Energetical optimization of central heating systems

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Abstract: The main goal of the central heating design process is to obtain appropriate thermal comfort minimising the investment and operational costs. This scope could be theoretical realised with the available modern control techniques. However practice has shown that, if the preconditions are not fulfilled, the correct operation of the heating system could not be assured even by the most sophisticated control equipment. Therefore, the control elements could not fulfil their function if the warm water flow differs from the designed value and the thermal comfort is not realised unless with higher energy consumption. 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 is studied the operation of unbalanced central heating systems, and are analysed the thermal comfort and the energy consumption. Also, 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, Hydraulic unbalanced operation, Thermal comfort, Indoor temperature, Energy consumption, Low forward temperature, Energetical analysis, Energy saving, Computational 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. 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.

Studies have shown that saving energy is the most cost effective method to reduce green house gas emissions. It has also pointed out that buildings represent the biggest and most cost effective potential for energy savings. The reduction of 26% energy use is set as a goal for buildings by the year 2020 which corresponds to 11% of the reduction of total energy use in European Union (EU) countries [1].

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.

Energy requirements and fuel consumption of HVAC systems directly affect a building's operating cost and indirectly affect the environment [3]. The main purpose in the design of the central heating system is to obtain indoor comfort with diminished investments and operating costs. This purpose could be theoretical realized with the available modern balance and control technique. However, the practice has shown that, if the preconditions are not fulfilled the correct operation of heating system could not be assured even by the most performed control equipment.

Thus, the control equipments could not fulfil their function if the hot-water flow differs from the designed value. This situation could often appear in practice because of lack of the proper balance equipment on one hand and differences between the executed and designed system on the other hand. Therefore the thermal comfort can be realised only with surplus energy consumption.

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 tempe-rature 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 for the heat generation process but also for the distribution of the thermal energy. One way to obtain higher efficiency of the heating systems is working with reduced temperature [2, 7]. In these systems the heat losses will decrease, the consumers will benefit a higher thermal comfort and is possibile to use renewable energy source with higher efficiency. The system must be controlled and optimized in corespondence with the ever-changed heat demand.

This paper presents a study of consequents by unbalanced operation of central heating systems and an analysis regarding thermal comfort assurance in buildings and the energy consumption of the system. Also, 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 Energy analysis of unbalanced central heating systems

The so called unbalanced operation of a pipe heating system is not technical because always the carrier (warm water) movement in a branched or looped system is natural balanced, the fluid flow distribution respects the energy and mass conservation laws. This name was adopted considering his often use when it is specified that the discharges through the pipes of a heating network is not established through design.

Avoiding and correcting the negative consequences by unbalanced operation of centralized heating systems, requires upgrading with special equipments for balancing and control such as pressure drop regulators, adjustable valves and thermostatic valves at the radiators. This equipments of the systems assures the dynamic state and in consequents the consumers self correction in heat supply. However, the control equipments could not fulfil their function if the hot-water flow differs from the designed value. Therefore the thermal comfort can be realised only with surplus energy consumption.

2.1 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 transfered 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 [21].

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 transfered by a radiator depends on the carrier temperature:

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

in which: q_0 is the specific thermal energy transfered by the radiator at nominal conditions ($\Delta t_0 = 60$ °C); α – an exponent with the value 1.24...1.36 for usual radiators; Δt – average logarithmical temperature difference, given by:

$$\Delta t = T_R = \frac{t_d - t_r}{\ln \dfrac{t_d - t_i}{t_r - t_i}} \qquad (2)$$

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 [13]:

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

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 necessary 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

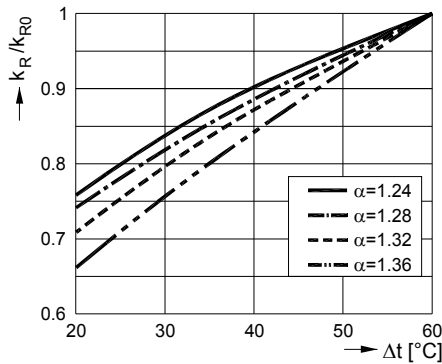decrease for the same temperature difference Δt value.

Fig. 1 Variation of heat transfer coefficient for the radiators
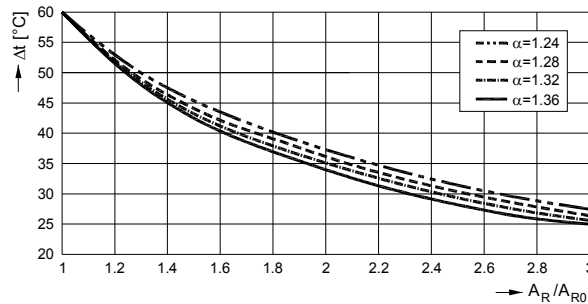

Fig. 2 Variation of radiator surface

2.2 Indoor temperature

The thermal comfort of a room is appropiate if the comfort parameters are kept at the values correspondingly to the building/room destination. One of the most important comfort parameter is the operative temperature, which can be defined as the average of the mean radiant and ambient air temperatures weighted by their respective heat transfer coefficients. The heat demand of rooms and the radiator outputs are calculated taking into account the indoor air set-point temperature.

Based on the specific degree-days curve the design value of the outdoor temperature occurs 2…5 times in a heating season, therefore the heating system output have to be controlled. In the case of a qualitative control system the hot-water forward temperature theoretical follow the control curve, where, depending of the design values of the forward temperature and temperature drop, the required values of the forward temperatures could be determined taking into account the outdoor temperature. In practice the characteristic curve of the control valves is linear, so thus two parallel lines gives the forward and return temperatures.

In a simplified form, the thermal balance is expressed by the following relations [16]:

$$Q_t = Q_R = Q_w \qquad (4)$$

where:

$$Q_t = \sum_j A_j k_j \left(t_i - t_e \right) \qquad (5)$$

$$Q_R = A_R k_R T_R \qquad (6)$$

$$Q_w = m c_p \left(t_d - t_r \right) \qquad (7)$$

in which: Q_t is the transmission heat losses in external building elements; 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; t_i – indoor air temperature; t_e – outdoor air temperature; A_R – area of the radiator surface; k_R – thermal transfer coefficient of radiator; T_R – average logarithmical temperature difference; m – mass flow of the hot-water; c_p – specific heat of the hot-water.

Using the equations (4), (5) and (7) the theoretical control curve can be determined, as follows: if the index „0" characterise the design situation, then for any other case the followed equation can be written [15], [19]:

$$\frac{m}{m_0} \frac{t_d - t_r}{t_{d0} - t_{r0}} = \frac{t_i - t_e}{t_{i0} - t_{e0}} \qquad (8)$$

At the same time, using equations (4), (5) and (6), the average logarithmical temperature difference T_R could be expressed:

$$T_R = T_{R0} \left(\frac{t_i - t_e}{t_{i0} - t_{e0}} \right)^{\frac{1}{\alpha}} \qquad (9)$$

In the case of a qualitative control system the ratio of the hot-water mass flow is constant and equal to 1 for any value of the outdoor air temperature. Consequently, using the equations (8) and (9) a general relation can be written:

$$F(t_d) = T_{R0} \left(\frac{t_i - t_e}{t_{i0} - t_{e0}} \right)^{\frac{1}{\alpha}} \ln \frac{t_d - t_i}{t_d - t_i - \frac{t_i - t_e}{t_{i0} - t_{e0}} \left(t_{d0} - t_{r0} \right)}$$

$$- \frac{t_i - t_e}{t_{i0} - t_{e0}} \left(t_{d0} - t_{r0} \right) = 0 \qquad (10)$$

In this equation if the indoor temperature is constant and equal to set-point value, at different outdoor temperature values the required forward temperature t_d could be calculated.

However, if the hot-water flow at radiators is different from the designed value, the return temperature of hot-water t_r is not according to the theoretical curve, as it could be seen in figure 3.
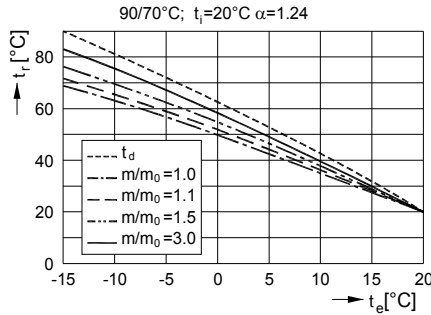
Fig. 3 Return temperature variation with outdoor
temperature at different flow ratio

In this case the indoor air temperature is different from the designed value. Assuming steady state process, the indoor air temperature values t_i could be determined from the following equation:

$$F(t_i) = T_{R0}\left(\frac{t_i - t_e}{t_{i0} - t_{e0}}\right)^{\frac{1}{\alpha}} \ln \frac{t_d - t_i}{t_d - t_i - \frac{t_i - t_e}{t_{i0} - t_{e0}}(t_{d0} - t_{r0})\frac{m_0}{m}} -$$

$$- \frac{t_i - t_e}{t_{i0} - t_{e0}}(t_{d0} - t_{r0})\frac{m_0}{m} = 0, \qquad (11)$$

in which the values of the forward temperature are introduced according to the control curve.

Indoor air temperatures values obtained for different mass flow rates, are presented in figure 4, where the linearly variation could be observed. Therefore for given forward temperature and temperature drop and given indoor set-point temperature, the variation of indoor air temperature could be written as:

$$t_i = t_{i0} + (t_{i0} - t_e)\mathrm{grad}\,t_i \qquad (12)$$

where the gradient of the indoor temperature is given by:

$$\mathrm{grad}\,t_i = \frac{t_{i(t_{e0})} - t_{i0}}{t_{i0} - t_{e0}} \qquad (13)$$

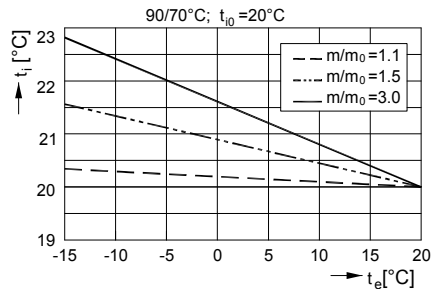in which t_{e0} is the design value of the outdoor temperature.

Fig. 4 Indoor air temperature variation
in steady state conditions

Consequently is enough to determine the indoor temperature value at a single value of the outdoor temperature (e.g. design value t_{e0}) at any other

outdoor temperatures the indoor temperatures could be determined using equation (12). Figure 5 presents the variation of indoor air temperature t_i, taking into account the mass flow ratio m/m_0, for different values of set-point temperature t_{i0}. In figure 6 is presented the indoor temperature variation t_i according to mass flow ratio m/m_0 for different values of temperature drop t_d/t_r in the system.

At different values of the set-point temperature if the mass flow exceds the design value the difference between the resulted values of the internal temperature hardly varies. Nevertheless at mass flow ratio values lower than 1 the difference between the obtained indoor temperatures decrease.
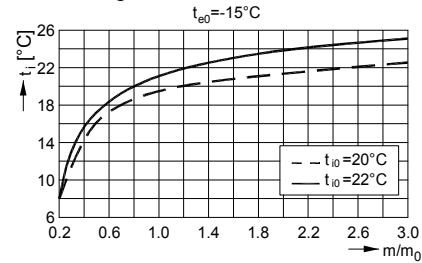

Fig. 5 Indoor temperature variation at
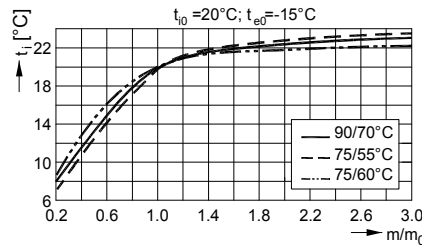90/70 $^{\circ}$C temperature drop

Fig. 6 Indoor temperature variation with flow ratio
at different temperature drops

So, at higher values of the the set-point temperatures if the hot-water flow is lower than the designed one the differences between the set-point and the obtained indoor temperature are higher. At the same time the designing value of the forward temperature and of the temperature drop influence as well the obtained indoor temperature. At the same value of the temperature drop the lower forward temperature will lead to higher absolute values of the differences between the obtained indoor temperature and the set-point values if mass flow differs from the design value. At the same value of the forward temperature the lower temperature drop will lead to a higher stability of the indoor temperature.

2.3 Heat losses and energy consumption
In the followings the heat losses by transmission through the internal walls will be neglected. If the hot-water flow at the radiators is higher than the

required value, then the indoor air temperature will exceed the set-point value and the heat losses of the room will increase. The heat losses growth depending on the outdoor temperature at a 50% extra flow is presented in figure 7.

The annual heating energy consumption of the room could be determined with the relation:

$$E = \frac{K}{\eta} \int_0^N (t_i - t_e) \mathrm{d}x \qquad (14)$$

where: K is total heat losses coefficient of the room/building; η – efficiency of the heating system; N –number of days in a heating period.
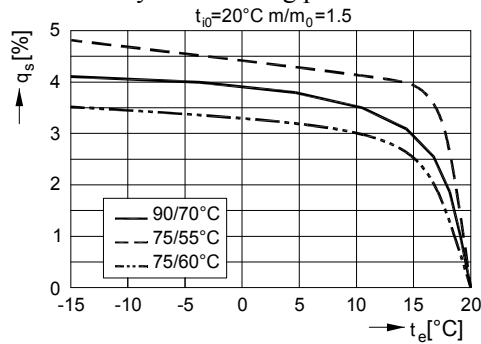

Fig. 7 Heat losses variation with outdoor
air temperature

In this equation the indoor set-point temperature and the average daily outdoor temperature are taken into account. Therefore the energy consumption ratio when the hot-water flow differs from the design value will be:

$$\frac{E}{E_0} = \frac{\displaystyle\int_0^N (t_i - t_e)\mathrm{d}x}{\displaystyle\int_0^N (t_{i0} - t_e)\mathrm{d}x} \qquad (15)$$

In this relation the indoor set-point temperature t_{i0} is known, the mean outdoor temperature variation could be described based on the degree-days curve [14], [19] and the obtained indoor temperature value t_i could be calculated with relation (12).

Using the geometrical interpolation method [12] the degree-days curve, for Timisoara, can be approximated with a function:

$$t_e = -15 + 3.55 x^{0.3835} \qquad (16)$$

in which x is number of days with the same average outdoor temperature t_e.

Substituting the known values, we obtain:

$$\int_0^N (t_{i0} - t_e)\mathrm{d}x = (t_{i0} - t_{e0})N - 2.566 N^{1.3835} \qquad (17)$$

respective:

$$\int_0^N (t_i - t_e)\mathrm{d}x = \left[(t_{i0} - t_{e0})N - 2.566 N^{1.3835}\right](\mathrm{grad}t_i + 1) \qquad (18)$$

Thus, the ratio of the annual energy consumption becomes:

$$\frac{E}{E_0} = \mathrm{grad}t_i + 1 \qquad (19)$$

So that the surplus energy consumption Δe of a room when the hot-water flow exceeds the design value will be:

$$\Delta e = \frac{E}{E_0} - 1 = \mathrm{grad}t_i \qquad (20)$$

The value of surplus energy consumption is independent from the indoor set-point temperature value. Figure 8 presents the percentage variation of excess energy consumption in function of different operational parameters of the central heating system ($t_{i0} = 20$ °C).
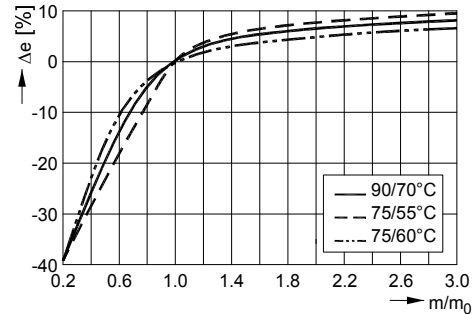

Fig. 8 Excess energy consumption variation

2.4 Building elements influence
The above presented results are valid for a room characterised by a time constant equal to zero. Nevertheless, if the heat storage capacity of the building elements is taken into account the obtained indoor temperature will be closer to the set-point value when the hot-water flow differs from the design value. The decrease or increase of the obtained indoor temperature is a result of the stored heat variation. For the heat storage calculation it was considered only the active layers with 3 cm thickness at the internal surface of the building elements [20]. To simplify the calculation the temperature of this active layer is assumed to be equal to surface temperature.

The specific surface stability of a building element expresses the relation between the specific heat flow q and the variation of indoor air temperature t_i. Thus, the specific surface stability B could be written as:

$$B = \frac{\mathrm{d}q}{\mathrm{d}t_i} = \frac{\alpha_i U_i}{\alpha_i + U_i} \qquad (21)$$

in which α_i is the convective heat transfer coefficient at the inside of element.

The heat absorption coefficient U_i of the internal surface is defined as a ratio of heat flow q and surface temperature variation t_s:

$$U_i = \frac{dq}{dt_s} \qquad (22)$$

Using the relation (21) and (22) is obtained the relation between surface and indoor air temperature:

$$dt_s = \frac{\alpha_i}{\alpha_i + U_i} dt_i \qquad (23)$$

If the time period taken into account is $\tau_0 = 24$ h and if $\tau = 0$ the surface temperature $t_{s0} = t_{i0}$, then based on the energy balance it may be written:

$$mc_p(t_d - t_r) = K(t_i - t_e) + C(t_s - t_{s0}) \qquad (24)$$

where: c_p is specific heat of hot-water heating; C − heat capacity of the room.

Taken into account the relation (23), the equation (24) becomes:

$$\frac{mc_p(t_d - t_r)}{K} = (t_i - t_e) + \frac{T}{24} \frac{\alpha_i}{\alpha_i + U_i} (t_i - t_{i0}) \qquad (25)$$

in which T is time constant of room/building.

If equation (25) is written for the design condition and for any other one, we obtain the ratio:

$$\frac{m}{m_0} \frac{t_d - t_r}{t_{d0} - t_{r0}} = \frac{(t_i - t_e) + \frac{T}{24} \frac{\alpha_i}{\alpha_i + U_i} (t_i - t_{i0})}{t_{i0} - t_{e0}} \qquad (26)$$

Using the equations (26) and (9), as well as the heat storage capacity of building elements, the equivalent of the equation (11) could be written as:

$$F(t_i) = T_{R0} \left(\frac{(t_i - t_e) + \frac{T}{24} \frac{\alpha_i}{\alpha_i + U_i} (t_i - t_{i0})}{t_{i0} - t_{e0}} \right)^{\frac{1}{\alpha}} \times$$

$$\times \ln \frac{t_d - t_i}{t_d - t_i - \frac{(t_i - t_e) + \frac{T}{24} \frac{\alpha_i}{\alpha_i + U_i} (t_i - t_{i0})}{t_{i0} - t_{e0}} (t_{d0} - t_{r0}) \frac{m_0}{m}} -$$

$$- \frac{m_0}{m} (t_{d0} - t_{r0}) \frac{(t_i - t_e) + \frac{T}{24} \frac{\alpha_i}{\alpha_i + U_i} (t_i - t_{i0})}{t_{i0} - t_{e0}} = 0$$

$$(27)$$

By solving this equation, we obtain the indoor air temperature values depending the mass flow ratio at different values of the outdoor air temperatures (fig.9). The calculation was made for a room with two external walls, before and after thermal rehabilitation (12 cm expanded polystyrene on the external surface). The initial time constant was $T =$

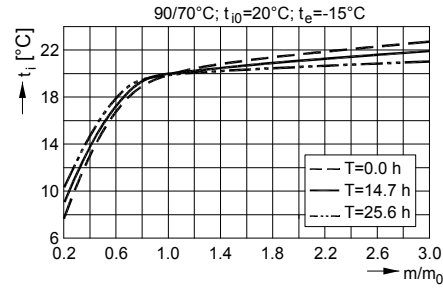14.7 h and became $T = 25.6$ h after thermal rehabilitation.

Fig. 9 Indoor temperature variation with flow ratio at different time constant values

Figure 10 presents the surplus energy consumption variation Δe depending the hot-water flow ratio for indoor temperature $t_{i0} = 20$ °C and the time constant $T = 14.7$ h. The excess energy consumption will be lower with 2…3% after thermal rehabilitation, according to the mass flow ratio. Furthermore the system could be steadier at lower forward temperature and temperature drop.
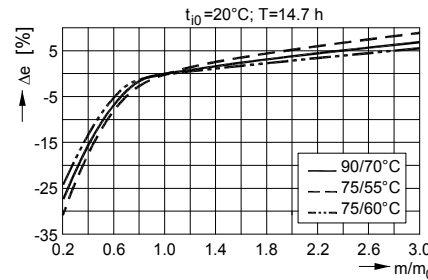

Fig. 10 Excess energy consumption at various operating parameters of the heating system

3 Energy efficiency of low temperature central heating systems

It is known that the energy and exergy efficiency of the central heating systems is higher at lower temperature of the hot-water [16], but based on figure 8 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.

3.1 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:

$$T_a = t_i - t_e \qquad (28)$$

$$T_d = t_d - t_i \qquad (29)$$

$$T_r = t_r - t_i \qquad (30)$$

According to the relation (29) and (30) the expression (2) for the radiators average logarithmical temperature difference T_R will be:

$$T_R = \frac{T_d - T_r}{\ln \dfrac{T_d}{T_r}} \qquad (31)$$

3.1.1 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:

$$T_R = T_{R0} \left(\frac{T_a}{T_{a0}} \right)^{\frac{1}{\alpha}} \qquad (32)$$

– forward temperature of the carrier:

$$T_d = \frac{\dfrac{T_a}{T_{a0}} (T_{d0} - T_{r0}) \exp\left[\left(\dfrac{T_a}{T_{a0}} \right)^{\frac{\alpha-1}{\alpha}} \ln \dfrac{T_{d0}}{T_{r0}} \right]}{\exp\left[\left(\dfrac{T_a}{T_{a0}} \right)^{\frac{\alpha-1}{\alpha}} \ln \dfrac{T_{d0}}{T_{r0}} \right] - 1} \qquad (33)$$

– return temperature of the carrier:

$$T_r = \frac{\dfrac{T_a}{T_{a0}} (T_{d0} - T_{r0})}{\exp\left[\left(\dfrac{T_a}{T_{a0}} \right)^{\frac{\alpha-1}{\alpha}} \ln \dfrac{T_{d0}}{T_{r0}} \right] - 1} \qquad (34)$$

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.

3.1.2 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 tempe-rature, the return temperature will be lower.

Using the equations (2) and (32) for the initial value of the outdoor air temperature T_{a0} and for any else one T_a, will be obtain:

$$\frac{T_a}{T_{a0}} = \left(\frac{T_d - T_r}{\ln \dfrac{T_d}{T_r}} \cdot \frac{\ln \dfrac{T_d}{T_{r0}}}{T_d - T_{r0}} \right)^{\alpha} \qquad (35)$$

In figure 11 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.
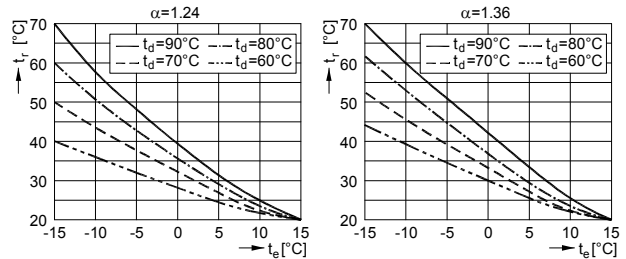

Fig. 11 Return temperature variation in function of outdoor air temperature

At the same time, writing the equation (4) for the initial value of the outdoor temperature T_{a0} and for any else T_a, the relations (5) and (7) results in:

$$\frac{m}{m_0} \cdot \frac{T_d - T_r}{T_d - T_{r0}} = \frac{T_a}{T_{a0}} \qquad (36)$$

Thus, based on the relation (35) results:

$$\frac{m}{m_0} = \left(\frac{T_d - T_r}{T_d - T_{r0}} \right)^{\alpha-1} \left(\frac{\ln \dfrac{T_d}{T_{r0}}}{\ln \dfrac{T_d}{T_r}} \right)^{\alpha} \qquad (37)$$
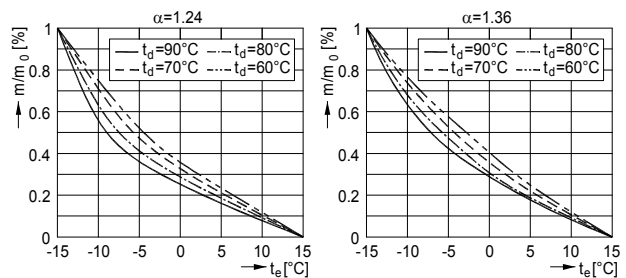

Fig. 12 Carrier flow rate variation

In figure 12 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.

3.2 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:

$$k = \frac{1}{\dfrac{1}{\alpha_i \pi D_i} + \dfrac{1}{2\pi\lambda}\ln\dfrac{D_e}{D_i} + \dfrac{1}{\alpha_e \pi D_e}} \qquad (38)$$

where: α_i, α_e are the convective heat transfer coefficients at the internal and external surface of the pipe respectively, in W/(m²·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 (38) become:

$$k = \alpha_e \pi D_e \qquad (39)$$

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

$$q = k\Delta t = \alpha_e \pi \Delta t \qquad (40)$$

where:

$$\Delta t = \frac{t_{wi} + t_{we}}{2} - t_e \qquad (41)$$

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:

$$\eta_{iz} = \frac{q - q_i}{q} = 1 - \frac{q_i}{q} \qquad (42)$$

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

From the relations (39) and (42) results:

$$q_i = \alpha_e \pi D_e \Delta t (1 - \eta_i) \qquad (43)$$

At the same time:

$$q_i = mc_p \Delta t_w = mc_p (t_{wi} - t_{we}) \qquad (44)$$

The temperature difference from the equation (45) can be written as:

$$\Delta t = \frac{t_{wi} + t_{we}}{2} - t_e = t_{wi} - \frac{\Delta t_w}{2} - t_e \qquad (45)$$

Using the relations (43) and (45) and taking into consideration that $mc_p \gg \alpha_e \pi D_e(1-\eta_{iz})/2$ results:

$$\Delta t_w = \frac{\alpha_e \pi D_e (t_{wi} - t_e)(1 - \eta_{iz})}{mc_p} \qquad (46)$$

The external surface conductance α_e depends on the difference between the pipe surface temperature t_s and outdoor air temperature t_e [22]:

$$\alpha_e = 8{,}1 + 0{,}045(t_s - t_e) \qquad (47)$$

Taking into consideration the thermal insulation efficiency, in the case of metal pipes the equation (46) becomes:

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

$$(48)$$

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

$$\frac{\Delta t_w}{\Delta t_{w0}} = \frac{[8{,}1 + 0{,}045(t_{wi} - t_e)(1 - \eta_{iz})](t_{wi} - t_e)\rho_0}{[8{,}1 + 0{,}045(t_{wi0} - t_e)(1 - \eta_{iz})](t_{wi0} - t_e)\rho} \qquad (49)$$

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

3.3 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 °C. 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 [12], can be written in simplified form:

– for η_{iz}=0: $\Delta t_w = \left(-5 + 0.32t^{1.17}\right)\cdot 10^{-3}$ (50)

– for η_{iz}=0.7: $\Delta t_w = \left(-1 + 0.05t^{1.28}\right)\cdot 10^{-3}$ (51)

Using the relations (50) and (51) 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 1 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 13.

In figure 14 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 15 the energy saving e_s at different values of the carrier temperature t_d is presented, improved thermal insulation level ($\eta_{iz} = 0.7$) 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 1. Real values of the temperature drop for qualitative control

| t_d [°C]	t_e [°C]	$\eta_{iz}=0$		$\eta_{iz}=0.7$		t_d [°C]	t_e [°C]	$\eta_{iz}=0$		$\eta_{iz}=0.7$	
		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
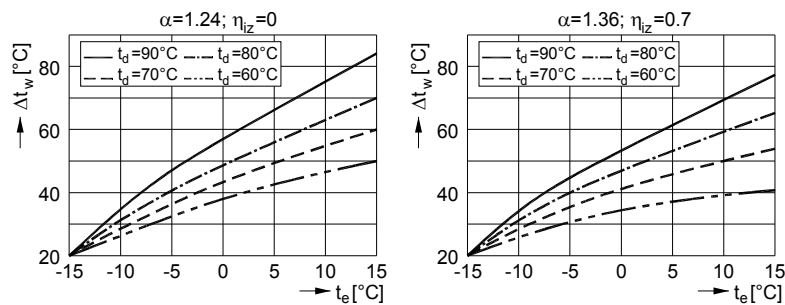
Fig. 13 Variation of the real temperature drop values for quantitative control
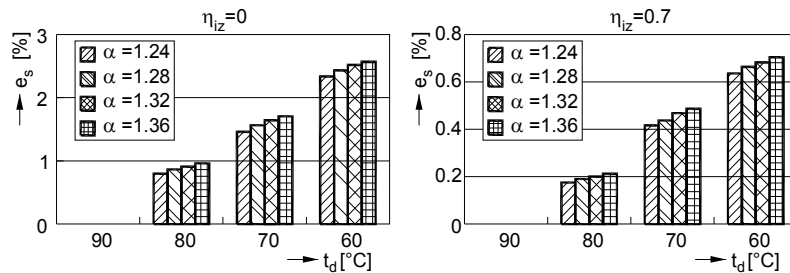

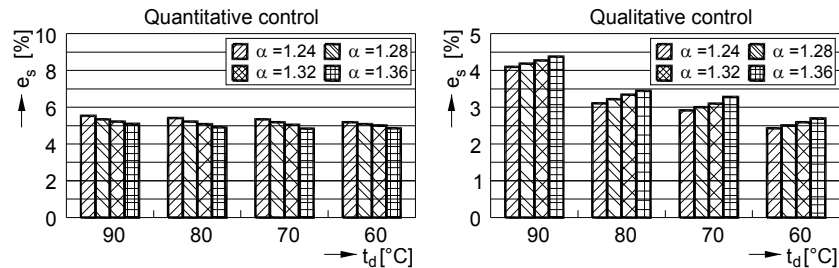Fig. 14 Energy saving in function of forward temperature

Fig. 15 Energy saving by pipe insulation in function of control system

4 Conclusions

If a central heating system is unbalanced and the real hot-water flow at the radiator is higher than the required value, the difference between the expected indoor temperature and the set-point value is independent by the indoor temperature design value. When the real flow at the radiator is lower than the required value for a higher set-point temperature, as higher will be the decrease of the expected indoor temperature. At the same time the indoor temperature is influenced by the design value of the forward temperature and temperature drop. For the same value of the temperature drop the difference between the obtained indoor temperature and the set-point value is higher at lower values of the design forward temperature. The stability of the low temperature heating system could be improved decreasing the temperature drop.

The gradient of indoor temperature variation will give the difference between the real annual energy consumption of a room and the design value. The gradient variation is independent by the indoor set-point temperature value.

To avoid and correct the negative consequences of an unbalanced operation of the heat supply system, it is recommended to upgrade the system with specialized balance and control equipments (differential pressure regulators, adjustable valves, thermostatic valves, a.s.o.).

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.

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