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STUDY CONCERNING THE OPERATING PARAMETERS OF CENTRAL HEATING PLANTS IN CONDENSATION REGIME

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Abstract: *A method, of reducing gaseous fuel consumption used in heating systems, is by using condensing boilers which have a higher heating efficiency, due to the condensations of vapors from the gases that result from the combustion process. Cooling the gases under the saturation temperatures, implies the use of heating water at low temperatures: flow/return 60/40 °C, which leads to oversized heating elements thus raising the costs of investment. Therefore it is reasonable to design heating elements at temperatures of 80/60 °C flow/return. The aim of this paper is to determine the average temperature of the heating fluid depending on the exterior temperatures and to identify the periods when the condensation of vapors occurs, resulting in very high efficiency of the condensation boilers.*

Key words*: condensing boilers, vapor condensation temperature, flow/return temperature, heating elements.*

1. INTRODUCTION

The reduction of energy consumption in buildings consists in reducing the specific losses of the building elements by improving the thermal insulation of the envelope, comprised of walls and windows.

Another approach to reducing the energy consumption is through the use of condensing boilers that are known for their low fuel consumption. The use of condensing boilers involves relatively low temperatures of the heat transfer fluid (**HTF**), which must be below the vapor condensation temperature from the combustion gases.

This study investigates the temperature of the **HTF** of a heating system, in order to identify the periods during which condensation occurs and the flow/return temperatures at which de heating elements must be adopted.

2. CONDESING BOILERS

Condensing boilers are characterized by the partial condensation - possibly quasi-total condensation - of the vapors from the combustion gases resulting in the release of latent heat, leading to significantly increased thermal efficiency in comparison to the classical boiler. This is due to the difference between the higher and lower calorific values of the fuel, which is about 11 % in the case of natural gaseous fuels.

The condensation of the vapors takes place below the saturation temperature, which is between: 55.4 – 56.8 °C, in the case of natural gaseous fuels. Consequently, the temperature of the **HTF**, in the return pipe of the central heating system, must be below the condensation temperature of the vapors.

Table 1 presents the variation of the thermal efficiency of the condensing boilers depending on temperature of the **HTF**.

Table 1

Thermal efficiency depending on HTF temp. [1]

Flow/ Return Temp. [°C]	**Thermal efficiency [%] related to**	
	the lower calorific values	**the higher calorific values**
50/30	105.5 – 107.8	94.95 – 97.1
60/40	97.6 – 101.3	87.8 – 91.2
70/50	96.2 – 96.8	86.6 – 87.1
80/60	95.7 – 96.3	86.2 – 86.7

Consequently, in order to cover the heat demand - calculated according to the conventional outdoor calculation temperature - specific to each climatic zone, the heating elements are sized for flow/return temperatures of: 80/60 °C. This thermal regime prevents the condensation of vapors. Considering that the actual heat demand is reduced due to the increase of the outside temperature, the flow/return temperatures of the **HTF** decreases.

The question arises as to what external temperature value, the heat demand may be covered by the heating elements operating at low flow/return temperatures, allowing the condensation of vapors, resulting in high thermal efficiency of the boiler.

3. CENTRAL HEATING SYSTEM

The estimation of the heat demand for each chamber: Q_{dem} is calculated in accordance with the ref. [2], by determining the thermal losses for each chamber of the building under conventional calculation conditions, based on the general relation:

$$Q_{dem} = Q_T \cdot \left(1 + \frac{A_c + A_o}{100}\right) + Q_a \text{, [W] (1)}$$

where: Q_T - the heat flow lost through global heat transfer, in [W];

Q_a - the heat flow required for the heating of fresh air, in [W];

A_c - the addition coefficient that compensates the effect of cold surfaces, in [%];

A_o - the addition coefficient that takes into account the orientation of the building, in [%].

A_c can be determined using the diagram from ref. [2] based on the average heat transfer resistance of the room, determined with the equation:

$$R_{avg} = \frac{A_T \cdot (t_{in} - t_{out}) \cdot C_M}{Q_T} \text{, [m}^2\text{·K/W] (2)}$$

where: A_T - the total surface area of the chamber, in [m^2].

C_M - the correction coefficient that takes into account the specific mass of the building, [adim];

t_{in} - the conventional indoor temperature, in [°C], given by ref. [3];

t_{out} - the temperature outside of the considered chamber, in [°C], which is either the conventional outdoor calculation temperature: t_{out} adopted based on the climatic zones of Romania or the conventional indoor room temperature of the adjoining rooms, $t_{out,in}$ according to ref. [3].

Based on the values given by the ref. [2] the mathematical equation of the variation of A_c with R_{avg} was obtained using the Origin-Lab software:

$$A_c = 4{,}47425 + \frac{5{,}06665}{R_{avg} - 0{,}18242} \text{ . [\%] (3)}$$

The proposed mathematical equation has a R-square value of: R^2 = 0.99695, indicating the accuracy of the proposed relation.

3.1 Thermal flux losses through global heat transfer

For each chamber, the heat losses were determined as the sum of the heat losses, through global heat transfer, to the exterior environment and the heat losses to the soil:

$$Q_T = \sum Q_{T,BE} + Q_S \text{ . [W] (4)}$$

The heat flow transmitted through each building element is determined based on the equation:

$$Q_{T,BE} = C_M \cdot m \cdot A \cdot \frac{t_{in} - t_{out}}{R'} \text{, [W] (5)}$$

where: m - the coefficient of thermal mass, [adim], calculated based on the thermal inertia of the building element;

A - the surface area of each building element, in [m^2];

R' - the corrected thermal resistance of each considered building element, in [m^2·K/W].

The heat flow loss to the soil is determined as the sum of: the flow loss to the soil layer between the floor and the depth of 7 m or the groundwater layer and the flow yielded through the soil to the surrounding environment or the adjoining room:

$$Q_S = A_F \cdot \frac{t_{in} - t_S}{R_S} + C_M \cdot \frac{m_S}{n_S} \cdot \frac{t_{in} - t_{out}}{R_{SC}} \cdot A_{SC,out} + $$
$$+ C_M \cdot \frac{1}{n_S} \cdot \frac{t_{in} - t_{out,in}}{R_{SC}} \cdot A_{SC,in}, \quad [W] \quad (6)$$

where: A_F - the floor and the walls area below ground level, [m^2];

$A_{SC,\ out}$ - the area of the stripe along the outer contour of the room, in [m^2];

$A_{SC,\ in}$ - the area of the stripe along the contour to adjacent rooms, in [m^2];

t_S - the temperature of the soil at 7 m depth or the temperature of the groundwater, in [°C], depending on climatic zone;

R_S - the specific thermal resistance of the floor and a layer of the soil up to 7 m depth or groundwater layer, in [$m^2 \cdot K/W$];

R_{SC} - the thermal resistance of the stripe area along the outer contour of the room or the strip area along the contour to adjacent rooms, in [$m^2 \cdot K/W$];

m_S - the thermal mass coefficient of the soil, [adim];

n_S - the correction coefficient that takes into account the thermal conductivity of the soil, [adim].

Thus the thermal flux lost through global heat transfer is determined based on the following factors, according to the general relation:

$$Q_T = \begin{cases} \underbrace{C_M, m, m_S, n_S}_{\text{correction coefficient}}; \\ \underbrace{A, A_T, A_F, A_{SC,out}, A_{SC,in}}_{\text{chamber dimensions}}; \\ \underbrace{R', R_F, R_{SC}}_{\text{thermal resistance}}; \\ \underbrace{t_{in} - t_{out}}_{\text{temperature difference}}; \end{cases} ; [W] \quad (7)$$

3.2 Heat flow required for the heating of fresh air

The heat flow required for the heating of fresh air is determined as the maximum value between: Q_{a1} - the thermal load required for the heating of the fresh air that assures the physiological comfort conditions, and Q_{a2} - the thermal load required for the heating of the air penetrated through the imperfections of the window system, in [W], according to the relations:

$$Q_{a1} = \left[n_{ah} \cdot C_M \cdot V \cdot \rho \cdot c_p \cdot (t_{in} - t_{out}) + Q_D\right] \cdot \left(1 + \frac{A_c}{100}\right)$$
$$[W] \quad (8)$$

$$Q_{a2} = \left\{C_M \cdot \left[E \sum i \cdot L \cdot v^{\frac{4}{3}} \cdot (t_{in} - t_{out})\right] + Q_D\right\} \cdot \left(1 + \frac{A_c}{100}\right)$$
$$[W] \quad (9)$$

where: n_{ah} - the specific air flow that assures the physiological comfort conditions, according to ref. [2], in $\left[\frac{m^3/s}{m^3}\right]$;

V - the volume of the chamber, in [m^3];

ρ, c_p - the density of the air, in [kg/m^3], and the specific heat capacity of the air, in [$kJ/kg \cdot K$] respectively, at indoor temperature;

E - the correction coefficient that takes into account the height of the building, [adim];

i - the infiltration coefficient through the imperfections in the window systems, in $\left[\frac{W}{m \cdot (m/s)^{4/3} \cdot K}\right]$, according to ref. [2];

L - the perimeter of the doors and windows, in [m];

v - the conventional wind velocity, in [m/s], depending on the wind area;

Q_D - the thermal load required for the heating of the air flow resulting from the opening of doors, in [W], according to the relation:

$$Q_D = 3{,}6 \cdot A_D \cdot n \cdot (t_{in} - t_{out}) \cdot C_M \ , [W] \quad (10)$$

where: A_D - the surface area of the doors, in [m^2];

n - the number of door openings in one hour.

Thus the thermal load required for the heating of the fresh air that assures the physiological comfort conditions and for the heating of the penetrated air is determined based on the general relation:

$$Q_a = \begin{cases} \underbrace{C_M, A_c, E;}_{\text{correction coefficients}} \\ \underbrace{V, L, A_D;}_{\text{chamber dimensions, window system}} \\ \underbrace{n_{ah}, i, v, n_{ah};}_{\text{outdoor air}} \\ \underbrace{t_{in} - t_{out};}_{\text{temperature difference}} \end{cases} . \ [W] \quad (11)$$

3.3 Thermal resistance in conventional conditions

The specific thermal resistance of the building elements corresponding to the one dimensional global thermal transfer, is determined by natural convection on the inner surface: α_{in}, one dimensional conduction through the flat wall composed of homogeneous layers - characterized by the layer thickness δ and the thermal conductivity coefficient λ - and forced convection on the outer surface: α_{out}, according to the general relation:

$$R = \frac{1}{\alpha_{in}} + \sum \frac{\delta}{\lambda} + \frac{1}{\alpha_{out}} . \left[\frac{m^2 \cdot K}{W}\right] \quad (12)$$

In preliminary design calculations, the corrected thermal resistance, that takes into account the influence of thermal bridging, is used:

$$R' = \frac{1}{\alpha_{in}} + (1 - r) \cdot \sum \frac{\delta}{\lambda} + \frac{1}{\alpha_{out}} . \left[\frac{m^2 \cdot K}{W}\right] \quad (13)$$

where: r- correction coefficient according to ref. [4], [adim.], which reduces the thermal resistance of the wall system.

3.4 The heating demand under conventional conditions

Analyzing the relations for the calculation of the heat demand, it is noted that all the above-mentioned thermal flows depend on the temperature difference between the indoor and outdoor conventional calculation temperatures. The general relation of the heat demand results, by adding all the calculated heat flows:

$$Q_{dem} = \begin{Bmatrix} \underbrace{A_o, A_c;}_{\text{addition coefficients}} \\ \underbrace{C_M, m, m_s, n_s, E;}_{\text{correction coefficients}} \\ \underbrace{A, A_F, A_{SC,out}, A_{SC,in}, V, L, A_D;}_{\text{chamber dimensions, window system}} \\ \underbrace{n_{ah}, i, v, n_{ah};}_{\text{outdoor air}} \\ \underbrace{R', R_F, R_{SC};}_{\text{thermal resistance}} \end{Bmatrix} \cdot (t_{in} - t_{out})$$

[W] (14)

3.5 The sizing of the heating elements under conventional conditions

On the basis of the heat demand for each room, the heating elements are adopted according to their nominal heating capacity given in the manufacturer's technical specifications. The nominal heating capacity is corrected based on the general relation:

- steel panel radiators:

$$Q_{h,c} = \sum \phi_L \cdot L \cdot c_t \cdot c_r \cdot c_m \cdot c_h \cdot c_v , \ [W] \ (15)$$

- cast iron or aluminum radiators:

$$Q_{h,c} = \sum \phi_n \cdot N \cdot a \cdot c_t \cdot c_r \cdot c_m \cdot c_h \cdot c_v , [W] \ (16)$$

where: Φ_L - the specific nominal heating capacity per unit of length, in [W/m];
Φ_n - the specific nominal heating capacity per unit element, in [W/el];
c_r, c_m, c_h, c_v – the correction coefficients defined in [6];
c_t - the correction coefficient taking into account the difference between the nominal and effective parameters, in general: $t_{n,flow}$= 90 °C, $t_{n,return}$ = 70 °C, $t_{n,in}$ = 20 °C, calculated as:

$$c_t = \left(\frac{\frac{t_{flow} + t_{return}}{2} - t_{in}}{\frac{t_{n,flow} + t_{n,return}}{2} - t_{n,in}} \right)^n ,$$

rezulting the general relation:

$$c_t = C(t_{n,flow}, t_{n,return}, t_{n,in}) \cdot (t_{avg,HTF} - t_{in})^n , \ (17)$$

where: n - the thermal exponent depending on the type and size of the radiator: n = 1,29 ÷ 1,36;

$t_{avg,HTF}$ - the average temperature of the HTF, in [°C].

Thus, the effective heating capacity of the heating element, depends on the specific heating capacity, according to the length or number of elements, correction coefficients – parameters that do not dependent on the temperature of the **HTF**, and the average temperature of the **HTF**, according to the general relation:

$$Q_{h,c} = \left\{ \begin{array}{l} \underbrace{\phi_L \cdot L \text{ or } \phi_N \cdot N}_{\text{specific heating capacity}} \\ \underbrace{t_{n,flow}, t_{n,return}, t_{n,in}}_{\substack{\text{correction: nominal} \\ \text{and effective parameters}}} \\ \underbrace{c_r, c_m, c_h, c_v}_{\text{correction coefficients}} \end{array} \right\} \cdot \left(t_{avg,HFT} - t_{in}\right)^n. \quad [W] \quad (18)$$

It is obvious that the heating elements are selected based on the heating demand determined for the conventional exterior temperature, so that: $Q_{h,c} \geq Q_{dem}$.

Consequently, the real case of a one-family villa, located in the four climatic zones, was considered. The villa is equipped with a central heating system with steel panel radiators that are designed for flow/return temperatures of: 80/60 [°C].

The calculation of the heating demand was carried out depending on: the thermal resistances of the building elements, the structure and quality of the selected building materials and the heating demand for the infiltrated and fresh air.

Based on the heating demand for each room, the steel panel radiators have been adopted, thus resulting the effective thermal power of the heating system, corresponding to each climatic zone. The heating demand and the nominal heating capacity of the steel panel radiators for each climatic zone are presented in table 2.

4. THE OPERATING OF THE HEATING SYSTEM UNDER REAL CONDITIONS

4.1 The real heating demand

Analyzing the thermal characteristics of the construction, according to rel. (14), it is observed that the actual heating demand for each room varies depending on the real outside temperature: $t_{out,R}$ and the actual wind speed: v_R, according to the general relation:

$$Q_{dem,R} = \left\{ \begin{array}{l} \underbrace{A_o, A_c;}_{\text{addition coefficients}} \\ \underbrace{C_M, m, m_s, n_s, E;}_{\text{correction coefficients}} \\ \underbrace{A, A_F, A_{SC,out}, {}_{SC,in}, V, L, A_D;}_{\substack{\text{chamber dimensions,} \\ \text{window system}}} \\ \underbrace{n_{ah}, i, v_R, n_{ah};}_{\text{outdoor air}} \\ \underbrace{R'_R, R_F, R_{SC};}_{\text{thermal resistance}} \end{array} \right\} \cdot \left(t_{in} - t_{out,R}\right) \quad [W] \quad (19)$$

The real wind velocity influences the thermal resistance of the building elements - by forced convection on the outer surface, and the flow of fresh air penetrated due to the imperfections of the window system, which implies the recalculation of the flow Q_{a2}, based on the relation (9).

Table 2

The heating demand in conventional parameters and the heating elements

Chamber name, no., type and size of radiators	Climatic zone I Constanța t_{out} = -12°C			Climatic zone II București t_{out} = -15°C			Climatic zone III Cluj-Napoca t_{out}=-18°C			Climatic zone IV Brașov t_{out} = -21°C		
	Q_{dem}	L	$Q_{h,l}$	Q_{dem}	L	$Q_{h,l}$	Q_{dem}	L	$Q_{h,l}$	Q_{dem}	L	$Q_{h,l}$
	W	mm	W	W	mm	W	W	mm	W	W	mm	W
Ground floor												
Living P1: 2*22-600	2105	700	2280	2297	800	2606	2360	800	2606	2543	800	2606
Kitchen P2:1*22-600	1268	800	1373	1396	900	1545	1524	900	1545	1653	1000	1716
Bathroom P3:	530	400	617	568	400	617	606	400	617	644	500	771
Hall P4: 1*22-600	562	400	593	613	500	741	664	500	741	715	500	741
Staircase P5:1*22-600	343	400	652	375	400	652	407	400	652	439	400	652
Chamber P6: 1*22-600	1031	700	1140	1126	700	1140	1221	800	1303	1316	900	1466
Total ground floor	5838		6655	6375		7301	6782		7464	7309		7952
Floor												
Bedroom E1:2*22-600	1648	600	1955	1779	600	1955	1909	600	1955	2039	700	2280
Bedroom E2:2*22-600	1154	400	1303	1245	400	1303	1335	500	1629	1426	500	1629
Bathroom E3:	955	700	1080	1003	700	1080	1052	700	1080	1100	800	1234
Dressing E4:1*11-600	164	400	368	176	400	368	188	400	368	200	400	368
Hall E5:1*22-600	606	500	741	652	500	741	698	500	741	744	600	889
Staircase E6:1*11-600	376	500	442	403	500	442	430	500	442	457	600	530
Bedroom E7:1*22-600	967	600	977	1043	700	1140	1120	700	1140	1196	800	1303
Total floor	5871		6866	6301		7029	6731		7355	7161		8234
Total villa	11709		13521	12676		14330	13513		14819	14470		16187

The corrected thermal resistance of the building elements under real conditions changes due to the coefficient of heat transfer by forced convection: $\alpha_{out,R}$, according to the relation:

$$R'_R = \frac{1}{\alpha_{in}} + (1-r)\cdot\sum\frac{\delta}{\lambda} + \frac{1}{\alpha_{out,R}}\,. \quad [m^2\cdot K/W] \qquad (20)$$

Ref. [5] gives the values of the external superificial thermal resistance based on the actual wind velocity, which enabled the use of the OriginLab software to determine the mathematical equation describing the variation of the external superificial thermal resistance with the wind velocity, resulting the relation:

$$R_{out,R} = \frac{1}{\alpha_{out,R}} = 0{,}01779 + \frac{0{,}08062}{e^{-\frac{v_R}{3{,}42755}}}\,. \quad [m^2\cdot K/W] \qquad (21)$$

The proposed mathematical equation has a R-square value of: R^2 = 0.97829, indicating the accuracy of the proposed relation.

Consequently, using the real outdoor temperatures and the actual wind velocity - which determines the variation of the corrected thermal resistance of the building elements - the real heating demand can be determined with relation (19).

4.2 The average temperature of the HTF

Considering the dimensions of the steel panel radiators designed for the four climatic zones, the average temperature, as well as the flow and return temperatures of the **HTF** which will ensure the real heating requirements calculated for the actual climatic parameters, were determined.

In the case of a steady state heat transfer for different values of the real climatic parameters, the real heating demand for each room must be equal to the heat flux supplied by the heating elements: $Q_{dem,R} = Q_{h,c,R}$.

The equilibrium is reached for the same operating point of the thermal characteristics of the building and the thermal characteristics of the central heating system. For a certain real outside temperature and wind velocity there is a certain heat demand, which satisfies the conditions of thermal equilibrium thus determining a certain average temperature of the **HTF** corresponding to each room, which is calculated by solving the following equation:

$$\left\{\begin{matrix} \underbrace{\phi_L \cdot L \text{ or } \phi_N \cdot N}_{\text{specific heating load}} \\ \underbrace{C(t_{n,flow}, t_{n,return}, t_{n,in}}_{\substack{\text{correction nominal}\\ \text{and effective parameters}}} \\ \underbrace{c_r, c_m, c_h, c_v}_{\text{correction coefficients}} \end{matrix}\right\} \cdot \left(t_{avg,HFT} - t_{in}\right)^n = \left\{\begin{matrix} \underbrace{A_o, A_c;}_{\text{addition coefficients}} \\ \underbrace{C_M, m, m_s, n_s, E;}_{\text{correction coefficients}} \\ \underbrace{A, A_p, A_{bc}, V, L, A_u;}_{\substack{\text{chamber dimensions,}\\ \text{window system}}} \\ \underbrace{n_{ao}, i, v_R, n_{ah};}_{\text{outdoor air}} \\ \underbrace{R'_R, R_F, R_{SC};}_{\text{thermal resistance}} \end{matrix}\right\} \cdot \left(t_{in} - t_{out,R}\right) \qquad (22)$$

The calculation of the average temperature of the **HTF** supplied by the boiler and the flow and returns temperatures, was realized with the assumption that the flow rate of the **HTF** is constant, and is determined by the operating point of the circulation pump resulting from the intersection of the pump's characteristic and the hydraulic characteristic of the system. The average temperature of the **HTF** was determined as the weighted average value of the mean temperatures and the flow rate of the **HTF** corresponding to each heating element: m'_{HTFi}, according to the relation:

$$t_{avg,HTF} = \frac{\sum m'_{HTF\ i} * t_{avg,HTFi}}{\sum m'_{HTFi}}\ .\ [°C] \qquad (23)$$

The flow and return temperatures of the **HTF**, under real operating conditions of the heating system were similarly determined.

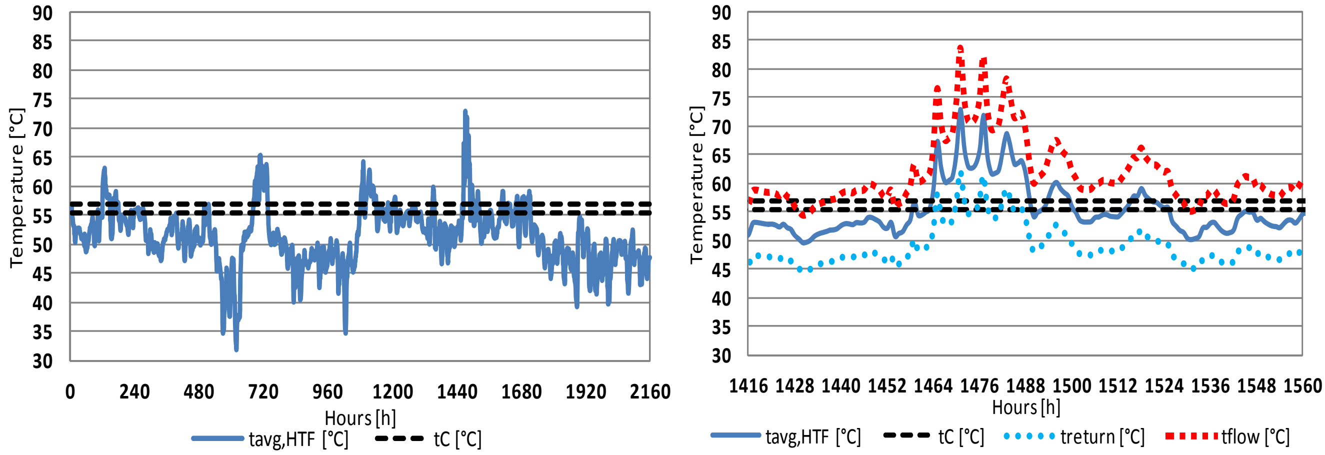
4.3 The HTF temperatures of the heating system under real conditions

The real average **HTF** temperatures were determined based on the thermal parameters given in the typical meteorological year (**TMY**), for the four climatic conditions, so that the total heating demand is covered.

Based on the climatic parameters for the winter period: December 1st - February 28th, provided by the **TMY** for cities located in the four climatic zones, the average **HTF** temperature fluctuation is compared with the condensation range and presented for each city in the figures as follows:

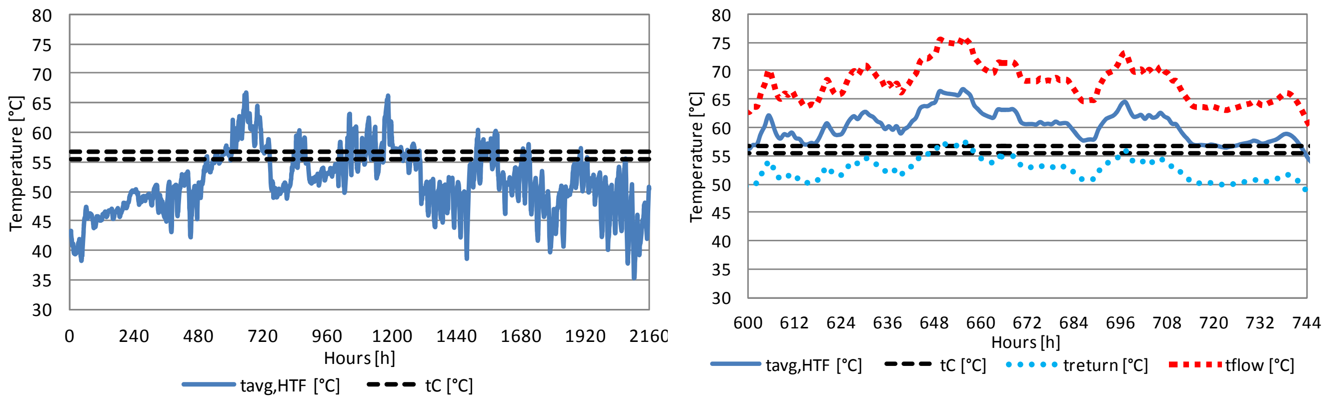
- for Constanța: climatic zone I: t_{out} = -12 °C, wind area II: v=5 m/s, shown in fig. 1;
- for Bucureşti: climatic zone II: t_{out} = -15 °C, wind area II: v=5 m/s, shown in fig. 2;
- for Cluj-Napoca: climatic zone III: t_{out} = -18 °C, wind area IV: v=4 m/s, shown in fig. 3;
- for Braşov: climatic zone IV: t_{out} = - 21 °C, wind area IV: v=4 m/s, shown in fig. 4.

For periods with maximum heat demand, both the average **HTF** temperature fluctuations as well as the variation of the flow and return temperatures of the **HTF** are presented in comparison with the condensation range in fig. 1-4.



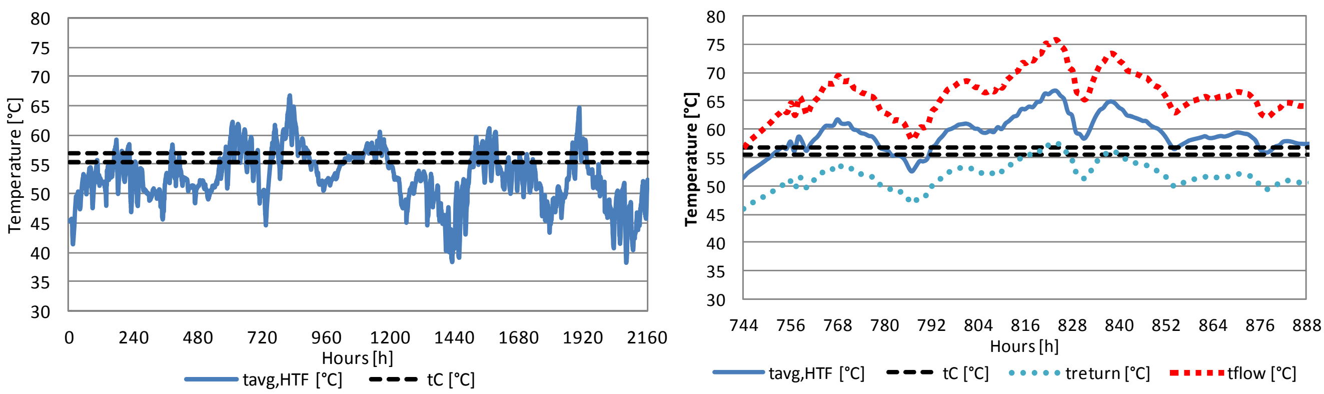
a – average **HTF** temperatures fluctuation, **b** –**HTF** temperatures for peak heat demand,

Fig. 1. HTF temperatures fluctuation during the winter Dec.1-Feb.28 – climatic zone I Constanta



a – average **HTF** temperatures fluctuation, **b** –**HTF** temperatures for peak heat demand,

Fig. 2. HTF temperatures fluctuation during the winter Dec.1-Feb.28 – climatic zone II Bucureşti

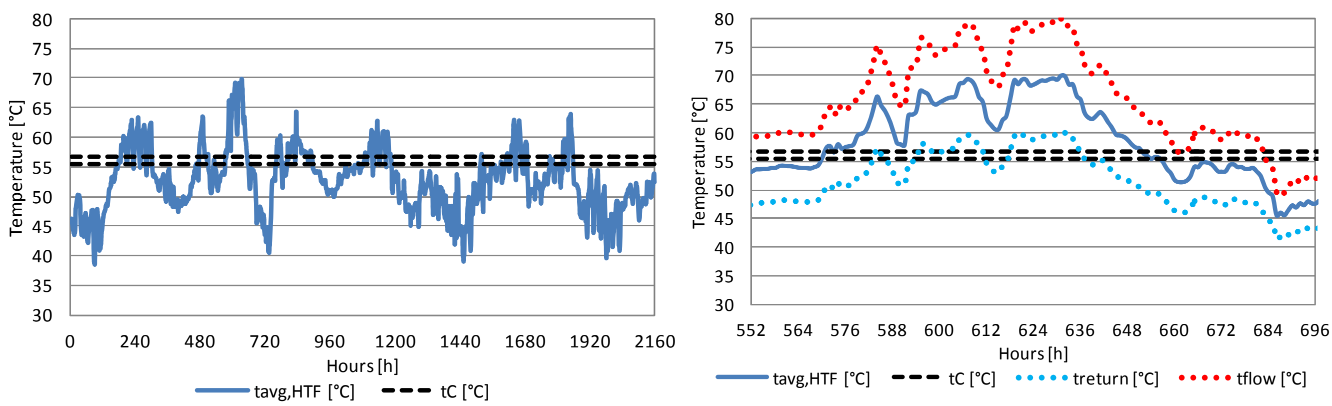


a – average **HTF** temperatures fluctuation, **b** –**HTF** temperatures for peak heat demand,

Fig. 3. HTF temperatures fluctuation during the winter Dec.1-Feb.28 – climatic zone III Cluj-Napoca

5. CONCLUSION

By analyzing the average temperature fluctuations, as well as the flow and return temperatures of the **HTF** in comparison with the vapor condensation temperature range, as shown in fig. 1-4, during the winter period, the following can be observed:



a – average **HTF** temperatures fluctuation, **b** –**HTF** temperatures for peak heat demand,

Fig. 4. HTF temperatures fluctuation during the winter Dec.1-Feb.28 – climatic zone IV Braşov

● In the chase of Constanța:

- the average **HTF** temperature exceeds the condensation range for a duration of 238-398 h;
- the return **HTF** temperature exceeds the condensation range for a duration of 7-16 h;
- generally, the periods during which the **HTF** temperature exceeds the condensation range is only a few hours during the nights;
- the number of longer periods during which the average **HTF** temperature exceeds the condensation range is relatively low: 6 continuous periods of 12-24 h, 2 continuous periods of 24-36 h, 2 continuous periods of more than 36 h.

Regarding the climatic parameters of Constanța (**TMY**), the following are specified:

- the minimum outdoor temperature: -11.90 °C;
- the wind velocity:

Month	**Wind velocity**		
	v> 5 m/s	v> 8 m/s	v> 10 m/s
	Duration [h]		
December	237	113	30
January	190	113	40
February	148	51	9

The peak values of the real heat demand are determined mainly by the wind velocity rather than the outdoor temperatures ($t_{out,\ R}$=-4.2 ÷ -6.1 °C, v_R = 12 m/s).

● In the chase of Bucureşti:

- the average **HTF** temperature exceeds the condensation range for a duration of 379-557 h;
- the return **HTF** temperature exceeds the condensation range for a duration of 8-23 h;
- the short periods during which the average **HTF** temperature exceeds the condensation range are relatively few, in total 10 periods;
- the number of longer periods during which the average **HTF** temperature exceeds the condensation range is relatively high: 5 continuous periods of 12-24 h, 3 continuous periods of 24-36 h, 2 continuous periods of 36-48 h, 1 continuous period of 48-72 h, 2 continuous periods over 96 h.

The peak values of the real heat demand are determined by both the outdoor temperature as well as the wind velocity ($t_{out,\ R}$ = -12.3÷ -16.2 °C, v_R = 6÷8 m/s).

● In the chase of Cluj-Napoca:

- the average **HTF** temperature exceeds the condensation range for a duration of 308-601h;
- the return **HTF** temperature exceeds the condensation range for a duration of 5-16 h;
- the short periods during which the average **HTF** temperature exceeds the condensation range are relatively few, in total 11 periods;
- the number of longer periods during which the average **HTF** temperature exceeds the condensation range is relatively high: 8 continuous periods of 12-24 h, 2 continuous

periods of 24-36 h, 1 continuous period of 36-48 h, 1 continuous period of 48-72 h, 2 continuous periods over 72 h.

The peak values of the real heat demand are determined mainly by the outdoor temperatures rather than the wind velocity ($t_{out,\ R}$ = -15.1 ÷ -18.5 °C, v_R = 2÷3 m/s).

● In the chase of Braşov:

- the average **HTF** temperature exceeds the condensation range for a duration of 441-568 h;
- the return **HTF** temperature exceeds the condensation range for a duration of 32-42 h;
- the short periods during which the average **HTF** temperature exceeds the condensation range are relatively few, in total 10 periods;
- the number of longer periods during which the average **HTF** temperature exceeds the condensation range is relatively high: 1 continuous period of 12-24 h, 2 continuous periods of 24-36 h, 1 continuous period of 36-48 h, 2 continuous periods of 48-72 h, 3 continuous periods over 72 h.

Regarding the climatic parameters of Braşov (**TMY**), the following are specified:

- the real outdoor temperature reaches a minimum value of: $t_{out,R}$ = - 27.2 °C, far lower than the conventional outdoor calculation temperature t_{out} = - 21 °C, for a period of 33 h;
- the value of the real wind velocity exceeds 10 m/s (maximum value 24 m/s), being far higher than the conventional wind velocity.

It is considered that in the case of Braşov the climate zone V (t_{out} = - 25 °C), is in better agreement with the real climatic conditions, than the climatic zone IV.

The peak values of the real heat demand are determined mainly by the outdoor temperatures rather than the wind velocity ($t_{out,\ R}$ = -20.8 ÷ -27.2 °C, v_R = 0÷1 m/s).

From an economical point of view, due to the reduced periods where the average **HTF** temperature exceeds the condensation range, the over sizing of the heating elements at: 60/40 °C flow/return temperatures is not justified.

6. REFERENCES

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STUDIU PRIVIND PARAMETRII DE FUNCŢIONARE AI INSTALAŢIILOR DE INCALZIRE CENTRALA ÎN REGIM DE CONDENSAŢIE

Rezumat: *O posibilitate de reducere a consumurilor de combustibili gazoşi utilizaţi în instalaţiile de încălzire centrală constă în utilizarea cazanelor în condensaţie caracterizate prin randamente termice majorate datorită condensării vaporilor de apă din gazele rezultate ale arderii. Răcirea gazelor de ardere sub temperatura de saturaţie, implică temperaturi scăzute ale apei calde de încălzire tur/retur: 60/40 °C, ceea ce conduce la supradimensionarea corpurilor de încălzire, determinând majorarea semnificativă a costurilor investiţiei. În consecinţă, apare ca raţională alegerea corpurilor de încălzire pentru temperaturi de tur/retur: 80/60 °C. Lucrarea îşi propune determinarea temperaturii medii a apei calde de încălzire în funcţie de temperatura exterioară şi identificarea perioadelor când are loc condensarea vaporilor din gazele de ardere, ceea ce determină randamentul sporit al cazanelor în condensaţie.*

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