermal insulation features complying with the current German En EV building code. The building has been named “Haus Oberbayern”. The climate conditions for a building in Munich can be reasonably deemed as the average climate conditions in Europe, and thus, such a building can be used as a reference.

Heat requirements of well-insulated buildings like “Haus Oberbayern” are low, and thus, the radiator heating system can be designed using low water temperatures, down to 40/30 °C, with fairly normal radiator sizes, case 1 (table 3).

If the reference building were equipped with an exhaust ventilation system instead of balanced ventilation and heat recovery, (case 2 and 3), the additional heat demand is about 15 W/m^2 corresponding to an air change rate of 0.5 h^{-1}. Energy from the high enthalpy exhaust air can be recovered by heat pump using the exhaust air as heat source. This is an energy efficient option not only for new buildings but for renovation, too.

One can state that the water temperatures 55/45 °C and 45/35 °C are suitable for radiator selection, but the 40/30 °C water temperatures are too low without pre-heating of the supply air and using oversiyed radiators. However, by using typical ventilation radiators, such as a panel radiator equipped with a supply air device (case 3), normal radiator sizes can be selected even at the lowest design temperatures.

Table 3. Comparative study shows radiators tested with varying design temperatures in the bedroom of the reference building

Case – System	Design temps flow/rtn/air,°C	Typical radiator characteristics type–height–length
0	1	2
1 Reference case Balanced ventilation and heat recovery Heat demand 421 W (29.0 W/m^2)	40/30/21	22–600–1400
	50/30/21	21–600–1400
	45/35/21	21–500–1400
	55/45/21	11–400–1400
2 Exhaust ventilation No pre-heated supply air Heat demand 621 W (42.8 W/m^2)	40/30/21	too low water temps
	50/30/21	33–500–1400
	45/35/21	22–600–1400
	55/45/21	21–400–1400
3 Exhaust ventilation Pre-heating supply air and ventilation radiators Heat demand 621 W (42.8 W/m^2)	40/30/21	22–500–1400
	50/30/21	21–500–1400
	45/35/21	21–500–1400
	55/45/21	21–300–1400
4 “Pasive house” Heat demand 202 W (13.9 W/m^2)	40/30/21	21–500–1400
	50/30/21	11–500–1400
	45/35/21	11–400–1400
	55/45/21	11–300–1200

In the case 4 of “Pasive House” the design water temperatures can be very low without making radiator selection difficult. Small heating needs in “Pasive Houses” make it very attractive to use low temperature radiator heating, which allows rapid and accurate temperature control.

Operating water temperatures are most of the time lower than the design temperatures because the flow water temperature is controlled in accordance with outdoor temperatures.

7 Floor heating

Floor heating construction corresponds with the mostly used wet system i.e. PEX pipes in the concrete slab.

Design temperatures of 40/30 °C are required when the floor surface material is parquet and/or a floor surface temperature of 27 °C is required in the bathroom. The temperature level 35/28 °C can be used when the heat conduction of the floor surface is high and the bathroom air temperature has been set at 23 °C.

Design temperature differences 7…10 °C between forward and return water are often too big when using the normal pipe dimensions in buildings characteristic of low heat demands. In practice higher water flow rates are in use leading typically up to 4…6 °C water temperature differences.

There are two main reasons behind the higher energy consumption of floor heating. Firstly, floor heating generates higher heat losses to the ground, although in the calculations insulation quality was acctually better than current German EnEV prescriptions, and secondly, the poorer controllability of the floor heating. Other research studies show that both lighter building construction and weaker thermal insulation increase the energy consumption difference between radiator and floor heating up to 16 percent [7].

Energy efficiency of floor heating can be improved substantially, up to level of radiator heating, by adding additional thermal insulation to the ground floor and using so-called dry floor heating construction. Unfortunately this is only carried out in about 2% of narkets, owing to slightly higher investment costs. The heat losses through the external wall behind the radiators depend on the radiator type, the radiator projection area on the back wall and also water temperatures. Calculations show figures typically less than 1% of the heating energy consumption in a well-insulated building.

The often-advertised self-control function of floor heating seems to be ineffective. When the indoor air temperature rises up to the floor temperature level, the convective heat flux from the floor stops, but the heat radiation continues from the floor to the other room surfaces with lower temperatures such as windows and external walls.

Calculated results of reference cases show that operative temperatures with floor heating during the heating season are on average only 0.22 °C higher than operative temperatures with radiators. Conclusion is that a reduction in indoor air temperature for floor heating due to the higher operative temperature will not give much benefit in terms of energy saving compared to radiator heating.

Despite the fact that the reference calculations made with the “Haus Oberbayern” show that the energy consumption differences do not run higher than 14% between radiators and floor heating to radiators benefit, measurements in the field actually show much larger differences though. Several measurements taken in Sweden and Denmark show that heating energy consumption of floor heated buildings was typically up to 30% higher than in buildings heated by radiators [8].

The reason for this increased energy consumption seems to be due to the activities of the building users. An important factor is the floor surface material’s impact on temperature perception. In a well-insulated building, the heat demand is low and even in the design circumstances, when the water temperatures are highest, the floor surface temperature is only about 23.8 °C in the reference building. If the floor materials conduct heat well, for example stone and tiles, the floor feels cool even at this temperature.

Floor surface temperatures do not differ significantly between radiator and floor heating in the reference building. Living room floor temperature duration curves do not show more than 1.3 °C differences (fig. 8).

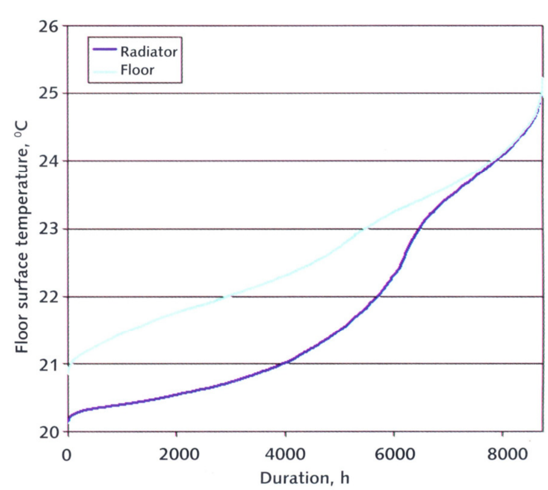

Fig. 8 Variation of floor surface temperature in time

However, the building occupants want the floor to feel warm to the feet and this is why they increase the water temperature to a level that makes the floor feel warm, sometimes even in summer. The warm temperature is typically more than 27 °C for stone based materials. The excess heat must be ventilated/ cooled in order to retain acceptable indoor air temperatures. This causes a huge increase in energy consumption. Cases in which the energy consumption has doubled have been observed in studies. In a well-insulated building the selected floor surface material is of crucial importance when it comes to how warm the floor feels. For example oak parquet at a temperature of 21 °C and stone floor at a temperature of 26 °C feel neutral and roughly the same under a bare foot [28].

8 Energy efficiency and heat pump

The efficiency of a heat pump is described generally as the coefficient of performance COP, defined simply as the quantity of heat delivered by the heat pump divided by the energy need to drive the process.

It is possible to calculate the annual coefficient of performance COP_a values at an applicable accuracy for comparison purposes [29]. COP_a value is an estimation based on calculation of energy consumption and behaviour of a building over one year period. COP_a is often expressed also as SPF (Seasonal Performance Factor) [23]. For example in a building like “Haus Oberbayern”, where typically the energy consumption of the heating is about 9000 kWh/a and the domestic hot water energy about 4500 kWh/a, the following values in table 4 are obtained.

Table 4. Values calculated for an electric ground-source heat pump that heats also the domestic hot water

Design temperatures frwd/rtn/air, °C	Condensate temperature °C	COP_a
0	1	2
55/45/21	49.2	3.2
50/40/21	44.0	3.3
50/30/21	38.7	3.5
45/35/21	38.8	3.5
40/30/21	33.7	3.6
35/28/21	30.2	3.8

Calculation were made by taking into account the thermal features of reference building and also the corrected heat pump condensate temperatures that depend on both heating system flow and return water temperatures. For instance for radiator design suitable temperature systems 50/30/21 °C and 45/35/21 °C give practically the same condense temperatures and respectively the same COP_a values of 3.5 (table 4). On the other hand, from the radiator selection point of view design temperatures of 45/35/21 °C are more useful than 50/30/21 °C due to higher excess temperature (logarithmic mean temperature difference) and therewith 11% higher radiator heat outputs.

Several field test figures with floor-heated buildings have given COP_a values typically 3.4…3.8 calculated COP_a value of 3.6 in table 4 corresponds with a typical floor heating design temperature level of 40/30/21 °C or alternatively e.g. 37/32/21 °C that gives also the same level condensate temperature according to the laboratory measurements.

9 Conclusions

Radiator heating is an energy efficient heating system, which can be used successfully in both new and renovated buildings.

Optimum control of heating systems can be realized only if other than design parameters are taken into account, as well.

The operation of heating systems at lower temperature results in considerable energy saving. It can be applied in both new and existing systems. After a thermal rehabilitation of the building the surface of existing emitters would exceed and the decreased heating load and the output of the system can be adjusted to the new conditions by reducing the forward temperature.

Reduced forward temperature facilitates the use of renewable energy sources and saving of fossil fuel.

Radiator heating system in well-insulated buildings can be designed for very low water temperatures and in these cases the energy efficiency of the radiator heated building is up to 14% better than for floor heating according to the dynamic assessment above. It is simply not possible to raise the floor heating to the prime position as far as energy consumption is concerned, and not even when using heat pumps due to the noted smaller differences in the COP values.

There is plainly no reason to use higher than 55/45 °C design water temperatures in well-insulated buildings. Instead of commonly used term “Hot water radiator heating” it is scientifically reasonable and commercially sensible to use nowadays the term “Low temperature radiator heating”.

Design water temperatures 45/35 °C give also a good common basis for a combined heating system using radiators as primary heat emitters and floor heating in rooms, where higher floor temperatures are prefered e.g. in bathroom [9].

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