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Experimental and theoretical study regarding the thermal performances of the heat exchangers with steel panels and extended surfaces

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Abstract: *The present paper refers to the thermal performances of the heat exchangers with steel panels, used in the buildings' heating installations. The heat exchanger was set up in a thermostatic test room, built in conformity with SR EN 442-2:2002/A2:2004 and was tested in standard functioning conditions. The experimental were compared with those obtained by applying the correlations that characterize the heat transfer for the flow of liquids through narrow vertical spaces and the flow of air by natural convection in different situations.*

Keywords: heat exchanger; thermal performances

Nomenclature

c_p	specific heat capacity, [kJ/kgK]	*Greek letters*	
D_h	hydraulic diameter, [m]		
Gr	Grashof adimensional criteria	α	convective coef. [W/m^{2}K]
k	global heat transfer, [W/m^{2}K]	ε	emissivity [-]
L	length [m]	λ	conductivity coef. [W/mK]
$\dot{m}$	water flow, [kg/s]	ρ	water density [kg/m^3]
Nu	Nusselt adimensional criteria	υ	kinematic viscosity [m^2/s]
Pr	Prandtl adimensional criteria	σ	Stefan-Boltzmann constant
Re	Reynolds adimensional criteria	δ	thickness [m]
$\dot{Q}$	heat flux, [W]		
$S_{transfer}$	heat transfer surface, [-]	*Subscripts*	
S	flowing section, [m^2]		
w	water velocity, [m/s]	w	water

1. Introduction

Radiators are part of the family of heat exchangers used for living spaces heating. These can be made in several variants and by using different materials of which the most common are steel, cast iron, and aluminum. Among all types of heaters, the most currently used are the steel panels.

The authors analyze how heat transfer occurs from the surface of a radiator to the environment air, using relationships Criteria and ultimately determining the heat flux transmitted. The tested radiator is a steel panels type PKKP 1000x600 mm with the following dimensional characteristics measured and calculated by the authors.

- The hydraulic diameter of the flow of the water through vertical channels is 0.0138 m;
- Total heat transfer surface, corresponding to a length of 1 m, is 6.1 m^2, of which 2,564 m^2 is the panels' surface (4 x 0.641 m^2) and 3.54 m^2 is represented by the extended surface;
- Hexagonal section flow of the heating water is 0.000183 m^2. In total, an 1 m radiator has 54 such sections flow, resulting a total flow section of 0.009882 m^2.

The total heat transfer area is the sum of the exterior surface of the panel and the total area of the extended surfaces corresponding to an 1 m radiator length.

The outer surface of the panel resulted by applying a finning coefficient given by the producer to the value of the plane surface of the panel.

Extended surface area was calculated as the product of length corresponding to 1 m linear radiator and its height. The measured height of the sheet forming the extended surface has a value of 0.4 m from the total radiator's total height of 0.6 m.

Experimental research in order to determine the thermal power of the radiator were held in a specially thermostatic test room builted in the Laboratory of Thermotechnics from Technical University of Civil Engineering, Bucharest.

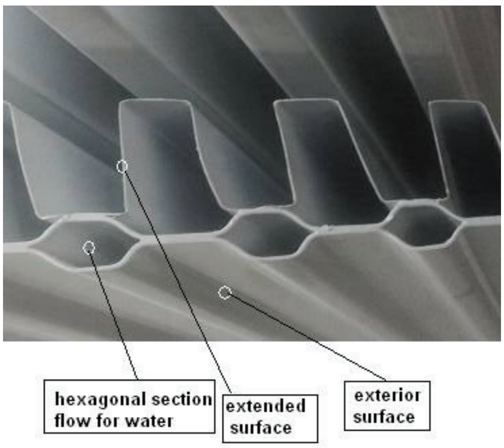

Figure 1. Hexagonal sections of the heat flow

The testing room was built in accordance with European Standard EN 442-2 for the experimental determination of heat power of radiators and convectors.

2. Experimental study for the determination of radiator's heat power

2.1 Experimental stand

The test room is an unventilated space where the tested radiator was mounted. The walls, ceiling and floor of this room are made of sandwich panels containing a copper coil Φ12x1. Inside this coil circulates cooling water as a medium to stabilize the room internal temperature regardless of the heat generated inside and regardless of the temperature outside the room. Cooling water is circulated by a group of five pumps located on the top of the test chamber. Schematically, the camera is shown in Figure 2.

Air temperature in the test chamber is maintained at 20±0.5 °C by adjusting the parameters (flow and temperature) of the cooling water that circulates through coils in the walls of the test chamber. Room air temperature is measured with thermocouples NiCr-Ni, type T190-0 in 4 points on the vertical axis located in the geometric center of the room.

The circulating fluid through the radiator is hot water, prepared in a boiler that feeds a tank first and then by freefall the tested radiator. This mode is used for maintaining constant the hot water flow and is a standard requirement.

To adjust the hot water temperature, an electrical resistance heater is provided into the tank, which start in steps, depending on the water temperature. Heat flow is measured by a Corriollis flowmeter mounted at the output of the radiator heating circuit. Water temperature at the inlet and the outlet of the radiator is measured by two Pt100 temperature sensors.

Radiator surface temperature was determined as an average of 12 thermocouples indications, mounted at equal distances.

Temperature of the air inside the radiator was determined as an average of the medium of five thermocouples mounted on the top of the radiator and the test chamber air temperature at the enter on the bottom of the radiator.

Sensors to measure air and water temperature were connected to a data acquisition device.

The experiments were carried out to an average temperature of the hot water at about 70 ° C (water temperature at the radiator inlet: 75 ° C and the water temperature at the exit of the radiator: 65 ° C) and the temperature of the air in the test chamber at about 20 ° C. In Figure 1 it can be seen the feeding scheme of the coils inside the chamber walls and the radiator mount mode for testing.

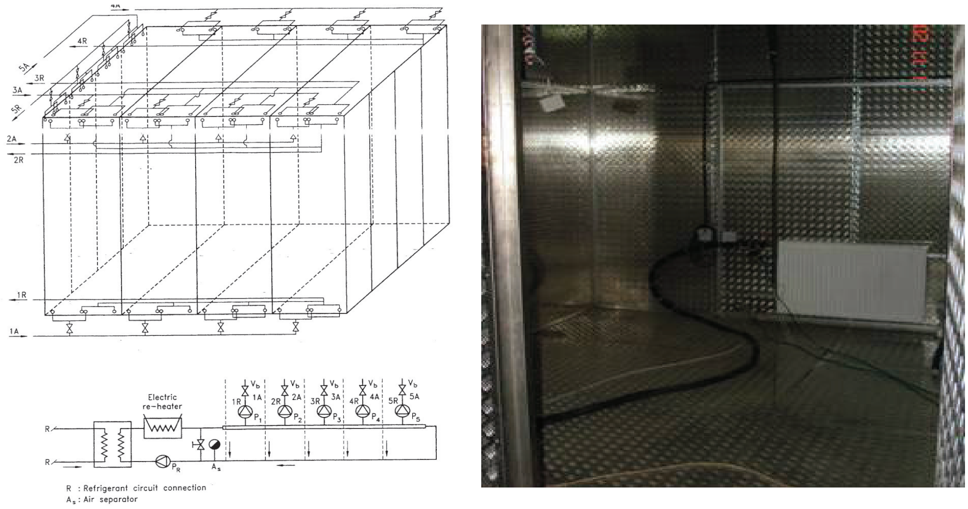

Figure 2. Test chamber scheme

2.2 The testing methodology

Measurements for the determination of the heat power were made after installation of a quasi-stationary regime of operation.

It was considered that the quasi-stationary operating regime is installed when:

- Temperature of water flow and the test chamber air temperature did not vary during 30 minutes with more than ± 0.1 K;
- Water flow did not vary during 30 minutes with more than ± 1%.

2.3 Results obtained by direct measurement

In Table 1 are presented the measured values of the parameters of interest, after installing the quasi-stationary regime.

Table 1

Date and hour of the test	H.E. type	Testing room temp. [°C]	Water Flow [kg/s]	Air medium temp. at the exit from the H.E. [°C]	Inlet water temp. [°C]	Outlet water temp. [°C]	H.E. surface medium temperature [°C]
30.05.13 10:06	0.6 x 1 m	20.5	0.0298	43.1	75.5	65	69.3
30.05.13 10:16	0.6 x 1 m	20.5	0.0298	43.2	75.4	64.9	69.2
30.05.13 10:26	0.6 x 1 m	20.4	0.0298	43.1	75.5	65	69.2
30.05.13 10:36	0.6 x 1 m	20.4	0.0298	43.1	75.5	64.9	69.2
30.05.13 10:46	0.6 x 1 m	20.4	0.0298	43.2	75.6	64.9	69.3

3. Theoretic study for determination of thermal power of the radiator

Theoretical study sought to determine the heat power of a radiator with geometric and functional characteristics of corresponding test criteria.

3.1 Methodology

The input data for theoretical study are the measured values of water flow and medium temperature (hot water and room air).

Using the equation of continuity and having known volume flow and flow section we could determine the speed of the water inside the hexagonal channels of the radiator. Reynolds criteria values obtained indicates that the flow is laminar.

3.2. Calculation of the heat flow

Heat transfer in the case studied is complex. Thus we have 3 situations:

- **From the front panel of the radiator to the test chamber air- free convection and radiation in large spaces;**

- *The water heat convection coefficient*

For water flow in canals and pipelines we have the following relationship, [1]:

$$Nu = 3.66 + \frac{0.065 \cdot (D/L) \cdot \mathrm{Re} \cdot \mathrm{Pr}}{1 + 0.04 \cdot [(D/L) \cdot \mathrm{Re} \cdot \mathrm{Pr}]^{2/3}} \tag{1}$$

It will be used the thermophysical properties of water at a temperature of 70.2 ° C calculated as an average between the inlet and outlet of the radiator.

In order to determine the dimensionless Reynolds criteria, velocity of water in the radiator is obtained from the equation of continuity, depending on the mass flow rate, flow section and density of water at ambient temperature.

$$w = \frac{\dot{m}}{S \cdot \rho} = \frac{0{,}0298}{0.009882 \cdot 974.8} = 0{,}0031 m/s \tag{2}$$

$$\mathrm{Re} = \frac{w \cdot l_C}{\upsilon} = \frac{w \cdot D_h}{\upsilon} = \frac{0.0031 \cdot 0.0138}{0.39 \cdot 10^{-6}} = 109.7 \tag{3}$$

$$Nu = 3.66 + \frac{0.065 \cdot (D/L) \cdot \mathrm{Re} \cdot \mathrm{Pr}}{1 + 0.04 \cdot [(D/L) \cdot \mathrm{Re} \cdot \mathrm{Pr}]^{2/3}} = 4.005 \tag{4}$$

With this value known we can determine the coefficient of heat transfer from the water.

$$Nu = \frac{\alpha_1 \cdot l_C}{\lambda} \Rightarrow \alpha_1 = \frac{Nu \cdot \lambda}{D_h} = \frac{4.005 \cdot 0.671}{0.0138} = 194.74\ W/m^2K \tag{5}$$

- *The heat transfer coefficient from the air*

The free convection flow regime type is determined by the product $Gr \cdot \Pr$.

The medium temperature has a value of 44.9 °C, given by the average surface temperature of the radiator and the air temperature inside the test chamber. Depending on the temperature we could determine the physical properties of air.

Characteristic length is given by the dimension in the direction of air flow, respectively the radiator height of 0.6 m

$$Gr \cdot \Pr = \beta \cdot \frac{g \cdot l_C^3}{\upsilon^2} \cdot \Delta T \cdot \frac{\upsilon}{a}; \quad \beta = \frac{1}{T_m} = \frac{1}{318.05} \tag{6}$$

$$Gr \cdot \Pr = 3.67 \cdot 10^8 > 2 \cdot 10^7 \tag{7}$$

According to the obtained results, the air flow regime is turbulent one.

In calculating the heat transfer coefficient by convection was chosen an appropriate criteria ecuation corresponding to a free convection for a vertical plate in turbulent flow. We use the following equation given by Miheev.

$$Nu = 0{,}135 \cdot (Gr \cdot \Pr)^{0{,}33} = 90.5 \tag{8}$$

$$Nu = \frac{\alpha \cdot l_C}{\lambda} \Rightarrow \alpha_2 = \frac{Nu \cdot \lambda}{h} = 4.18\ W/m^2K \tag{9}$$

The heat transfer from the hot water to the air in the test chamber takes place simultaneously by convection and radiation, and therefore it will be calculated also the heat transfer coefficient of radiation from the radiator surface to the air.

Coefficient of heat transfer by radiation is determined by the empirical relationship, [1]:

$$\alpha_3 = \varepsilon \cdot \sigma_0 \cdot \frac{T_S^4 - T_{aer}^4}{T_S - T_{aer}} = 6.92\ W/m^2K \tag{10}$$

The overall coefficient of heat transfer from the radiator to the air in the room will be:

$$\alpha_{total} = \alpha_2 + \alpha_3 = 11.1\ W/m^2K \tag{11}$$

The heat flux transferred in this specific case will be:

$$\dot{Q}_1 = \alpha \cdot S \cdot \Delta t = 11.1 \cdot 0.641 \cdot 48.8 = 347.2\ W \tag{12}$$

- **From the front panel of the radiator placed towards the wall of the test chamber, to the air**

We have free convection and radiation in limited spaces between flat wall surface of the radiator and indoor air.

In order to determine the convective heat transfer coefficient will be used Berkovska relations referring to free convection in space bounded by two plates of different temperatures. This relationship is valid for Pr <10 and $Gr \cdot \Pr < 10^{10}$, [3]. Characteristic length is considered to be the height of the radiator face, along which there is air circulation.

$$\overline{Nu_\delta} = 0.22 \cdot \left(\frac{L}{\delta}\right)^{-1/4} \cdot \left(\frac{\Pr}{0.2 + \Pr} \cdot Gr \cdot \Pr\right)^{0.28} = 73.87 \tag{13}$$

Using thermophysical properties of air at average ambient temperature we could determine convective heat transfer coefficient:

$$\overline{Nu_\delta} = \frac{\bar{\alpha} \cdot l_C}{\lambda} \Rightarrow \alpha_4 = \frac{\overline{Nu_\delta} \cdot \lambda}{h} = 3.398\ W / m^2 K \tag{14}$$

The heat flux transferred in this case is:

$$\dot{Q}_{2,cv} = \alpha_4 \cdot S \cdot \Delta t = 3.398 \cdot 0.641 \cdot 48.8 = 106.3\ W \tag{15}$$

To analyze the flow component related to radiation, we consider flux transmitted to be equal to the heat flow exchanged by radiation between two paralel walls, [4].

$$\dot{Q}_{2,rad} = \varepsilon \cdot \varphi \cdot S \cdot C_0 \cdot \left[\left(\frac{T_{S1}}{100}\right)^4 - \left(\frac{T_{S2}}{100}\right)^4\right] = 133.09\ W \tag{16}$$

The reduced emissivity is depending on the emissivities of the two surfaces which exchange heat by radiation. Emissivity of the steel sheet, has a value of 0.95, and the emissivity value of stainless steel sheet from which is made the test chamber wall has a value of 0.75. For the reduced emissivity result a value of $\varepsilon = 0.58$.

We make the simplifying assumption that the factor of mutual irradiation has value $\varphi = 1$ for the two flat parallel surfaces.

Finally, the total flux transmitted to the air is calculated as a sum of convective and radiative component.

$$\dot{Q}_2 = \dot{Q}_{2,cv} + \dot{Q}_{2,rad} = 239.4\ W \tag{17}$$

- **From the inner sides of the radiator and extended surface to the air that circulates through the inside of the radiator**

We have heat transfer by conduction through extended surfaces between the inner walls of the radiator and extended surface, heat transfer by free convection between the radiator inner flat walls including extended surface, and air;

- *Determining the plane inner surface of the radiator*

To determine the plane surface of the radiator in contact with indoor air will be subtracted from the total area of the inner face of the radiator surface the area occupied by the base of the extended surface, which have a thickness of 0.5 mm.

$$S_{plane} = S_{tot} - S_{base,ext.surfaces} = 1.282 - 0.5 \cdot 10^{-3} \cdot 0.4 \cdot 54 \cdot 2 = 1.271 m^2 \tag{18}$$

The total area of the extended surface in contact with air is $S_{ext.surfaces} = 3.542\ m^2$.

- *The total heat flux transferred in this case*

$$\dot{Q}_3 = \dot{Q}_{plane,surfaces} + \dot{Q}_{ext,surfaces} \tag{19}$$

Initially, we calculate the heat flux transmitted to the air inside the radiator, without considering the extended surface. Flux transmitted by the inner plane surface of the radiator is determined by the average surface temperature and the air inside the radiator's temperature measured as a media between inlet and outlet from radiator, 31.8 ° C.

Coefficient of heat transfer by convection is determined by applying the same methodology as the first point, because the distance between the two radiator panels is large enough to no longer stay within limited space situation. It results a total convection heat ransfer coefficient: $\alpha_{4,cv} = 4{,}15\ W / m^2 K$

$$\dot{Q}_4 = \alpha_{4,cv} \cdot S \cdot \Delta t_{\text{int}} = 4.15 \cdot 0.641 \cdot 2 \cdot 37.5 = 199.51\ W \tag{20}$$

The purpose of this calculation is to demonstrate further the usefulness of applying extended surfaces.

Next, using the above data, it will be determined the heat flux transmitted through the inner plane surface of the radiator where it is applied the base of the extended surfaces.

$$\dot{Q}_{plane,surfaces} = \alpha_{4,cv} \cdot S_{plane} \cdot \Delta t_{\text{int}} = 4.15 \cdot 1.271 \cdot 37.5 = 197.8\ W \tag{21}$$

To obtain the heat flux transmitted through extended surface we calculate the average temperature of the sheet from which is made that area. Since U-shaped extended surface has a total length of 0.082 m, to simplify the calculation we will consider it composed of two rectangular fins of equal length joined at an end, and will determine the temperature at mid, which we will consider further the medium temperature of the extended surfaces.

$$t(x) = t_f + (t_0 - t_f)\frac{\cosh m \cdot (L-x) + \dfrac{\alpha}{\lambda_{OL} \cdot m}\sinh m \cdot (L-x)}{\cosh m \cdot L + \dfrac{\alpha}{\lambda_{OL} \cdot m}\sinh m \cdot L} \qquad (22)$$

In this equation the factor m is calculated according to the area and perimeter of the fin section. After calculation resulted in a value m = 19.5 [m^{-1}].
Distance x, for which the calculation is done at the mid of the fin was measured and has a value of 0.0205 m. It results an average extended surface temperature
$t_{m,\ ext\ surface}$= 63.7 °C.

The heat flux transmitted from the extended surface to the air inside the radiator will be:

$$\dot{Q}_{ext,surface} = \alpha_{total,2} \cdot S_{ext,surface} \cdot \Delta t_{ext\ surface-air} = 4.15 \cdot 3.542 \cdot 31.9 = 468.91\ W \qquad (23)$$

The total heat flux transmitted to the air from the interior of the testing room is calculated as a sum of the four components calculated above. It has a value of 1253 W.

4. Analysis of the results

The analysis of data obtained from mathematical modeling of heat transfer processes taking place between the body surface of the radiator and the air from testing room have highlighted the following:

• Heat transfer area of the radiator was calculated as a sum between the heating panel plane surface, corrected by a factor of finning and the extended surface area represented by the corrugated sheet. Note that the heat transfer surface grows about 2.4 times with this expansion;

• The coefficient of heat transfer from the radiator to the air in the enclosure is a complex one that takes into account both the coefficient of convection and radiation at the surface of the radiator to the air from the test chamber. Of the total heat transfer coefficient, 38% is the contribution of heat transfer by convection and the radiation contribution is represented by the remaining 62%;

• When considering the situation in which additional radiator area was not extended with corrugated sheet, we obtained a total heat flux transferred to the indoor air with 37% down as compared to the situation when those extended surfaces would have been applied;

• The radiator heat power can be determined according to European Standard EN 442, [2], as the product of mass flow rate of the heating fluid, specific heat of water and the temperature difference between inlet and outlet of the water.

$$\dot{Q} = \dot{m} \cdot c_p \cdot \Delta t_w = 1310\ W \tag{24}$$

- By adding all the flows obtained from the mathematical modeling of the heat transfer proceses we obtain a value with 57 W lower as compared to the application of the standard formula.

5. Conclusions

The study made in the present paper highlights the usefulness of extending the surface of heat transfer, the heat flux transmitted by this additional area representing 37% of the total flux transmitted in the analysed situation.

By comparing the result of the mathematical modelling with the value obtained for the total flux transmitted by the radiator surface to the air inside the testing room resulting from applying the formula given in the relevant European Standard for heaters, results a deviation of 4.3%, which is lower than that allowed in the literature. Thus, by applying appropriate formulas for each heat transfer situation, we get a validation of the standard calculation formula.

6. Bibliography

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