Environmental Engineering and Management Journal

November 2014, Vol.13, No. 11, 2789-2795
http://omicron.ch.tuiasi.ro/EEMJ/

"Gheorghe Asachi" Technical University of Iasi, Romania

HYDRAULIC BALANCING ANALYSIS OF A CENTRAL HEATING SYSTEM WITH CONSTANT SUPPLY TEMPERATURE

Sándor Hámori*, Ferenc Kalmár

University of Debrecen, Faculty of Engineering, 4028 Debrecen, Ótemető u. 2-4, Hungary

Abstract

In European countries with temperate climates, heating represents approximately 70% of the total energy consumption of a residential building. Different Directives of the European Parliament and Council prescribe the enhancement of energy efficiency and the reduction of energy demand of buildings. Consequently, the control of delivered heat is indispensable. The proper operation of the control system is possible only for balanced central heating systems. In the case of central heating systems with a constant supply temperature, the effects of mass flow deviation on the indoor temperature can be determined using only iterative methods. In this paper, the indoor temperature variation is analyzed for different mass flow deviations, and the effects of unbalancing are analyzed for a refurbished building.

Key words: heating system, hydraulic balancing, indoor temperature, iteration

Received: February, 2014; Revised final: October, 2014; Accepted: October, 2014

1. Introduction

Different Directives of the European Parliament and Council prescribe the enhancement of energy efficiency and the reduction of energy demand of buildings (EC Directive, 2010). Thermal comfort is a special issue of building services engineering because a subjective sensation must be rated with objective indexes. The comfort sensation of occupants is influenced by numerous factors (lighting, noise, temperature, health, age etc.). Thermal comfort depends particularly on: indoor air temperature, mean radiant temperature, relative humidity, relative velocity of air, thermal insulation of clothing and activity level.

From these six parameters, four can be influenced by installed heating, ventilation and air conditioning (HVAC) systems (Kalmár and Kalmár, 2010, 2011, 2012, 2013). The objective of HVAC systems design is to ensure optimal thermal conditions for the occupants to obtain maximal efficiency of the work performed or to ensure optimal conditions for regeneration.

This objective can be fulfilled only with properly balanced central heating systems (Kalmár, 2011). The aim of central heating system balancing is to ensure the designed mass flow in each heater (radiator, fan coil, floor heating circuit etc.) at the lowest possible pump rotation speed. Hydronic balancing is an essential procedure for the successful operation and optimization of a radiant floor heating system (Rhee et al., 2010).

In a central heating system, the resistances of the heating circuits are different. For example, the higher hydraulic resistance can be determined by the physically shortest circuit. The hydraulic resistance of a circuit practically means the sum of the major and the minor pressure losses.

In the case of an open system, further resistance is generated by the geodetic lift and available pressure at the consumers. Central heating systems are usually closed systems; consequently, in this case, geodetic lift and available pressure at consumers do not exist. The total pressure loss (sum of major and minor) is given by Eq. (1):

* Author to whom all correspondence should be addressed: e-mail: banyai.orsolya@law.unideb.hu, Phone: +36-52/512-700/74808

$$\Delta p' = \frac{1}{2} \cdot \rho \cdot w^2 \cdot \left(\lambda \cdot \frac{L}{d} + \sum \zeta \right) \tag{1}$$

From the equation, the pressure loss varies with the square of the flow velocity. This practically means that the pressure loss depends on the square of the mass flow. Consequently, the system characteristic is parabolic and the affinity laws are valid.

If the hydraulic resistances of the heating circuits are different, the fluid flows are prioritized in circuits with low resistance. Consequently, in circuits with low resistance, a supplementary hydraulic resistance must be built in artificially. This process practically represents the hydraulic balancing of the heating system. The balancing process is appropriate if the supplementary hydraulic resistance has the minimal required value. In this case, the pump will operate at the designed duty point.

All other cases indicate that the heating system is not properly balanced. In unbalanced heating systems, the energy use is higher than the minimum necessary value. Furthermore, there will be overheated rooms and rooms in which the indoor temperature will be lower than the set point value. Consequently, the thermal comfort conditions will be worse at a higher energy use.

In Hungary, practice has shown that in 90% of central heating systems, the measured mass flow is 150% of the designed value. According to thermal comfort theory, even in the case of properly designed and operated HVAC systems, 5% of the occupants will be dissatisfied with the obtained thermal comfort. Fanger built his PMV-PPD diagram in 1970 (Fanger, 1970).

2. Requirements of balancing

The 40/2012 (VIII.13) BM decree prescribed the balancing of central heating systems, setting of the pump duty point and the randomized control of 10% of the balancing valves setting, but it does not prescribe a requirement related to accepted mass flow deviation (Regulation 40, 2012).

Consequently, based on the experience of foreign practitioners, in Hungary, the accepted deviation of the mass flow is ±10%. There are special cases, e.g., the accepted deviation of the mass flow is established in common by the owner and the contractor.

3. Mathematical model of balancing

Under steady state conditions in a heated room, the equality of three equations is fulfilled, (Kalmár, 2011):

a) the equation of heat carrier cooling in the radiator (Eq. 2):

$$\dot{Q} = c_W \cdot \dot{m} \cdot (t_s - t_r) \tag{2}$$

b) the heat transfer equation between heat carrier and indoor air (Eq. 3):

$$\dot{Q} = A_{rad} \cdot k \cdot \Delta t_{mln} \tag{3}$$

where the mean logarithmical temperature difference Δt_{mln} is given by Eq. (4).

$$\Delta t_{mln} = \frac{t_s - t_r}{ln\left(\frac{t_s - t_i}{t_r - t_i}\right)} \tag{4}$$

c) the heat loss equation of room/building:

$$\dot{Q} = \sum A \cdot U \cdot (t_i - t_e) + \sum l \cdot \psi \cdot (t_i - t_e) + V \cdot n_a \cdot c_{p\ air} \cdot \rho_e \cdot (t_i - t_e) \tag{5}$$

Some notifications are considered:

- the radiator surface variation dependence on temperature is neglected;
- the variation of heat transfer coefficient of radiator and of building elements dependence on temperature is neglected.

Eqs. (2), (3) and (5) are influenced by three parameters:

- return temperature of the central heating system, which affects the Eq. (2) and (3);
- indoor temperature, which affects Eq. (3) and (5);
- mass flow of the heat carrier (warm water), which affects Eq. (2).

The system of Eqs. (2), (3), (5) can be solved only by using iterative methods.

Assuming a certain operation state and the design operation state of the central heating system, the equation system can be written as given by Eq. (6).

$$\frac{\dot{Q}}{\dot{Q}_0} = \frac{\dot{m}}{\dot{m}_0} \cdot \frac{(t_s - t_r)}{(t_{s0} - t_{r0})} = \frac{\dfrac{t_s - t_r}{ln\left(\frac{t_s - t_i}{t_r - t_i}\right)}}{\dfrac{t_{s0} - t_{r0}}{ln\left(\frac{t_{s0} - t_{i0}}{t_{r0} - t_{i0}}\right)}} = \frac{(t_i - t_e) \cdot \left(\sum A \cdot U + \sum l \cdot \psi + V \cdot n_a \cdot c_{p\ air} \cdot \rho_e\right)}{(t_{i0} - t_e) \cdot \left(\sum A \cdot U + \sum l \cdot \psi + V \cdot n_a \cdot c_{p\ air} \cdot \rho_{e0}\right)} \tag{6}$$

where the parameters with index "0" involve the design conditions, and the parameters without index "0" involve a certain operation state.

In the following, the effects of hydraulic balancing of a central heating system with constant supply temperature on the indoor temperature are analyzed in the case of the designed outdoor temperature. The design values of t_{s0}, t_{r0} and t_{e0} are assumed to be known. Because the supply temperature is assumed to be constant, $t_s = t_{s0}$ (in the following we use t_s), the outdoor temperature is $t_e = t_{e0}$ (in the following we use t_e), and the air density is $\rho_e = \rho_{e0}$, Eq. (6) can be written as expressed by Eq. (7):

$$\frac{\dot{Q}}{\dot{Q}_0} = \frac{\dot{m}}{\dot{m}_0} \cdot \frac{(t_s - t_r)}{(t_s - t_{r0})} = \frac{\dfrac{t_s - t_r}{ln\left(\dfrac{t_s - t_i}{t_r - t_i}\right)}}{\dfrac{t_s - t_{r0}}{ln\left(\dfrac{t_s - t_{i0}}{t_{r0} - t_{i0}}\right)}} = \frac{(t_i - t_e)}{(t_{i0} - t_e)} \tag{7}$$

For the water side, Eq. (8) can be written as:

$$\frac{\dot{Q}}{\dot{Q}_0} = \frac{\dfrac{t_s - t_r}{ln\left(\dfrac{t_s - t_i}{t_r - t_i}\right)}}{\dfrac{t_s - t_{r0}}{ln\left(\dfrac{t_s - t_{i0}}{t_{r0} - t_{i0}}\right)}} \tag{8}$$

From Eq. (8), the indoor temperature can be expressed in two steps (Eqs. 9, 10):

$$\text{1. step: } \left(\frac{t_s - t_i}{t_r - t_i}\right) = e^{\dfrac{t_s - t_r}{\dfrac{t_s - t_{r0}}{ln\left(\dfrac{t_s - t_{i0}}{t_{r0} - t_{i0}}\right)} \cdot \dfrac{\dot{Q}}{\dot{Q}_0}}} \tag{9}$$

$$\text{2. step: } t_i = \frac{t_r \cdot e^{\dfrac{t_s - t_r}{\dfrac{\dot{Q}}{\dot{Q}_0} \cdot \dfrac{t_s - t_{r0}}{ln\left(\dfrac{t_s - t_{i0}}{t_{r0} - t_{i0}}\right)}}} - t_s}{e^{\dfrac{t_s - t_r}{\dfrac{\dot{Q}}{\dot{Q}_0} \cdot \dfrac{t_s - t_{r0}}{ln\left(\dfrac{t_s - t_{i0}}{t_{r0} - t_{i0}}\right)}}} - 1} \tag{10}$$

Based on Eq. (10), each return temperature (t_r) fits an indoor temperature (t_i). Each calculated indoor temperature value involves a Q/Q_0 ratio. If the difference between this Q/Q_0 ratio and the Q/Q_0 ratio determined from Eq. (8) is lower than 10^{-5}, then the iterative calculation process is solved. The iterative calculation process was performed for radiator exponents between 1.2 – 1.4, with a step of 0.01.

The effect of mass flow deviation was analyzed for mass flow ratios between: 0.25 – 1.5 with a step of 0.05. For mass flow ratios higher than 1.5, the step of the ratio increment was increased because these deviations are rare in practice. The analysis was performed even for extremely high mass flow ratios to verify the theoretical validity of the mathematical model.

4. Effects of balancing of the constant supply temperature central heating system on the indoor temperature

The Technical Report (1998) defines three comfort categories for buildings. For these A, B and C categories, the operative temperature is prescribed with a certain accepted deviation. The operative temperature depends on the indoor air temperature and the mean radiant temperature of enclosure. Practice has shown that in the case of air velocities lower than 0.2 m/s and for differences between the indoor air temperature and the mean radiant temperature lower than 4^0K, the operative temperature can be assumed as the mean of these temperatures. In that special case when the air temperature and the mean radiant temperature are equal, the operative temperature is equal to the air temperature. To simplify the calculations, the analysis was performed for this special situation.

Performing the iterative calculations, it can be stated that increasing the mass flow results in the indoor temperature converging to a certain value. Consequently, in the case of a constant supply temperature central heating system, under steady state conditions, at the designed outdoor temperature, the maximum value of the indoor temperature does not depends on the heat demand of the room. The maximum of the indoor temperature, which is obtained at infinitely high mass flow, is influenced by the indoor set point temperature, by the temperature drop of the heating system and the radiator exponent.

For example, in case of a central heating system with a temperature drop of 80/60°C, outdoor design temperature of -15°C and indoor set point temperature of 22°C, for values of the radiator exponents between 1.2-1.4, the maximum value of indoor temperature is given by Eq. (11).

$$t_{i\,max} = 2.839\,ln(n) + 26.695 \tag{11}$$

For the accepted deviations of the operative temperature given by CR 1752, a certain deviation of the mass flow can be identified. For example, in the case of the “A” comfort category, for the ±1 °C accepted indoor temperature deviation, if the radiator exponent is between 1.20-1.25, the corresponding acceptable mass flow deviation is -15% - +25%. If the radiator exponent is between 1.25-1.4, the accepted mass flow deviation is -15% - +20%. This means that the balancing process must be performed with an accuracy of 85%-125% and 85%-120% for the radiator exponent of 1.20-1.25 and 1.25-1.4, respectively (Table 1).

In the case of the “B” comfort category of a building, the accepted indoor temperature deviation is ±2 °C. For this deviation, the accepted mass flow ratio interval can be determined (Table 2). In this case, the minimum accepted balancing is 75% and the maximum is 150%. In the case of the “C” comfort category of a building, the accepted indoor temperature deviation is ±3 ˚C. For this deviation, the accepted mass flow ratio interval can be determined (Table 3). In this case, the minimum accepted balancing is 65% and the maximum is 200%. The required mass flow balancing depending on the building comfort category and radiator exponent is presented in Table 4.

Solving the balancing equations, it can be stated the variation of the indoor temperature is concave. In practice, the infinitely high mass flow is absurd, and even a mass flow ratio of 100 is not realistic, but from a theoretical point of view, it is interesting to observe the effects of such a high mass flow deviation. Performing the calculation for different mass deviations, it can be stated that for mass flow ratios tending to infinity, the deviation of indoor temperature exceeds only by 0.6 ˚C the deviation of the indoor temperature obtained for a mass flow deviation of 10.

Taking as a reference the indoor temperature deviation obtained for an infinitely high mass flow ratio, the differences of indoor temperature deviations obtained for other mass flow ratios, depending on the radiator exponents, are presented in Table 5.

Table 1. Indoor temperatures depending on the mass flow deviation and the radiator exponent ("A" comfort category)

rad.	***Mass flow deviation***						
exp.	***0.85***	***0.90***	***1.00***	***1.10***	***1.15***	***1.20***	***1.25***
1.20	21.158	21.467	22	22.442	22.637	22.816	22.982
1.21	21.155	21.465	22	22.444	22.639	22.819	22.985
1.22	21.152	21.463	22	22.446	22.641	22.822	22.989
1.23	21.149	21.461	22	22.447	22.644	22.825	22.993
1.24	21.146	21.459	22	22.449	22.646	22.828	22.997
1.25	21.143	21.457	22	22.450	22.649	22.831	23.000
1.26	21.140	21.456	22	22.452	22.651	22.834	23.004
1.27	21.137	21.454	22	22.454	22.653	22.837	23.007
1.28	21.135	21.452	22	22.455	22.655	22.840	23.011
1.29	21.132	21.450	22	22.457	22.658	22.843	23.015
1.30	21.129	21.448	22	22.458	22.660	22.846	23.018
1.31	21.126	21.447	22	22.460	22.662	22.849	23.022
1.32	21.124	21.445	22	22.461	22.664	22.852	23.025
1.33	21.121	21.443	22	22.463	22.666	22.854	23.028
1.34	21.118	21.441	22	22.464	22.669	22.857	23.032
1.35	21.116	21.440	22	22.466	22.671	22.860	23.035
1.36	21.113	21.438	22	22.467	22.673	22.863	23.038
1.37	21.110	21.436	22	22.469	22.675	22.865	23.042
1.38	21.108	21.435	22	22.470	22.677	22.868	23.045
1.39	21.105	21.433	22	22.471	22.679	22.871	23.048
1.40	21.103	21.432	22	22.473	22.681	22.873	23.051

Table 2. Indoor temperatures depending on the mass flow deviation and the radiator exponent ("B" comfort category)

rad.	***Mass flow deviation***						
exp.	***0.75***	***0.80***	***1.00***	***1.25***	***1.30***	***1.45***	***1.50***
1.2	20.429	20.814	22	22.982	23.135	23.536	23.653
1.21	20.424	20.810	22	22.985	23.140	23.543	23.660
1.22	20.418	20.806	22	22.989	23.144	23.549	23.666
1.23	20.413	20.802	22	22.993	23.149	23.554	23.673
1.24	20.408	20.798	22	22.997	23.153	23.560	23.679
1.25	20.403	20.794	22	23.000	23.157	23.566	23.685
1.26	20.397	20.790	22	23.004	23.161	23.572	23.692
1.27	20.392	20.786	22	23.007	23.165	23.578	23.698
1.28	20.387	20.782	22	23.011	23.170	23.583	23.704
1.29	20.382	20.778	22	23.015	23.174	23.589	23.710
1.3	20.377	20.774	22	23.018	23.178	23.595	23.716
1.31	20.372	20.770	22	23.022	23.182	23.600	23.722
1.32	20.367	20.767	22	23.025	23.186	23.606	23.728
1.33	20.363	20.763	22	23.028	23.190	23.611	23.734
1.34	20.358	20.759	22	23.032	23.194	23.617	23.740
1.35	20.353	20.756	22	23.035	23.198	23.622	23.746
1.36	20.348	20.752	22	23.038	23.202	23.627	23.752
1.37	20.344	20.748	22	23.042	23.205	23.633	23.757
1.38	20.339	20.745	22	23.045	23.209	23.638	23.763
1.39	20.335	20.741	22	23.048	23.213	23.643	23.769
1.4	20.330	20.738	22	23.051	23.217	23.648	23.774

Table 3. Indoor temperatures depending on the mass flow deviation and the radiator exponent ("C" comfort category)

rad. exp.	*Mass flow deviation*				
	0.65	*0.70*	*1.00*	*1.75*	*2.0*
1.2	19.503	19.995	22	24.141	24.511
1.21	19.495	19.989	22	24.150	24.522
1.22	19.487	19.982	22	24.158	24.532
1.23	19.479	19.975	22	24.167	24.542
1.24	19.471	19.969	22	24.175	24.552
1.25	19.463	19.962	22	24.184	24.562
1.26	19.455	19.956	22	24.192	24.572
1.27	19.448	19.949	22	24.200	24.582
1.28	19.440	19.943	22	24.208	24.591
1.29	19.432	19.937	22	24.216	24.601
1.3	19.425	19.931	22	24.224	24.611
1.31	19.417	19.925	22	24.232	24.620
1.32	19.410	19.919	22	24.240	24.629
1.33	19.403	19.913	22	24.248	24.639
1.34	19.395	19.907	22	24.256	24.648
1.35	19.388	19.901	22	24.263	24.657
1.36	19.381	19.895	22	24.271	24.666
1.37	19.374	19.889	22	24.279	24.675
1.38	19.367	19.883	22	24.286	24.684
1.39	19.360	19.878	22	24.294	24.693
1.4	19.353	19.872	22	24.301	24.702

Table 4. Required balancing

CR 1752 kat.	*1.2 ≤ n < 1.25*		*1.25 ≤ n ≤ 1.40*	
	Min. [%]	*Max. [%]*	*Min. [%]*	*Max. [%]*
A	80	125	80	120
B	75	150	75	150
C	65	200	65	200

Table 5. Indoor temperature deviations with reference to an infinite mass flow ratio

n	*m/m_0*							
	10	*20*	*50*	*100*	*1 000*	*10 000*	*100 000*	∞
1.2	0.55498	0.27842	0.11159	0.05583	0.00559	0.00056	0.00006	0
1.21	0.55788	0.27988	0.11218	0.05613	0.00561	0.00056	0.00006	0
1.22	0.56076	0.28134	0.11276	0.05642	0.00564	0.00056	0.00006	0
1.23	0.56362	0.28278	0.11335	0.05671	0.00567	0.00057	0.00006	0
1.24	0.56646	0.28422	0.11393	0.05700	0.00570	0.00057	0.00006	0
1.25	0.56928	0.28565	0.11450	0.05729	0.00573	0.00057	0.00006	0
1.26	0.57208	0.28706	0.11507	0.05757	0.00576	0.00058	0.00006	0
1.27	0.57486	0.28847	0.11563	0.05786	0.00579	0.00058	0.00006	0
1.28	0.57762	0.28986	0.11620	0.05814	0.00582	0.00058	0.00006	0
1.29	0.58036	0.29125	0.11675	0.05842	0.00584	0.00058	0.00006	0
1.3	0.58308	0.29262	0.11731	0.05870	0.00587	0.00059	0.00006	0
1.31	0.58579	0.29399	0.11786	0.05897	0.00590	0.00059	0.00006	0
1.32	0.58847	0.29535	0.11841	0.05925	0.00593	0.00059	0.00006	0
1.33	0.59113	0.29669	0.11895	0.05952	0.00596	0.00060	0.00006	0
1.34	0.59378	0.29803	0.11949	0.05979	0.00598	0.00060	0.00006	0
1.35	0.59640	0.29936	0.12002	0.06006	0.00601	0.00060	0.00006	0
1.36	0.59901	0.30068	0.12055	0.06032	0.00604	0.00060	0.00006	0
1.37	0.60160	0.30199	0.12108	0.06059	0.00606	0.00061	0.00006	0
1.38	0.60418	0.30329	0.12161	0.06085	0.00609	0.00061	0.00006	0
1.39	0.60673	0.30458	0.12212	0.06111	0.00611	0.00061	0.00006	0
1.4	0.60927	0.30587	0.12264	0.06137	0.00614	0.00061	0.00006	0

In practice, the accuracy of "traditional" air temperature sensors is ±0.5 ˚C; consequently, the 0.6 ˚C temperature deviation is barely detectable. At the same time, the infinitely high mass flow theoretically means an infinitely high amount of energy consumption.

5. Case study

Using the mathematical model presented above, we analyze the effects of hydraulic balancing in the case of a building built in the year 2000. The thermal properties of the building envelope and the quality of the HVAC systems correspond to the requirements of that period. Assuming the refurbishment of the analyzed building envelope in accordance to the requirements of TNM 7/2006 Regulation (Regulation 7, 2006) and the requirements expected in year 2020, the effects of unbalancing is analyzed considering the original central heating system.

The net floor area of the analyzed building is 138 m^2, the overall heat transfer coefficient of external wall is 0.58 W/m^2K, the heat transfer coefficient of the windows and doors is 2.0 W/m^2K, the overall heat transfer coefficient of the flat roof is 0.29 W/m^2K and the fictive linear heat transfer coefficient of the floor placed directly on the ground is 0.95 W/mK. The air change rate (ACH) was assumed to be 0.8 h^{-1}. The obtained heat demand of the building is 13.38 kW. Refurbishing the building according to TNM Regulation 7/2006 (R1) using EPS polystyrene, the thickness of the additional insulation layer in the case of the external wall is 4 cm; in the case of the flat roof, the additional thickness is 20 cm.

The new windows and doors have a heat transfer coefficient of 1.6 W/m^2K. Because the building air tightness is improved, the ACH is 0.5 h^{-1}. In this case, the heat loss of the refurbished building will be Q_{2006} = 10.1 kW. Assuming the expected requirements in year 2020 (R2), the thickness of the additional insulation layer in the case of an external wall should be increased further to 16 cm, and in the case of a flat roof, the further additional insulation layer thickness is 10 cm. The new windows and doors have a heat transfer coefficient of 1.0 W/m^2K. The heat loss caused by infiltration is reduced to ACH = 0.1 h^{-1}. The heat loss of the refurbished building will be Q_{2020}=5.04 kW.

The heat transfer coefficients of the external building elements are presented in Table 6. The floor structure was not changed after refurbishments.

Table 6. Heat transfer coefficients of the building elements

Year	U_{wall}, [W/m^2K]	U_{win}, [W/m^2K]	U_{roof}, [W/m^2K]	Ψ_{floor}, [W/mK]
2001	0.58	2.0	0.29	0.95
2006 (R1)	0.37	1.6	0.22	0.95
2020 (R2)	0.15	1.0	0.15	0.95

The heat loss values of the building in its original state and after refurbishments are presented in Fig. 1.

After the first refurbishment, according to the TNM 7/2006 Regulation requirements for refurbishment, the heat loss of the analyzed building decreased by 25%. Assuming a refurbishment according to the expected requirements in 2020, the heat loss of the building is expected to decrease by 62%. In this analysis, the variation of the physical parameters of the insulation materials was neglected (Lakatos and Kalmár, 2013a, 2013b; Lakatos et al., 2013a, 2013b; Lakatos, 2014).

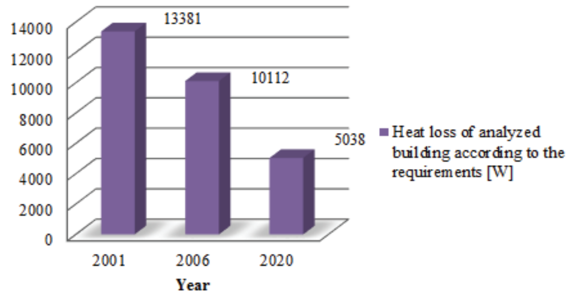

Fig. 1. Heat loss of the analyzed building

If the central heating system remains unchanged, after building refurbishments because the heat demand decreased considerably, this system will be over dimensioned. For example, in the living room, having a net floor area of 51.6 m^2, the original heat loss was 5441 W; after refurbishment R1, the heat loss decreased by 25%, and after refurbishment R2, the heat loss is expected to be reduced by 62%. If the pump is not changed or the duty point is not set to the required flow and head, assuming that before refurbishment the heating system was properly balanced, because of the heat demand reduction, the original mass flow will be higher by 33% in the case of refurbishment R1 and by 138% in the case of refurbishment R2. Consequently, 22 higher indoor temperatures will be obtained. These values are presented in Table 7.

Table 7. Indoor temperatures after building envelope refurbishments

rad. exp.	*Refurbishment year*	
	2006	*2020*
1.2	23.25479352	24.95730223
1.21	23.25969156	24.96963828
1.22	23.2645394	24.98186097
1.23	23.26934427	24.99398829
1.24	23.27410676	25.00601296
1.25	23.2788274	25.01793624
1.26	23.28350674	25.02975938
1.27	23.28814531	25.04148989
1.28	23.29274363	25.053116
1.29	23.29730223	25.06463989
1.3	23.3018216	25.07607437
1.31	23.30630224	25.08741459
1.32	23.31074465	25.09866167
1.33	23.3151493	25.10981672
1.34	23.31951667	25.12088083
1.35	23.32384723	25.13185507
1.36	23.32814142	25.1427405
1.37	23.33239971	25.15353817
1.38	23.33662254	25.16424909
1.39	23.34081033	25.17487429
1.4	23.34496353	25.18541713

6. Conclusions

Balancing of central heating systems is a fundamental requirement to minimize the energy as much as possible. However, the majority of heating systems are unbalanced; consequently, the operation is wasteful from the energy point of view. Furthermore, in the case of unbalanced heating systems, the indoor air temperature will be higher or lower than the set point value; consequently, comfort problems may appear. To ensure the requirements of CR 1752, related to different comfort categories, the maximum admissible mass flow deviation was determined. In the case of building refurbishment, it is of high importance to adjust the central heating system to the new energy requirements of the building; otherwise, overheating, unbalancing and discomfort problems may appear.

Acknowledgments
The work is supported by the TÁMOP-4.2.2.A-11/1/KONV-2012-0041 project. The project is co-financed by the European Union and the European Social Fund.

Nomenclature

ρ – density [kg/m^3]
w – average velocity of the fluid [m/s]
λ – friction coefficient [-]
L – length of the pipe [m]
d – hydraulic diameter [m]
ζ – minor loss coefficient [-]
$\Delta p'$ – pressure loss [Pa]
t_s – supply temperature [°C]
t_r – return temperature [°C]
t_i – internal temperature [°C]
t_e – external temperature [°C]
U – heat transmission coefficient [W/m^{2}K]
n – radiator exponent [-]
c_w – specific heat capacity of water 4,187 [kJ/kgK]
c_{pair} – isobaric specific heat capacity of air 1004 [J/kgK]
m – mass flow [kg/s]
A_{rad} – radiator surface [m^2]
Δt_{mln} – mean logarithmical temperature [°C]
ψ – fictive linear heat transfer coefficient [W/mK]
l – length of floor placed directly on the ground [m]
n_a – air change rate [1/s]
PMV – Predicted Mean Vote
PPD – Predicted Percentage Dissatisfied

References

EC Directive, (2010), Directive 2010/31/EU of the European Parliament and of the Council of 19 May 2010 on the energy performance of buildings (recast), *Official Journal of the European Communities*, **L 153**/13, 18.6.2010, Brussels.

Fanger P.O., (1970), *Thermal Comfort. Analysis and Applications in Environmental Engineering*, Danish Technical Press, Copenghagen.

Kalmár T., Kalmár F., (2010), Comfort and energy analysis of heating up, *International Review of Applied Sciences and Engineering*, **1**, 35-43.

Kalmár F., Kalmár T., (2011), Analysis of floor and ceiling heating with intermittent operation, *Environmntal Engineering and Management Journal*, **10**, 1243-1248.

Kalmár F., Kalmár T., (2012), Interrelation between room geometry and mean radiant temperature, *Energy and Buildings*, **55**, 414-421.

Kalmár F., Kalmár T., (2013), Alternative personalized ventilation, *Energy and Buildings*, **65**, 37-44.

Kalmár F., (2011), Energy conscious heating, Akadémiai Kiadó, 2011, 62-79.

Lakatos Á., Kalmár F., (2013a), Analysis of water sorption and thermal conductivity of polystyrene insulation materials, *Building Service Engineering Research and Technology*, **34**, 407-416.

Lakatos Á., Kalmár F., (2013b), Investigation of thickness and density dependence of thermal conductivity of Expanded Polystyrene insulation materials, *Materials and Structures* **46**, 1101-1105.

Lakatos Á., Szigeti S., Kalmár F., (2013a), Measurement of the water uptaking capability of a thermal insulating paint, *International Review of Applied Sciences and Engineering*, **4**, 157-161.

Lakatos Á., Csáky I., Kalmár F., (2013b), Thermal conductivity measurements with different methods: a procedure for the estimation of the retardation time, *Materials and Structures*, DOI 10.1617/s11527-013-0238-7.

Lakatos Á., (2014), Comparison of the thermal properties of different insulating materials, *Advanced Materials Research*, **899**, 381-386.

Rhee K.N., Ryu S.R., Yeo M.S., Kim K.W., (2010), Simulation study on hydronic balancing to improve individual room control for radiant floor heating system, *Building Services Research and Technology*, **31**, 57-73.

Regulation 7, (2006), Regulation 7/2006 TNM on the determination of energy performance of buildings, (V. 24.) (in Hungarian), Minister Without Portfolio, Budapest.

Regulation 40, (2012), Regulation 40/2012 on the determination of energy performance of buildings (in Hungarian), Ministry of Interior, Budapest, Hungary.

Technical Report, (1998), Ventilation for buildings - Design criteria for the indoor environment, Technical Report CR 1752, European Committee for Standardization, Brussels.