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SIMULATION AND EXPERIMENTAL VALIDATION OF A HELICAL COIL USED AS CONDENSER IN A HEAT PUMP FOR DOMESTIC WATER HEATING IN A TANK

Piñeiro-Pontevedra C., Fernández-Seara J.*, Dopazo A. and Diz R.
*Author for correspondence
Área de Máquinas y Motores Térmicos, E.T.S. Ingenieros Industriales. Universidad de Vigo.
Campus Lagoas-Marcosende Nº 9,
36310. Vigo.
España,
E-mail: jseara@uvigo.es

ABSTRACT

Although the use of helical coils are widely used in engineering applications for heating and cooling production there is little information available about their thermal behavior. The purpose of this work was to simulate the behavior of a helical coil used as condenser in a heat pump for domestic hot water heating in a tank. The simulation was carried out by developing a mathematical model. The mathematical model was implemented in a computer program. The program results were validated against the experimental results obtained by using an experimental set-up. In the paper the mathematical model is detailed, the experimental setup and the experimental procedure are described and the results are presented and discussed.

INTRODUCTION

The use of helical coil heat exchangers (HCHE) as condensers in domestic solar water heating instalations has increased over the last years. Helical coils have a heat transfer coefficient greater than straight tube exchangers due to the curvature of the coil, which causes centrifugal forces that generate a secondary flow inside the tube that increases the turbulence [1-4]. Furthermore, at the coil´s outer surface the heated fluid around a turn tends to rise due to the increased buoyancy forces, affecting the upper coil [5].

Although there are many studies on heat transfer and pressure drop inside straight pipes, the number of studies for ducts with curvature and torsion is much more limited, as shown in the literature reviews conducted by Naphon & Wongwises [6] and Subhasnhini Vashisth et al. [7]. The number of researches found is even smaller with respect to the behaviour of the fluid and the heat transfer on the outer surface of the coil.

Finally the work performed by Wongwises & Srisawad [8] has been the only one found by the authors relating to finned helical coils. In that work, heat transfer characteristics of a helically coiled finned tube where hot water was used as the inner working fluid and ambient air on the outer side was studied experimentally.

Literature for inner convection with two-phase flows is also sparse and only for very specific cases.

NOMENCLATURE

A	[m^2]	Area
Cp	[kJ·kg^{-1}·K^{-1}]	Specific heat capacity at constant pressure
D	[m]	Coil diameter
DHW	[-]	Domestic hot water
d	[m]	Tube diameter
e	[m]	Thickness
FCE	[-]	Forced convection by elements
FCL	[-]	Forced convection by layers
g	[m·s^{-2}]	Gravity acceleration
Gr	[-]	Grashof number
$HCHE$	[-]	Helical coil heat exchanger
h	[W·m^{-2}·K^{-1}]	Convection heat transfer coefficient
I	[-]	Bessel function of the first kind
K	[-]	Bessel function of the second kind
k	[W·m^{-1}·K^{-1}]	Thermal conductivity
m	[kg·s^{-1}]	Massflow rate
NC	[-]	Natural convection
Nu	[-]	Nusselt number
P	[bar]	Pressure
Pe	[m]	Perimeter
p	[m]	Coil pitch
Q	[kW]	Heat transfer
R	[m]	Radius
Re	[-]	Reynolds number
r	[m]	Tube radius
T	[ºC]	Temperature
TEV	[-]	Thermostatic expansion valve
U	[W·m^{-2}K^{-1}]	Overall heat transfer coefficient
v	[m·s^{-1}]	Velocity
z	[-]	HCHE axis

Special characters

α	[º]	Helical angle
β	[1·K^{-1}]	Thermal expansion coefficient

η	[-]	Efficiency
μ	$[kg\cdot m^{-1}\cdot s^{-1}]$	Dynamic viscosity
π	[-]	Pi number
ρ	$[kg\cdot m^{-3}]$	Density
τ	[-]	Stress tensor
θ	[º]	Control volume angle

Subscripts

c	Coil
crit	Critical
eq	Equivalent
exp	Experimental
F	Forced convection
f	Fin
fi	Inner foul
fo	Outer foul
i	Inner
if	HCHE inner fluid
m	Mean
mod	Model
N	Natural convection
o	Outer
of	HCHE outer fluid
st	Storage tank
t	Tube
w	Water in the tank

In this research, the main objective was to develop a detailed simulation model in order to predict and evaluate the performance of a HCHE working as a condenser placed into a water storage tank. The numerical model and its implementation are detailed. The model results are compared to the experimental data collected from our own experimental facility evaluated under several operating conditions. The experimental set-up and the experimental procedure are also described. Finally, results are shown and discussed.

MATHEMATICAL MODEL

The numerical model was developed in order to predict the heat transfer phenomena in a HCHE located inside a water storage tank, commonly used in domestic hot water (DHW) systems. Convection heat transfer in the tube, conduction through the tube wall and convection heat transfer to the fluid stored in the tank were considered as the dominant heat transfer modes. The HCHE was modelled as an helically coiled tube divided into several small control volumes. The control volumes are connected in the HCHE's inner fluid flow direction. Each control volume is composed from a portion of tube defined as a model parameter, and a corresponding tank volume portion, as show in Figure 1.

Three systems are considered at each control volume, i.e. "inner fluid", "tube" and "outer fluid". The inner and outer fluid flows were R134a and water respectively.

The model equations were formulated from the mass, energy and momentum balances applied to each system in each control volume. In order to formulate the model, the following assumptions were applied; water is modelled as an incompressible fluid, the fluid and tube time-dependency physical properties are neglected (stationary processes), the inner fluid flow is unidirectional, i.e. the trajectory of all inner fluid particles are parallel to the tube wall; the outer water flow is also unidirectional, the physical properties of the inner fluid flow and tube wall are considered uniform in the tube radial direction, the axial heat conduction in the tubes and the variations of the inner flow's potential and kinetic energies are considered insignificant; the pressure drop in the water flow is neglected, the storage tank is isolated.

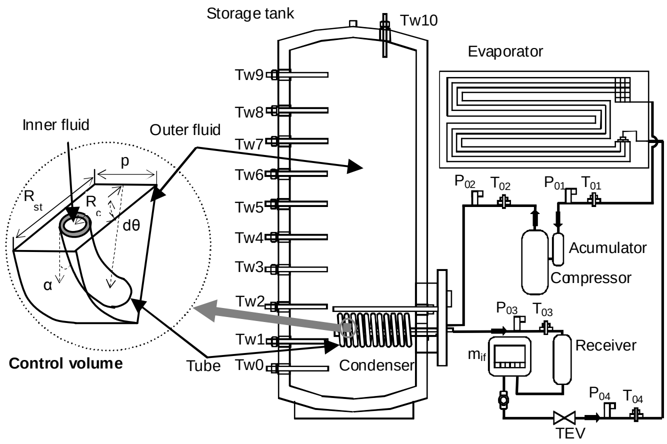

Figure 1 Schematic diagram of the established control volume and the experimental facility.

The mass, energy and momentum balances in each system yield Equations. (1) to (4), according to the nomenclature in Figure 1.

Inner fluid system for single-phase condition:

$$-\frac{\partial P}{\partial \theta} = \rho_{if}\cdot g\cdot R_c\cdot \sin\alpha + \tau\cdot R_c\cdot \frac{Pe}{A_{ti}} \tag{1}$$

$$v_{if}\cdot\rho_{if}\cdot\frac{1}{R_c}\cdot\frac{\partial h_{if}}{\partial\theta} = -U_i\cdot\left(T_{if}-T_t\right)\frac{4}{d_i} \tag{2}$$

Outer fluid system:

$$\frac{1}{2}\left(R_{st}^2\cdot p-\frac{\pi\cdot d_o^2}{2}\cdot R_c\right)\cdot v_{of}\cdot\rho_{of}\cdot Cp_{of}\cdot\frac{\partial T_{of}}{\partial z} = -U_o\cdot\pi\cdot d_o\cdot R_c\left(T_t-T_{of}\right) \tag{3}$$

Tube system;

$$Q = U_o\cdot d_o\cdot\left(T_t-T_{of}\right) = U_i\cdot d_i\cdot\left(T_{if}-T_t\right) \tag{4}$$

And the inner fluid system pressure drop on two-phase flow condition are calculated by using Friedel's correlation [9].

If the HCHE is finned, the efficiency of the finned surface is calculated in equations (5), (6), (7) and (8), where A_f is the fin area, A_T is the tube area without fin and η is the fin efficience.

$$\eta_{sup} = 1-\frac{A_f}{A_T}\cdot(1-\eta) \tag{5}$$

$$\eta = \frac{d_o}{M_T\cdot\left(r_{eq}^2-r^2\right)}\left[\frac{K_1(M_T\cdot r_o)I_1(M_T\cdot r_{eq})-K_1(M_T\cdot r_{eq})I_1(M_T\cdot r_o)}{K_1(M_T\cdot r_{eq})I_0(M_T\cdot r_o)-K_0(M_T\cdot r_o)I_1(M_T\cdot r_{eq})}\right] \tag{6}$$

$$M_T = \sqrt{\frac{2 \cdot h_o}{k_f \cdot e_f}} \tag{7}$$

$$r_{eq} = \frac{d_f}{2} + \frac{e_f}{2} \tag{8}$$

The inner and outer overall heat transfer coefficients are determined according to Eqs. (9) and (10), respectively.

$$U_i = \frac{1}{\frac{1}{h_i} + \frac{1}{h_{fi}} + \frac{r_i \cdot ln\left[\frac{r_m}{r_i}\right]}{k_t}} \tag{9}$$

$$U_o = \frac{1}{\frac{r_o \cdot ln\left[\frac{r_o}{r_m}\right]}{k_t} \frac{1}{h_o} + \frac{1}{h_{fo}}} \tag{10}$$

Correlations found in technical literature were used in the developed model to obtain the inner convection heat transfer coefficient and friction factor, as well as the outer convection heat transfer coefficients.

The inner convection heat transfer coefficient of the inner fluid is calculated separately for laminar, transitional and turbulent flow. The flow is classified based on the Reynolds number. The transitional regime boundaries is determined from the critical Reynolds number, equation (11). The transition region was determined between Re_{crit} and Re<22000.

$$Re_{crit} = 2300 \left[1 + 8.6 \cdot \left(\frac{d_i}{D_c} \right)^{0.45} \right] \tag{11}$$

The inner heat transfer coefficient of the fluid in the coil is obtained according to Smith's correlation [10] for laminar regimen, Petukhov's correlation [11] for complete turbulent flow (Re>22000) and Gnielinski's [12] correlation was used for the transition region. There are few correlations to calculate the inner convection coefficient for two-phase flow in helical coils. And these are not in line with the working conditions of this study. Therefore the inner convection coefficient for two-phase flow is obtained by the Travis et al. generic correlation [13].

The heat transfer convection coefficient on the outer surface calculation is very complex and no publications have been found that propose a general correlation to determine the convection coefficients on the outer surface of the coil. Therefore, the calculation of this coefficient is proposed by three different calculation methods called; natural convection (NC), forced convection by elements (FCE) and forced convection by layers (FCL). The first involves natural convection on the outer side of the coil so that the temperature of the surrounding water is constant and equal to the initial value. In the other two cases the movement of water and its temperature increases are taken into account. In the FCE method it is considered that the water flow temperature of the control volume increases with respect to the neighboring control volume located below. In the FCL method, the water surrounding the HCHE is divided into layers as a function of turns of the coil, and it is assumed that the water temperature is homogeneous at each layer.

The calculation method is defined based on the ratio M, determined in equation (12) as a function of Reynolds and Grassoff numbers.

$$M = \frac{Gr}{Re^2} \tag{12}$$

When the value of M is greater than 1, it is considered that the forced convection is negligible, ie**,** the fluid motion takes place due to differences in density, whereas if its value is less than unity, then it is the forced convection that is considered. A mixed convection zone when $0.8 < M < 1.2$ is defined.

As the HCHE studied is horizontal, each turn was divided into three areas to calculate the outer convection coefficient on natural convection. Area type 1 is considered as horizontal tube and Churchill & Chu's correlation [14] has been used to obtain the convection coefficient. Areas type 2 and 3 are considered as inclined and vertical tube respectively. Correlations to calculate natural convection on inclined and vertical tube have not been found. So the outer convection coefficient of areas type 2 and 3 were calculated by using Churchill & Chu's correlation [15] for vertical plate. Zhukauskas's correlation [16] has been used to obtain the outer convection coefficient on FCE and FCL methods. For the mixed convection zone, Churchill's [17] proposed equation (13) has been used, where the Nusselt number is calculated as a function of the natural and forced convection Nusselt numbers.

$$Nu^3 = Nu_N^3 + Nu_F^3 \tag{13}$$

Model implementation

A finite difference approach was used to solve the model equations. The system of discretized equations was solved in space, step by step, beginning with the control volume where the inner fluid flow enters the HCHE. From the known values at the inlet section, the values of the variables at the outlet of each control volume are iteratively obtained, advancing in the flow direction. The procedure is repeated until the last control volume is reached. The inlet temperature, pressure and mass flow of the inner fluid, the initial tube and outer fluid properties, and the HCHE and tank geometries, are used as inputs.

A computer code was developed for the model implementation by using Visual Basic .Net. From the software results, the inner and outer fluid outlet temperatures and the HCHE heat transfer rate are determined as well as other important parameters such as the inner pressure drop, the coil temperature distribution, inner and outer overall heat transfer coefficients.

The calculation process is as follows:

1. The initial values determine the inner fluid flow thermodynamic properties at the HCHE inlet and the water properties in the tank.
2. For each control volume at each turn:

2.1. Calculate the heat flux as follows:
   2.1.1. Guess an inside tube wall temperature, T_{ti}.
   2.1.2. Calculate the inner convection coefficient h_i.
   2.1.3. Calculate the heat flux between inner fluid and tube wall, $Q_{if\text{-}ti}$.
   2.1.4. Calculate the outside tube wall temperature, T_{to}.
   2.1.5. Calculate the outer convection coefficient, h_o, distinguishing between natural, forced or mixed convection.
   2.1.6. Calculate the overall heat transfer coefficient, the heat flux between inner coil fluid and water in the tank, $Q_{if\text{-}w}$.
   2.1.7. Calculate the inner tube wall temperature, T_{tiq}.
   2.1.8. Check the guessed inner tube wall temperature, T_{ti}, and the calculated inner tube wall temperature, T_{tiq}. If verified, continue, otherwise guess a new value and return to step 2.
2.2. Determine the inner fluid flow loss pressure.
2.3. Calculate the thermodynamic properties of inner fluid at the control volume outlet, which will be used as input for the following control volume.
3. The process is repeated for each control volume.

The water thermodynamic properties are obtained from the Refprop Database [18].

EXPERIMENTAL SETUP AND DATA REDUCTION

An own experimental facility was used to obtain the experimental data in order to validate the model results.

A cooper finned HCHE of 12.4/10.8 mm tube's outer/inner diameters, 12 turns, 22 mm pitch and 120 mm coil diameter placed into a water storage tank was tested. The fin diameter is 19 mm, with 3 mm step and 0.5 mm thickness.

The schematic diagram of the experimental facility is shown in Figure 1. The facility is composed of a 300-litre hot water storage tank with an integral helical coil heat exchanger as condenser, a thermostatic expansion valve (TEV), a R134a hermetic compressor and a bare solar collector as evaporator. The storage tank consists of a vertical cylindrical body and caps made of 2 mm thick stainless steel (AISI 316L) sheets. The height and diameter are 1515 mm and 480 mm, respectively. The storage tank is insulated with 30 mm thickness polyurethane foam (type S Spray). The insulation on the lateral, top and bottom surfaces is covered with high impact polystyrene white lacquer externally plain sheets. The integral condenser is placed horizontally in the bottom of the storage tank as shown in Figure 1. The TEV includes interior pressure equalization. A R-134a rotary-type hermetic compressor with rated input power 390-550 W and displacement volume of 2.1228 $m^3 \cdot h^{-1}$ is used. The solar collector dimensions are 2000 mm length and 800 mm wide, and they are made of two aluminium sheets with pressed dual-channel through the roll-bond process, including an anodic oxidation after pressing that gave them an external black colour. A detailed description of the experimental setup and methodology can be found in Fernández-Seara et al. [19].

The experimental facility was equipped with a data acquisition system based on a 16-bits data acquisition card and a PC. Inside the hot water storage tank, the water temperature distribution is measured by using eleven temperature sensors. The sensors were inserted from the lateral surface and, with the exception of sensors T_{w0} and T_{w10}, were distributed uniformly at every 150 mm interval along the height of the tank. According to Figure 1, sensor T_{w0} is located at 95 mm from sensor T_{w1} and at 80 mm from the bottom of the tank. Sensor T_{w10} is located at the cap of the tank. All sensors are A-Pt100 type, inserted in 190 mm long and 6 mm diameter stainless steel pockets.

In the heat pump system, the suction and discharge temperature of the R134a compressor unit are measured by using two sensors located in the refrigerant suction and discharge lines, close to the compressor (T_{01} and T_{02}, in Figure 1). The condensed R134a temperature is measured by using the sensor T_{03} (refer Figure 1), located in the liquid receiver outlet line. The evaporating temperature is measured by using the sensor T_{04}, located in the inlet line of the solar collector (evaporator). The suction and discharge pressures are measured by using the pressure transducers P_{01} and P_{02} installed in the compressor suction and discharge lines, respectively, also close to the compressor. In addition, the condenser and evaporator pressures are measured by using the pressure transducers P_{03} and P_{04}, located in the TEV and solar collector inlet lines, respectively (refer Figure 1). The R134a mass flow rate is measured by using a Coriolis-type mass flow meter located in the liquid receiver outlet line. Compressor power consumption is measured by using a network analyzer. All the temperature sensors used in the heat pump system are also A-Pt100, inserted in the pipelines by using 80 mm long and 3 mm diameter stainless steel pockets.

Tests consisted on collecting measurements during the heating process water in the tank. So the following experimental procedure was carried out. First the tank is emptied and filled with cold water. The heat pump is turned on so that the refrigerant starts to flow into the HCHE and the water in the tank is heated. Once the temperature of the tank reaches the desired value, the experimental data are collected for operating conditions during 10 minutes at intervals of 15 seconds.

RESULTS

Although the tests have been conducted in natural convection, without water circulation in the tank, the validation has been performed for the three different calculation methods. Therefore, a very low water circulation, about 0.006 m/s for FCE and FCL calculation methods, was used. The numerical results were compared to the data experimentally obtained. A resume of the comparison between numerical model and experimental results is shown as follows.

Figure 2 shows experimental and estimated condensation power as a function of the water tank temperature for one completed heating test. The condensation power was estimated by using the three types of calculation method, NC, FCL and FCE. It can be appreciated that the results of NC method are lower than the experimental data while the FCL and FCE results are closer to actual data. Furthermore, the NC values approach the experimental values as the test progresses and are closer to actual values than those obtained with FCL and FCE methods.

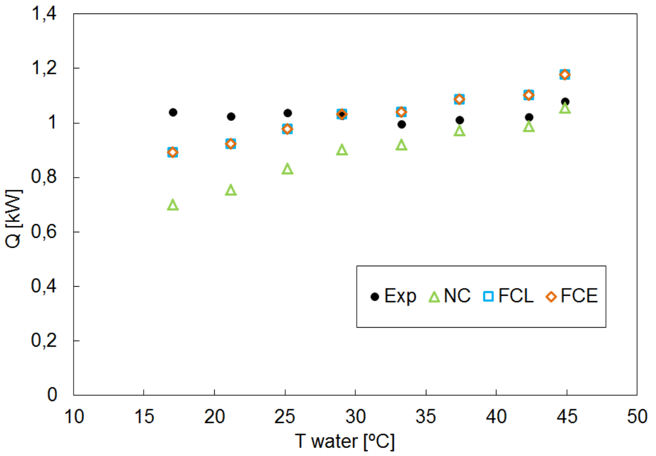

Figure 2 Experimental and numerical condensation power vs water tank temperature.

Figure 3 shows the error in the estimation of the refrigerant temperature at the output of the HCHE for several tests. The error value obtained by the NC method is less than ±3% in most of the cases. The numerical temperatures obtained with the other two calculation methods are mainly lower than the experimental ones and with an error less than 25%. The greater errors were observed at the end of the test when the water tank temperature was higher.

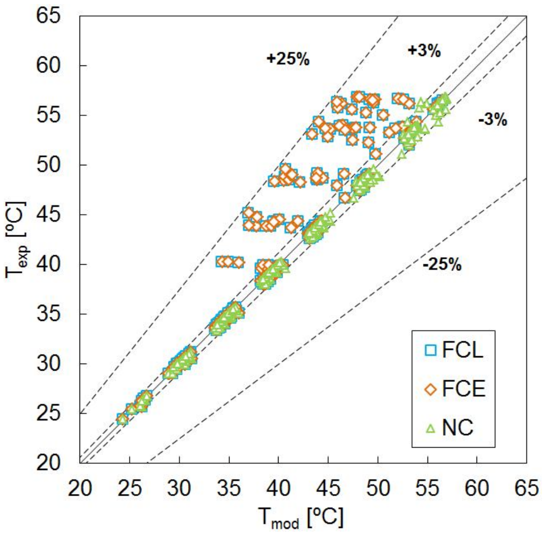

Figure 3 Error in estimating the refrigerant temperature at the output of the coil

Figure 4 shows that the error in estimating the pressure drops of the refrigerant at the condenser outlet is approximately ± 2%. According to Figure 4 there are no differences between the results of the different calculation methods.

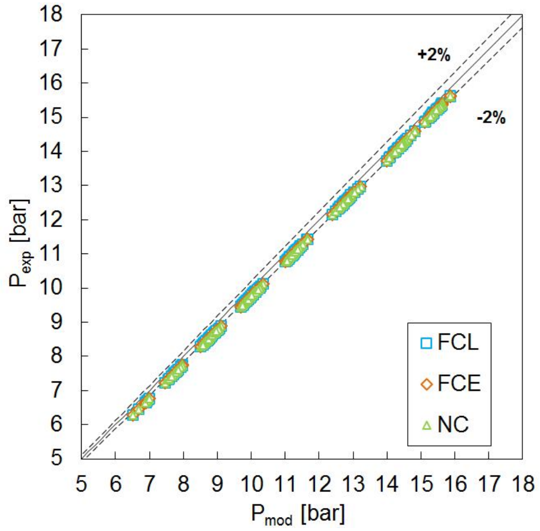

Figure 4 Error in estimating the refrigerant pressure drop at the output of the coil

On the other hand, it can be observed in Figure 2, Figure 3 and Figure 4 that FCL and FCE results are very similar and no differences have been appreciated.

The error in estimating the variation of the water tank temperature with FCL and FCE calculation methods, considering a little velocity of water, is ±1%, as is shown in Figure 5.

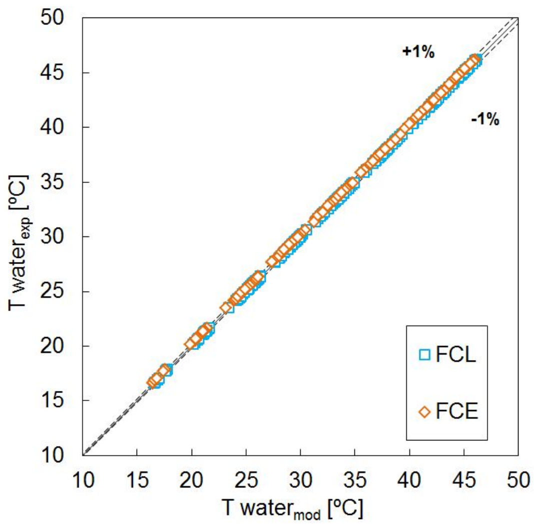

Figure 5 Error in estimating the water tank temperature with FCL and FCE calculation methods.

Therfore, if the three calculation methods are combined, modeling the first part of the test with NC and the latter part of the test with FCL or FCE, error values of ±15% of condensing power are obtained.

CONCLUSIONS

A numerical model was developed in order to predict the heat transfer phenomena and pressure drop in an HCHE working as a condenser of a heat pump and located inside a fluid storage tank. The model equations were formulated from the mass, energy and momentum balances. Several correlations found in technical literature were used in the model which was developed to obtain the inner convection heat transfer coefficient and friction factor, as well as the outer convection heat transfer coefficients. The model was validated with experimental data obtained from an own experimental facility with a finned HCHE working as condenser with R134a.

The results show that error values around ±15% of condensating power were obtained. The error on estimating the outlet refrigerant temperatrue and pressure drops were less than ±3% and ±2% respectivily.

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