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THERMAL ANALYSIS OF A PLASTIC HELICAL COIL HEAT EXCHANGER FOR A DOMESTIC WATER STORAGE TANK

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Abstract. In the present study, the heat transfer coefficients of helically coiled corrugated plastic tube heat exchanger inside of the solar boiler vessel were investigated experimentally. The metal coil of the conventional solar boiler for domestic usage was replaced by a plastic tube and the results were compared with the numerical simulation and the technical documentation of the initial solar boiler. All the required parameters like inlet and outlet temperatures of tube-side and stratified temperatures, flow rate of fluids, etc. were measured using appropriate instruments. The test runs were performed for different temperatures inside the tank ranging from 30-60°C and different flow rates from which the heat transfer coefficients were calculated.

Keywords: Plastic heat exchanger, Shell and coiled tube, Experimental, Heat transfer coefficient

1. INTRODUCTION

Solar domestic hot water systems are a cost-effective and a sustainable way to generate hot water for the houses, that can be used in any climate. The most solar water systems require a well-insulated stratified storage tank. The water in the tank can be heated either directly, by means of electric heaters or heated from another energy source as solar collectors. A helically coiled tubes are typically placed into the tank to discharge/charge the water inside the tank for several reasons: flexibility and therefore easy accommodating to required length, easy construction, low cost and enhanced inner convection heat transfer compared to straight pipes (Fernández-Seara *et al.*, 2007a). The heat transfer through the coil depends on the forced convection through the tube, natural convection around the tube and conduction through the tube wall.

The polymer heat exchangers were introduced in 1988 by Reay (1989) and they have been studied since then by several researchers as replacement for metallic heat exchangers due to their advantages in low cost, light weight and resistance to corrosion and fouling (Cevallos *et al*., 2012; T'Joen, 2009). They can be used in a wide range of applications such as heat recovery, evaporation, refrigeration, water heating via solar power, electrical fluid heating, electric device cooling, water desalination and distillation (Chen, *et al.*, 2016).

The thermal performance and the cost analysis of for tube-in-shell heat exchangers and immersed tube banks, both made of polymers, for the purpose of solar water heating were studied by Liu et al. (2000). The high temperature nylon (HTN) and cross linked polyethylene (PEX) were chosen as materials and compared with copper. By determining the surface areas required to provide heat transfers of 3kW and 6kW with fixed geometry and arrangement of plastic tubes was found, that polymer heat exchangers can provide thermal output equivalent to conventional copper heat exchangers, with 80% of the cost of a copper tube-in-shell heat exchanger. The dimensions of the tubes were 9.53 mm of outer diameter (OD) and 1.78 mm of wall thickness for PEX; and 3.81 mm of OD and 0.2 mm of wall thickness for HTN. In another study, stability over the life cycle of the plastic heat exchanger at constant pressure was studied (Wu, *et al.*, 2004): The solar collector system had a polymer tube bundle heat exchanger immersed in a tilted enclosure, filled with fluid, and heated by solar radiation. Via a study of natural convection involving a tube bundle, it was found that the nylon tubes were able to withstand a pressure of 0.55 MPa and temperature of 82°C for at least 10 years. In a study of polymer based water storage systems, Davidson, *et al.* used immersed heat exchanger to discharge (and/or charge) the stored energy of unpressurized polymer water storage tanks. They stated that the use of polymer heat exchangers for solar hot water storage systems posed thermal and material challenges but were also promising for lower cost systems. Sharif *et. al.* (2016) investigated the possibility of using polymeric helical coil heat exchanger as an alternative to metallic helical coil. They provided a model to calculate plastic coil dimensions with a thermal output equivalent to that of a conventional solar boiler. More experimental results on an ad-hoc demonstrator with a *helical coil* made of plastic could shed light on feasibility of replacing its metallic counterpart.

This paper describes an experimental work carried out to analyze the thermal performance of a vertical plastic helical coil in a domestic hot water storage tank. The heat exchanger is placed in the middle of the tank in two coils in parallel and filling almost whole height of the vessel in order to achieve required surface area for comparable heat transfer coefficient with a metallic coil. Water from the network was allowed to flow through the coil, while water in the tank was initially heated to the required temperature by an external electrical heater. This configuration, once reached the steady state, provided a practically constant water temperature around the coil in the tank. The test runs were performed for different initial tank temperatures in a range of 30-60°C and for case of 60°C initial temperature for different flow rates where the heat transfer coefficients were calculated.

NOMENCLATURE

A	[m^2]	Area
c_p/c_v	[J/kgK]	Specific heat at constant pressure/volume
D	[m]	Diameter
F	[-]	Correction factor
h	[W/m^{2}K]	Heat transfer coefficient
λ	[W/mK]	Thermal conductivity
L	[m]	Length
$\dot{m}$	[kg/s]	Mass flow rate
$\dot{Q}$	[W]	Heat transfer rate
T	[K]	Temperature
t	[s]	Time
τ	[m]	Wall thickness
U	[W/m^{2}K]	Overall heat transfer coefficient
$\dot{V}$	[m^3/s]	Volumetric flow rate
v	[m/s]	Velocity

Special characters

ρ	[kg/m^3]	Density
ΔT_{lm}	[K]	Logarithmic mean temperature difference

Subscripts

st	Storage tank water
in	Tube inlet
out	Tube outlet
t	Tube side

2. METHOD

2.1. Design of the heat exchanger

The helical coil as the plastic heat exchanger was designed for a pressure-free hot water storage tank ROTEX Sanitube INOX, and used to replace the original inner metallic corrugated tube. A helical coil made of plastic tube was designed using the mathematical model by Sharif *et. al.* (2016) assuming that the designed thermal output of the plastic tube should be same as the previously tested metallic tube. For the dimensions of the coil, namely the coil diameter D_c and the coil height H_c, the size of a vertical body of the tank (inside vessel dimensions 138x48x48 cm) was the limiting factor.
The design challenge is to overcome the resistance across the polymer wall with a poor conductivity. To achieve the same heat transfer performance, expressed by Eq. (1), where the thermal conductivity λ of the plastic is much lower, ratio of the outside/inside diameter, has to be decreased or the length of the tube has to be increased. Choosing smaller diameter also helps to improve plastic heat exchanger characteristics (Zaheed and Jachuck, 2004). When decreasing the wall thickness, a sufficient strength of material to withstand pressures and mechanical manipulation needs to be respected.

$$U_o A_o = \frac{1}{R_o + R_W + R_i} = \frac{1}{\frac{1}{h_o A_o} + \frac{ln\left(\frac{d_o}{d_i}\right)}{2\cdot\pi\cdot\lambda\cdot L} + \frac{1}{h_i A_i}} \qquad (1)$$

By increasing the length of the tube and therefore the surface area, the disadvantage of the low thermal conductivity can be eliminated. Using the model of Sharif *et. al.* (2016), when choosing a plastic tube of half outer diameter and half wall thickness compared to the metallic tube, the required length is 5 times larger for achieving the same thermal performance.

2.2. Experimental set-up

The experimental set-up consists of an open cold water cycle and a closed hot water cycle. The storage capacity of the tank is 300 l and it has a double walled jacket made of polypropylene with PUR hard foam heat insulation. The studied heat exchanger is shown in Fig. 1. It is a simple flexible floor heating pipe made of PEX with aluminum layer inside polymer matrix, available on the market. The tube outside diameter is 16 mm and a wall thickness is 2 mm. To achieve the maximal surface area, two parallel helical coils next to each other with diameters 30 cm and 26 cm and pitch 3.5 mm were attached to the circular frame. In such construction, 97 m of the plastic pipe was attached inside, supported in the fixed position in the tank by metal slats with plastic mounts.

The helical coil construction is placed to the middle of the ROTEX hot water storage tank on the support 4 cm from the bottom and 4 cm from the top to ensure that there is definite temperature stratification within a storage tank. The tank was closed with only tubes inlets/outlets and thermocouples connections.

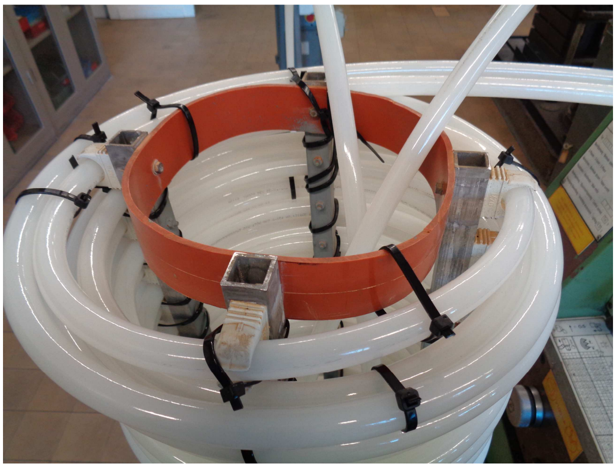

Figure 1. Picture of the studied helical coil heat exchanger.

A water supply to the plastic tube is a tap water, controlled by the valve and a Magnetic inductive flowmeter KROHNE OPTIFLUX 4300 C with accuracy less than 0.5 % of measured value for flow rate bigger than 2.5 L/min. The temperature is measured at the inlet and outlet of the tube. Two pressure sensors, UNIK 5000 with accuracy 0.2% of full scale, were installed at the inlet and outlet of the tube to determine differential pressure drop.

To provide heating, a closed water cycle is used. The heater has a maximal power of 9 kW and is equipped with PID controller, implemented via LabVIEW. The water pump has relay control and a motorized three way valve to control the closed circle mass flow rate to the solar boiler, keeping this below 50% of the maximal flow rate of the pump. The water mass flow rate through the heat exchanger is measured by a Coriolis mass flow meter (PROMASS 80-Endress + Hauser). Based on the calibration sheet, the relative error on the mass flow rate is less than 0.15% of full scale. The temperatures are measured at the inlet and at the outlet of the tank and at fifteen points inside the tank to determine stratification, where 9 thermocouples are attached to the tube going vertically in the middle of the tank with distance 15 mm between each other and the first thermocouple at the point of water inlet and 6 thermocouples in three points on both side of the coil. Such, the heat exchange can be better measured and controlled.

All water thermocouples are measured with K-type thermocouples, which were calibrated for the specific measuring range using a Duck DBC150 temperature calibrator furnace. The total absolute uncertainty of each thermocouple is around ±0.2 K. The pressure drop over the heat exchanger was measured with a pressure sensor UNIK 5000 with relative uncertainty 0,2 %. A summary of measurement equipment and their uncertainties can be found in Table 1.

Table 1. Measurement equipment and their accuracies

Measurement equipment	Accuracy
Magnetic inductive flowmeter (%) of measured value	± 0.5
Coriolis mass flow meter (%) of FS	± 0.1
Silicon pressure sensor 0-4 bar (%) of FS	± 0,2
K-type thermocouple (°C)	± 0,2

2.3. Measurement procedure

All measured values are acquired using a National Instrument Data Acquisition System. The water in the tank was first heated to the determined temperature by closed circuit, then the pump and heater was turned off and the water inside the tank was let to reach an uniform stratification to ensure that the measurements were performed under steady state condition.

Afterwards the water from the tap was brought to the plastic tube and the heat exchange was performed till the water inside the tank decreased to in average 20°C. The temperature difference between the inlet and the outlet to the tube was measured. All measurements were scanned and recorded at a time interval of 1 min.

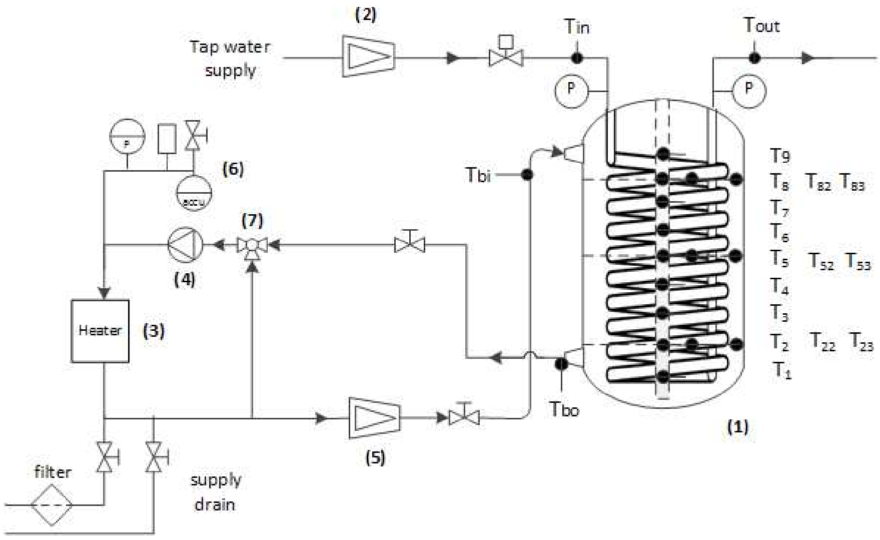

Figure 2. Experimental set-up: Water tank and closed hot water cycle: (1) hot water storage tank ROTEX, (2) Magnetic inductive flowmeter, (3) heater, (4) water pump, (5) Coriolis mass flow meter, (6) expansion tank, (7) three way valve.

2.4. Data reduction

With the measured inlet and outlet water temperatures and the total mass flow rate inside the tube, the heat transfer rate on the cold water side can be determined. The water properties in the data reduction process were obtained from Coolprop database.

$$\dot{Q}_t = \dot{m}_t \cdot c_{p,t}(T_{out} - T_{in}) \tag{2}$$

The thermal output *UA* is determined from the logarithmic mean temperature difference T_{lm} according to Eq. (5), where the correction factor is assumed to be the unity. In the Eq. (4) for T_{lm}, temperature T_b is a temperature inside the boiler calculated from the stratified temperature distribution. The temperatures in the three points beside the central line T_1-T_9 are taken in account where the thermocouples are placed on the both sides, from inside as well as from outside of the helical coil in the equal distance. The average of both temperatures is taken and the final temperature of the boiler is then the function of the volume weight of the corresponding segments of the storage tank, to which is this tank divided.

$$T_{st} = \frac{\left(H_1 \cdot \left(\frac{T_{22} + T_{23}}{2}\right) + H_2 \cdot \left(\frac{T_{52} + T_{53}}{2}\right) + H_3 \cdot \left(\frac{T_{82} + T_{83}}{2}\right)\right)}{H_1 + H_2 + H_3} \tag{3}$$

$$\Delta T_{lm} = \frac{(T_{st} - T_{in}) - (T_{st} - T_{out})}{\ln\left(\frac{T_{st} - T_{in}}{T_{st} - T_{out}}\right)} \tag{4}$$

$$U_o A_o = \frac{\dot{Q}_t}{\Delta T_{lm}} \tag{5}$$

The outside surface area was calculated from the outside diameters of the tube D_o= 16 mm and from the total length of the tube L, where possible decrease of the heat transfer due to the close contact of the tube coils were omitted. Therefore, the outside heat transfer area is quite high, A_o=4.88 m^2.

For control of the correctness of the results, the overall heat balance on warm and cold water side is determined based on the T_{in}, T_{out} and T_b. On the tube side, the heat flow is calculated according to Eq. (2), on the storage tank side according to Eq. (6) where the time step is taken as a difference between previous and following readings of one minute (the actual time needed for water to pass between inlet and outlet of the coil, $t=L/v_t$, is between 40s and 3.5 min).

$$\dot{Q}_{st} = \int_1^2 (V_{st} - V_t) \cdot \rho_{st} \cdot c_{v,st} \frac{dT_{st}}{dt} \tag{6}$$

To determine internal convection heat transfer coefficient, two different methods have been used, Wilson plot (Fernández-Seara *et al*., 2007; Rose, 2004) as a function of overall heat transfer resistance and reduced velocity ($1/v_r^n$) and Gnielinski correlation (Sharif *et. al.*, 2016, Kakac *et. al.*, 2012), leading to similar values. The heat resistance of the wall was calculated using thermal conductivity of the wall and the outer resistance using *UA* and two calculated heat resistances from Eq. (1).

3. RESULTS AND DISCUSSION

3.1. Uncertainty analysis

In order to be able to indicate the quality of the measurements, a thorough uncertainty analysis was performed according to Taylor (1997) using the overall uncertainty (root-sum-square method). The errors estimated on the thermodynamic properties of water were determined based on recommendations in open literature (Bell, *et. al.*, 2014) as following: Dynamic viscosity (1 %), Density (0.001 %), Specific heat capacity c_p (0.1%), Thermal conductivity λ (1.8 %). The error on the dimensions of the plastic tube measurements are ± 0.02 cm on inside/outside diameter and ±0.02 m for length of the tube and coil height/diameter and storage tank measurements. Table 6 presents the minimum, maximum, and average relative uncertainty for most of the calculated variables.

For all measurement performed in this study, the relative uncertainty on *Q* ranges from 0.8% to 2.5%, on T_{lm} from1.8% to 2.8%, on *UA* from 2.4% to 4.2%. The error on the dimensions of the plastic tube measurements are ± 0.01 cm on inside/outside diameter and ±0.05 m for length of the tube and coil height/diameter.

Table 2. Variable relative uncertainties: minimum, maximum, and mean values

Symbol	Uncertainty range (%)	Average uncertainty (%)
$\dot{Q}_t$	0.84-2.46	1.44
$\dot{Q}_{st}$	5.14-7.17	6.11
ΔT_{lm}	1.85-2.76	2.24
$U_o A_o$	2.42-4.25	3.11

3.2. Heat transfer

Experimental results were obtained for temperature inside the water tank 30, 40, 50 and 60°C for maximal flow rate (12 L/min) and for a small flow rate (5.5 L/min). The flow rates varied depending on actual tap water flow rate. For the high flow rate, the temperature difference between the inlet and outlet of the tube after 10 min of running each set of measurement was 5°C, 9°C, 12°C and 17.5 °C for initial temperatures 30°C, 40°C, 50°C and 60°C, respectively. At the Fig. 3 and Fig. 4 is shown the temperatures progress during the 1 hour set for 60°C initial temperature and 12 L/min, resp. 5.5 L/min flow rate. From the graphs it can clearly be seen that there is not a high stratification temperature difference between the thermocouples in the central line of the storage tank accept the top one which is close to the water level.

In Fig. 5 the comparison of the heat flow from the storage tank and into the tube is displayed. The heat transfer calculated within the solar boiler via change of mean temperature is seen as scattered data due to the fluid flow in the different spots in the enclosure due to natural convection. The difference between the mean value of the scatter and heat transfer calculated via the cooling water can be attributed to the distance between influence zone.

Figure 6 displays the overall heat transfer coefficient *U* for the different flow rates with the maximum flow rate determined by the flow rate of the tap water and minimum by the minimum opening of the valve. Within the uncertainty range, *U* follows the trend predicted by Sharif *et. al.* (2016), where we can observe sharp increase for very small flow rates and from the certain value change to constant or very slightly increasing value around 120 W/m^{2}K.

For the comparison with the original metallic tube, the thermal output *UA* was used where also the total surface area was taken into the consideration. An average value calculated from measurement for the PEX coil was *UA* = 500 W/K. The value stated for the original metal coil at product catalogue of Daikine (2008) is *UA* = 910 W/K.

Pressure drop in the helical coil was 0.6 bar for the high flow rate 12 L/min, 0.15 bar for the low flow rate 5.5 L/min.

From the heat transfer resistance ratio comparison (Fig. 7), it can be seen that both applied methods of internal convection coefficient calculation lead to the similar results and that in the case of the plastic tube the wall doesn't play the biggest role as the heat transfer resistance, although the effect of the wall resistance is much bigger than in the case of metallic heat exchanger.

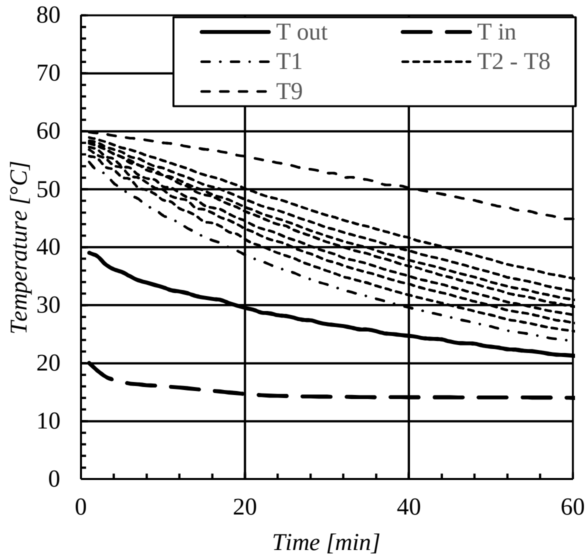

Figure 3. Temperature progress for 60°C initial temperature, 12 L/min, 1 h run.

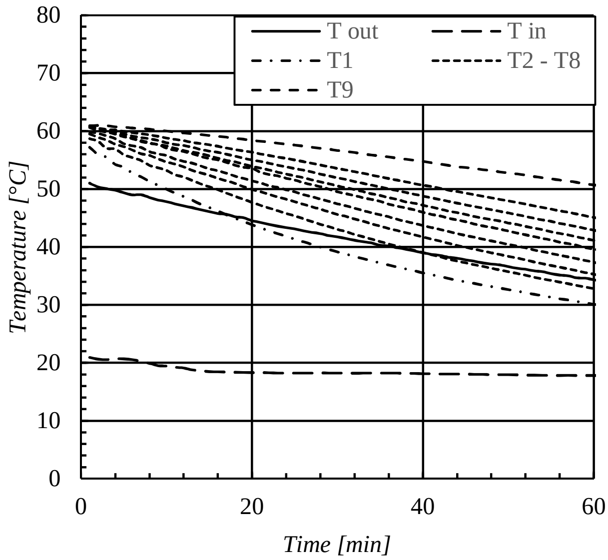

Figure 4. Temperature progress for 60°C initial temperature, 5.5 L/min, 1 h run.

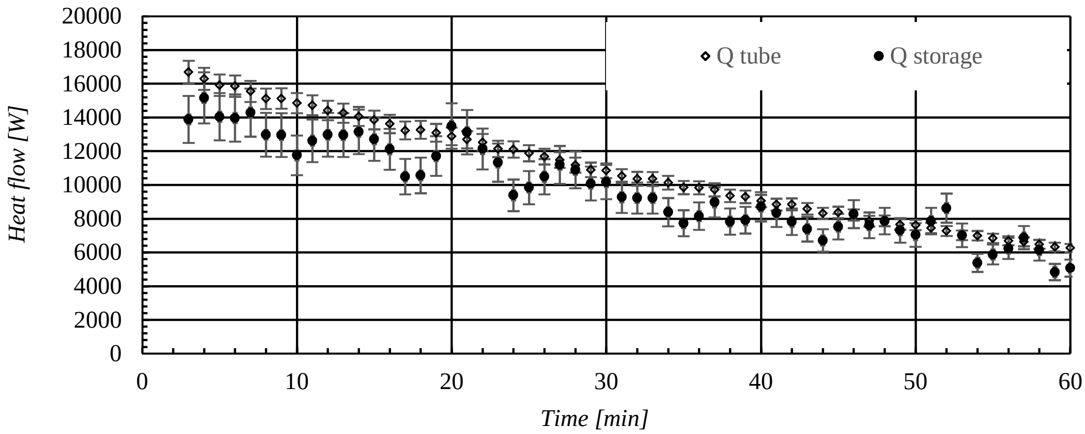

Figure 5. Thermo-balance between the storage water and the tube flow, 60°C initial temperature, 12 L/min, 1 h run.

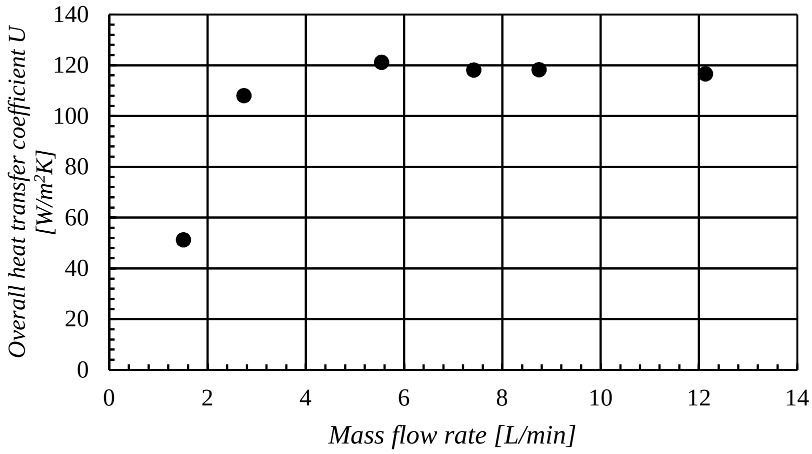

Figure 6. Overall heat transfer coefficient for different flow rates, 60°C initial temperature.

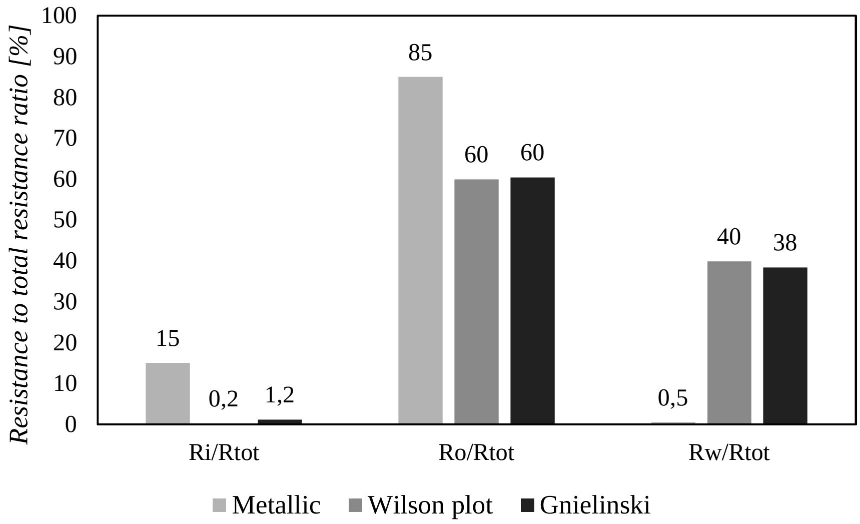

Figure 7. Heat transfer resistance ratio comparison for plastic tube (calculated by two different methods) and original metallic tube.

4. CONCLUSIONS

An experimental setup is fabricated to study heat transfer in a helical coil made of plastic inside a solar boiler water tank. The aim is better understanding of the thermal properties of plastic materials and their usage as heat exchangers to substitute classical ones made of metal. Temperature was measured temporally for different initial temperatures and mass flow rates. Using measured data, the overall heat transfer coefficient and average specific thermal output UA were calculated and later compared with the value stated in the product catalog of the original boiler with stainless steel corrugated tube. Wilson and Gnielinski method were used for determining heat transfer resistance values. The heat transfer resistance ratio comparison shows that the resistance of the outer side is of bigger significance than the resistance formed by plastic wall. For more precise understanding of the overall heat transfer coefficient behaviour, more a more precise method for controlling the flow rate should be employed.

5. ACKNOWLEDGEMENTS

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