E3S Web of Conferences 13, 02005 (2017) DOI: 10.1051/e3sconf/20171302005
WTiUE 2016

Experimental and theoretical investigations of special type coil heat exchanger with the nanofluid buffer layer

Robert Smusz[1], and *Joanna* Wilk[1,*]

[1]Rzeszów University of Technology, Thermodynamics Department, Faculty of Engineering and Aeronautics, Al. Powstańców Warszawy 12, 35-959 Rzeszów, Poland

Abstract. The paper presents the results of experimental and theoretical investigations of special type of coil heat exchanger. The tested device is equipped with three vertical coils and the temperature stratification system. Water is a heating medium in two coils. The refrigerant transferring the waste heat from air conditioning system is the heating medium in the third coil. The finned pipe of this coil has a double wall in which the annular buffer layer with nanofluid is mounted. Thermophysical properties of the applied water based Cu nanofluid cause the enhancement of heat transfer through the buffer layer. The paper presents thermal characteristics of the exchanger received on the basis of measurements performed on the industrial test stand. Measurements were conducted during the operation of the coil with refrigerant. Heat loss to the surroundings, distributions of water temperature in the storage tank, changes of water temperature in time and thermal power of the coil heat exchanger were obtained. The measurement results were compared with those received on the basis of theoretical analysis of the exchanger.

1 Introduction

Heat exchangers with helical coils have widespread domestic and industrial applications. They may be used in domestic hot water systems, air conditioning systems, refrigerators, chemical and nuclear reactors, steam generators, etc. Applied in heat exchangers the helical coils are often used since they can accommodate a large heat transfer area in compact space and heat transfer coefficients have high values. In recent time coil heat exchangers were the subject of many researchers. The exemplary works on the considered exchangers are [1–5]. Prabhanjan et al. presents in [1] the comparison of heat transfer between coil heat exchanger and straight tube heat exchanger. The work [2] by Zachar deals to improvement of heat transfer processes in coil heat exchanger with spirally corrugated wall. In the paper [3] Genic et al. presents the results of investigations of the influence of geometry parameters of coil heat exchangers on heat transfer coefficients. Urbanowicz and Wojtkowiak presents in [4] a comparative analysis of correlations for heat transfer at surfaces of coil and suggests an original method of calculation of heat exchanger with helical coil. In turn, characteristics of agitated helical coil heat exchanger operating with the nanofluid are described by Srinivas and Vinod in [5]. Tested in the present work the coil heat exchanger works with the nanofluid too. The investigated device is designed to warm domestic hot water. It is equipped with three vertical coils immersed in storage tank filled with heated water. The orientation of the coil system is shown in Fig.1. Additionally the temperature stratification system is applied in the storage tank. Water is a heating medium in two coils. The refrigerant transferring the waste heat from air conditioning system is the heating medium in the third coil. The finned pipe of this coil has a double wall in which the annular buffer layer with nanofluid is mounted. According to the rules the layer have to protect against the possible refrigerant leakage to the hot domestic water tank. The gap is filled by the nanofluid consisting of distilled water and coper nanoparticles. Thermophysical properties of the applied water based Cu nanofluid cause the enhancement of heat transfer through the buffer layer. Thus, the buffer layer with nanofluid acts as a protection, and in the other hand it improves thermal characteristics of the exchanger.

A detailed description of the idea and the analytical model of the proposed coil heat exchanger is contained in [6]. In turn the results of calculations of thermal power of water coil of the exchanger considered are included [7]. This paper presents also the experimental results of water coil in comparison with received theoretical results. The present work contains the results of thermal measurements which were conducted during the operation of the coil with refrigerant. Heat loss to the surroundings, distributions of water temperature in the storage tank, changes of water temperature in time and thermal power of the coil heat exchanger were obtained. The measurement results were compared with those received on the basis of theoretical analysis of the exchanger.

* Corresponding author: joanwilk@prz.edu.pl

 DOI: 10.1051/e3sconf/20171302005

2 Experimental investigations

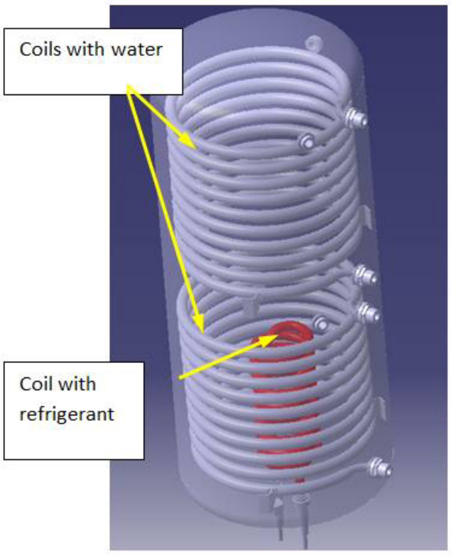


Fig. 1. System of coils applied in the heat exchanger

Experimental investigations on considered heat exchanger with the coil that uses the waste heat from air conditioning system were performed on industrial test stand. The detailed description including the technological scheme of the stand, the specification of applied elements and the research possibilities of the stand were presented in [8]. The simplified scheme of the experimental stand is shown in Fig.2.

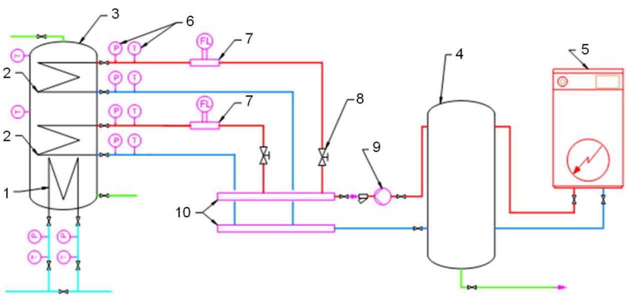


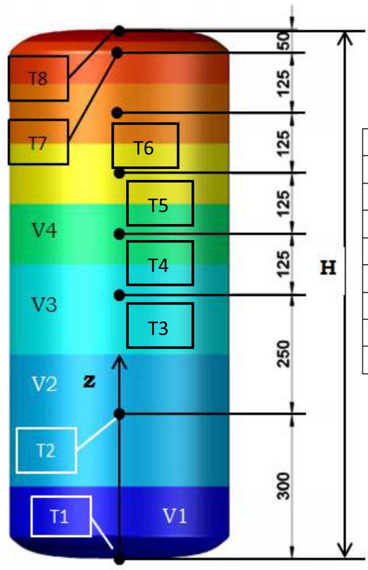
Fig. 2. Simplified scheme of experimental stand: 1 – coil with refrigerant, 2 – water coil, 3 – storage tank, 4 – additional buffer tank, 5 – thermostat, 6 – temperature and pressure sensors, 7 – flow meter, 8 – valve of flow control, 9 – pump, 10 – manifold

The measuring system of the experimental stand has enabled the measurements of temperature of the outer surface of the tank and the same determination of mean temperature of domestic hot water in the storage tank T_m was possible. Heat conduction through the wall of the storage tank has been omitted. T_m has been obtained with the use of eight resistance temperature detectors T_i [8] located on the surface of the tank (see Fig. 3) according to the formula

$$T_m = \sum_{i=1}^{8} T_i \cdot \frac{V_i}{V}, \tag{1}$$

where V_i – control volume assumed for the temperature measured T_i.

The basic geometrical parameters necessary to calculate T_m is summarised in Fig.3.



T_i	z/H	V_i/V
T1	0	0,124
T2	0,27	0,260
T3	0,5	0,175
T4	0,61	0,118
T5	0,73	0,118
T6	0,84	0,118
T7	0,95	0,072
T8	1	0,015

Fig. 3. Location of temperature detectors and relative control volumes of the storage tank

The results of investigations of the heating process of water in the storage tank are presented in Fig. 4. There are changes of water temperatures T1 – T8 during the thermal start-up of the exchanger operating with the use of coil with refrigerant – the exchanger of heat recovery system. The refrigerant R407C has been applied.

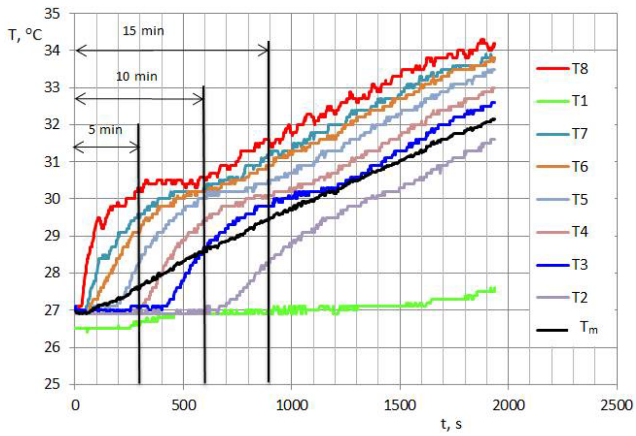


Fig. 4. Temperature of water in storage tank in function of time of water heating

The applied measuring system also enabled the measurement of the ambient temperature and supply and return temperatures of the coil. The experiment was performed at measured mean ambient temperature 26.4°C. The mean supply temperature during the water heating with the use of the coil with refrigerant was equal 44.74°C. The return temperature was respectively 40.2°C. Calculated according to eq.(1) the mean water temperature T_m is shown in Fig.5.

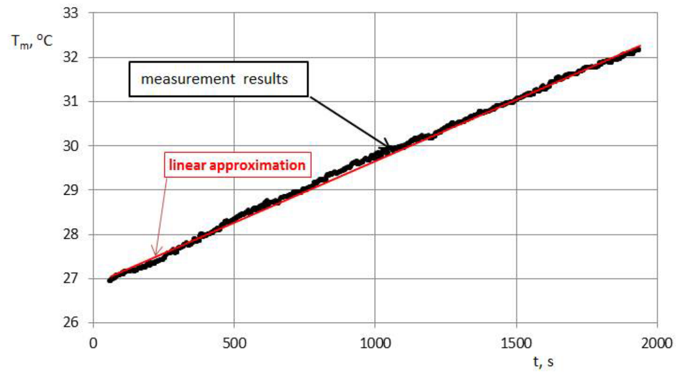


Fig. 5. Mean water temperature in function of heating time

The linear approximation of T_m in function of time has been made. It was necessary to obtain the rate of heating water in storage tank dT_m/dt.
Energy balance of the exchanger – heater – has a form

$$V\rho_w c_w \frac{dT_m}{dt} = \dot{Q} - \dot{Q}_{loss}\,, \tag{2}$$

where: V– volume of water in the tank, ρ_w, c_w – density and heat capacity of water, $\dot{Q}$ - thermal power of the exchanger, $\dot{Q}_{loss}$ - heat loss to the surroundings.

On the basis of eq.(2) the thermal power of coil exchanger of heat recovery system has been calculated. Heat losses from the storage tank have been measured with the use of three film heat flux sensors [8]. The results of the measurements and the linear approximation are presented in Fig.6. The product $\rho_w c_w$ in eq.(2) has been calculated on the basis of approximation of literature data [9] according to the formula

$$\rho_w c_w = 4211.7 - 1.6796\ T_m\,. \tag{3}$$

The obtained mean value of thermal power of coil heat exchanger operating with refrigerant as a heating medium is equal to 2.452 kW.

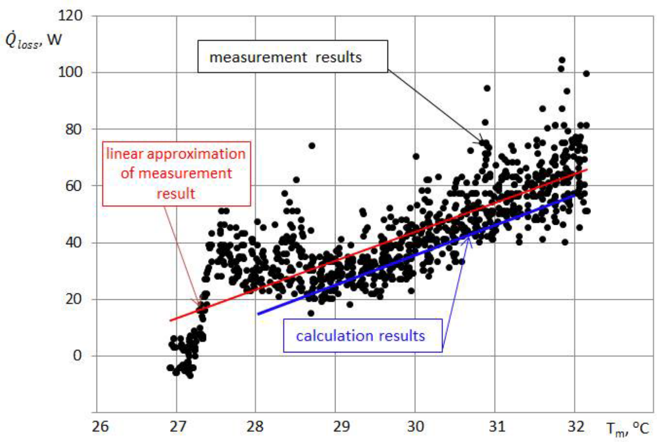


Fig. 6. Heat losses from the outer surface of the storage tank

3 Theoretical calculations

Theoretical calculations of the coil heat exchanger were based on heat transfer processes occurring in the exchanger during operation. The analysis has related to: convective and radiative heat transfer from the outer surface of the storage tank, convective heat transfer between water and the inner surface of the tank, natural convection from the outer surface of finned coil pipe, heat conduction through the coil wall with taking into consideration the additional thermal resistance of the nanofluid buffer layer and convective turbulent flow of the refrigerant inside the coil pipe. The main formulas used to calculations are summarised in Table 1. Characteristic dimensions occurring in dimensionless numbers were described in [6].

Table 1. Equations for the calculation of heat transfer process in the coil heat exchanger [10 – 12]

Heat transfer process	Formula
Radiative heat transfer from the outer surface of the tank	$h_r = \dfrac{\sigma \cdot \varepsilon \cdot \left(T_{wall}^4 - T_a^4\right)}{T_{wall} - T_a}$
Convective heat transfer between air or water and the storage tank walls	$Nu = 0.59 \cdot Ra^{0.25}$ $\quad 10^4 < Ra < 10^9$ $Nu = 0.1 \cdot Ra^{0.333}$ $\quad 10^9 < Ra < 10^{13}$
Natural convection from the surface of the finned pipe	$Nu = 0.028 \cdot Ra^{0.4}$ $1.2 \times 10^7 < Ra < 1.4 \times 10^8$
Heat transfer in annular buffer layer with nanofluid	$\dfrac{k_{eq}}{k} = 0.11 \cdot Ra^{0.29}$ $6000 < Ra < 10^6$
Heat transfer during the turbulent flow of refrigerant through the coil	$Nu = 0.023\left[1 + 3.6\left(1 - \dfrac{d_{in}}{D}\right)\left(\dfrac{d_{in}}{D}\right)^{0.8}\right] \times$ $\times \mathrm{Pr}^{1/3}\,\mathrm{Re}^{0.8}$ $2 \times 10^4 < \mathrm{Re} < 1.5 \times 10^5$

d_{in} – inner diameter of the coil pipe
D – characteristic dimension of the coil
h_r – radiative heat transfer coefficient
k, k_{eq} – thermal conductivity and equivalent thermal conductivity
Nu – Nusselt number
Pr – Prandtl number
Ra – Rayleigh number
Re – Reynolds number
T_{wall}, T_a– wall and ambient temperature
ε – surface emissivity
σ – Boltzmann constant

The presented procedure has enabled obtain heat losses from the storage tank and the thermal power of the coil heat exchanger. The thermal power of coil exchanger of heat recovery system – the coil with refrigerant – has been calculated with the use of the general formula

$$\dot{Q} = A \cdot U \cdot \frac{(T_1 - T_m) - (T_2 - T_m)}{\ln \dfrac{T_1 - T_m}{T_2 - T_m}}, \tag{4}$$

where: A – overall coil pipe surface; U – overall heat transfer coefficient; T_1, T_2 – temperature of refrigerant at the inlet and the outlet of the coil, respectively, T_m – assumed mean temperature of water. The overall heat transfer coefficient takes into account the following components: convective heat transfer inside coil, conduction through the inner wall of the coil pipe, heat transfer in the nanofluid buffer layer, conduction through the outer wall, convective heat transfer from finned wall of the coil. It is given by

$$U \cdot A = \tag{5}$$

$$= \frac{1}{\dfrac{1}{h_i \cdot A_{ip}} + R_{k1} + R_b + R_{k2} + \dfrac{1}{h_{eq} \cdot A_{in\_f} + h_{eq}\eta_f N_f A_f}},$$

where: h_i – heat transfer coefficient at the inner surface of the coil pipe A_{ip}, h_{eq} – equivalent heat transfer coefficient of the finned surface, A_{in_f} and A_f – intercostal surface and the surface of the fin respectively, η_f – efficiency of the fin, N_f – the number of fins. R_{k1} – conduction thermal resistance of the inner coil pipe; R_{k2} – conduction thermal resistance of the outer finned coil pipe; R_b – conduction thermal resistance of the buffer layer taking into account equivalent thermal conductivity of the nanofluid.

Calculations were performed with taking into account the temperature dependence of thermophysical properties of the applied medium. The latent heat of the refrigerant was omitted because of temperature range of the working medium. Since heat transfer coefficients were implicit temperature functions, the system of nonlinear equations was built. It has been solved using the secant method. Geometrical parameters of the tank and the coil were assumed as the same as in the real object.

Table 2. Geometry of the finned copper pipe of the coil with refrigerant and calculated thermal power of the exchanger

Outer and inner diameter of finned coil pipe	16mm, 12mm
Thickness of the inner wall of the double-wall coil pipe δ_i	0.8 mm
Thickness of the outer wall of coil pipe δ_o	0.8 mm
Thickness of buffer layer filled with nanofluid δ_b	0.4 mm
Nanofluid – Cu particles of purity 97% and spherical form of diameter about 40 nm dispersed in water of 1% mass concentration	
Mean coil diameter	130 mm
Coil pitch	31 mm
Number of coils	14
Coil pipe length	5.7 m
Fin height	3.5mm
Fin width	0.5 mm
Number of fins per inch of coil pipe length	11

Table 3. The results of thermal power calculations

Supply temperature of refrigerant	Thermal power of coil heat exchanger
60 °C	4.5 kW
55 °C	3.9 kW
55 °C	3.3 kW
45 °C	2.7 kW

The main data are summarised in Table 2 according to the presented in Fig. 7 the scheme of the double wall finned pipe. Table 2 also presents the main characteristic of the nanofluid applied in the buffer layer. The results of calculations of thermal power of the exchanger at assumed temperatures of the refrigerant in the coil are summarized in Table 3.

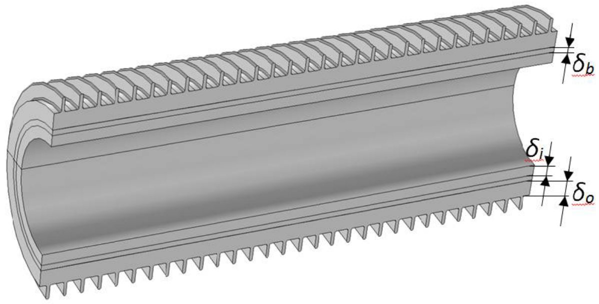


Fig. 7. Scheme of the double wall finned pipe

Table 4. Heat losses from the storage tank at T_a = 26.4°C, V = 200dm^3

Mean temperature of water in the storage tank	Heat loss from the tank
28 °C	14.1 W
29 °C	24.2 W
30 °C	34.8 W
31 °C	45.9 W
32 °C	57.3 W

In turn Table 4 includes the results of calculations of heat loss to the surrounding from the outer surface of the storage tank. Calculations were made for uninsulated tank (real conditions) at assumption ambient temperature 26.4°C. The linear approximation of the calculation results is shown in Fig.5.

4 Conclusions

Experimental and theoretical investigations on a special type of coil heat exchanger have allowed to formulate some conclusions from the obtained results.
The fast process of water heating in the storage tank was observed. By the time 5 minutes after the start of the exchanger the temperature of water is not rising to the 60% of the relative height of the tank. After 1900 second from the start of water heating with the use of coil with refrigerant the difference between T8 and T3 (see Fig. 4) is only 1.6 °C. Hot water is transported to the upper zone of the tank by a temperature stratification system. The obtained results indicate good operating of the system and may be helpful in optimization of the stratification system.
The calculation results of heat loss from the storage tank are in good agreement with measured heat loss. The linear approximation of the calculated heat loss is in the area of the scatter of measurement results. Mean value of heat loss measured at maximum water temperature 32°C is equal to 64.1W while the calculated value is 57.3W. The relative difference is about 10%. Good compatibility is also between the measured and calculated thermal power of the exchanger. Experimentally obtained thermal power is 2.452 kW at measured supply temperature of refrigerant 44.74 °C. Calculated values are respectively equal: 2.7 kW at 45 °C. The results obtained indicate a valid model for theoretical calculations. It can be applied to appropriate selection of insulation of the storage tank to reduce heat losses. Also it can be used for modeling of heat transfer in the buffer layer with nanofluid.

References

1. D.G. Prabhanjan, G.S.V. Raghavan, T.J. Rennie, Int. Comm. Heat Mass Transfer, **29**, 185-191 (2002)
2. A. Zachár, Int. J. of Heat Mass Transfer, **53**, 3928-3939 (2010)
3. S.B. Genić, B.M. Jaćimović, M.S. Jarić, N.J. Budimir, M.M. Dobrnjac, Int. J. of Heat and Mass Transfer, **55**, 4295-4300 (2012)
4. A.E. Urbanowicz, J. Wojtkowiak, *Metoda obliczeń wymienników ciepła z pionową wężownicą* (Wydawnictwo Politechniki Poznańskiej, Poznań 2013)
5. T. Srinivas, A.V. Vinod, Exp. Thermal and Fluid Science, **51,** 77-83 (2013)
6. R. Smusz, J. Wilk, Applied Mechanics and Materials, **831**, 223-231 (2016)
7. R. Smusz, Journal of Physics (to be published)
8. R. Smusz, J. Wilk, *Proceedings of 16th Int. Symposium on Heat Transfer and Renewable Sources of Energy*, 397-403 (2016)
9. Main IAPWS Thermodynamics Property Formulations, Updated July 6, http://www.iapws.org/newform.html (2015)
10. J. P. Holman, *Heat transfer* (McGraw-Hill Inc. 1992)
11. B. Niezgoda-Żelasko, W. Zalewski, *Chłodnicze i klimatyzacyjne wymienniki ciepła* (Wydawnictwo Politechniki Krakowskiej, Kraków 2013)
12. H. Berkel, N.K. Mitra, M. Fiebig, Wärme und Stoffübertragung, **18**, 167-173 (1984)