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PREDICTION IN OFF-DESIGN OPERATION FOR THE HELICAL HEAT RECOVERY EXCHANGER

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ABSTRACT

Helical heat exchangers are commonly used to exchange the thermal energy in waste heat recovery systems. Ammonia rectifier in absorption chiller and heat recovery steam generator are examples typically found in open literature. They are widely employed because of its compactness, high heat exchange rate, compensation of thermal expansion, vibration resistance, simple construction, and low capital cost. Heat exchanger researches have almost focused on straight tubes. However, study on helical heat exchanger has not been paid attention. In this study, a simplified model is developed to predict single phase heat transfer and pressure loss in the helical heat exchanger under various operating conditions without geometrical information inside the exchanger required. Methods of LMTD and ε-NTU, and selective empirical correlations are presented in the model. Simulation results were achieved good agreement with the experimental data with a moderate tolerance. The model would be expected as a good tool for designers to the pre-design and correct selection of helical heat exchanger in thermal network.

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1 INTRODUCTION

Heat exchangers are core components of thermal systems. Their improvements will allow for efficient use of energy. Therefore, researches on the heat exchangers are often paid attentions and published with a high density for the past few decades. There are two problems which are often mentioned in previous studies. Those are to enhance heat transfer rate of heat exchangers (Vitillo *et al*., 2015; Yujie *et al*., 2015) and predict off-design conditions of the available heat exchangers (Kayansayan, 1989; Rovira *et al*., 2011). Mostly the studies were concentrated on the straight tube heat exchangers as confirmed (Wongwises and Polsongkram, 2006). However, the helical coil heat exchangers show dominant advantages in comparison with the straight tube heat exchanger. Prabhanjan *et al*. (2002) showed experimentally that heat transfer rate of helical heat exchanger is more than that of straight tube heat exchanger due to centrifugal forces to act on the moving fluid, causing the formulation of secondary flow. Besides, helical capillary in refrigeration system has length shorter 14% than straight capillary as showed (Zareh *et al*., 2014). Furthermore, the helical heat exchangers have compactness, compensation of thermal expansion, vibration reduction, easy construction and low capital cost. In recent years, there has been a remarkable consideration on applications of helical heat exchanger for thermal systems. Seara *et al*. (2003) formed an analytical model to investigate the helical coil rectifier in an ammonia–water absorption chiller. Xiaowen and Lee (2009) studied experimentally the helical heat exchanger for heat recovery air-conditioners. Sogni and Chiesa (2014) developed a model to calculate heat recovery boiler using helical tube. Also, helical heat exchangers are regularly used for the liquid-to-suction heat exchanger in refrigeration cycles (Stoecker and Jones, 1982).

Generally, the previous studies are to find out the characteristics and design of helical heat exchangers. But estimation of off-design conditions (i.e. temperature, pressure drop, thermal duty) of an available helical heat exchanger has not been noted. In order to estimate those conditions, the geometrical parameters inside the exchanger should be given and inputted to heat transfer and pressure drop models. Unfortunately, the geometrical parameters are sometimes missed from manufacturer. Few parameters are known from manufacturer's catalogue. This causes obstacles in prediction of operating conditions different from design conditions. In practice heat exchangers usually run in part-load or overload modes. To overcome such a difficulty, Garcia *et al*. (2010) developed a model for the straight tube condenser and evaporator of refrigeration system. Errors of the predicted temperatures and capacities are from ±1 to ±7 % in comparison to the measured values. However, the pressure drop model is somewhat complicated and geometry of tube bundles has to be known. Furthermore, experimental validation of the pressure drop model was not performed. In this paper, a similar model to that of Garcia *et al*. (2010) is formed for the helical heat exchanger. Results of pressure drop are also presented both the modelling and experimental approaches.

2 MODEL FORMULATION

2.1 Heat transfer model

Figure 1 presents a schematic of a helical heat exchanger. A fluid is traveling inside a helical tube. Another fluid is passing across the helical tube. The fluids carry different thermal energies therefore heat is transferred from hot fluid to cold fluid through the surface of helical tube. General ideal of the mathematical model can be seen in report of Garcia *et al*. (2010). Some equations are presented here for the sake of easy understanding. Overall conductance *UA* of a heat exchanger can be written as below equation if fouling and wall resistances are neglected:

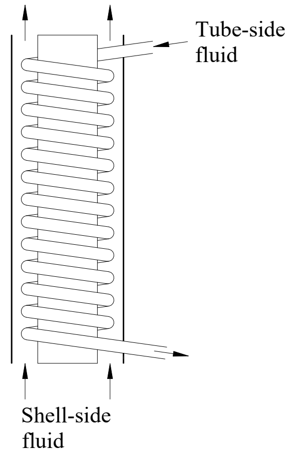

Fig. 1: Helical coil heat exchanger

$$\frac{1}{UA} = \frac{1}{h_i A_i} + \frac{1}{h_e A_e} \tag{1}$$

where h and A are heat transfer coefficient and area, respectively. Subscripts "i" and "e" respectively represent inner and outer surfaces.

The above equation can be rewritten as follows:

$$\frac{1}{UA} = \frac{h_e A_e + h_i A_i}{h_i A_i h_e A_e} \tag{2}$$

Let the subscript "*ref*" be reference parameters corresponding to known conditions. The known conditions, for example, can be obtained from manufacturer's catalogue or experiment. From Eq. (2) a ratio can be made between the operating conditions and the reference conditions of the same heat exchanger as follows:

$$\frac{UA}{UA_{ref}} = \frac{h_i h_e}{h_{i,ref} h_{e,ref}} \frac{h_{e,ref} A_e + h_{i,ref} A_i}{h_e A_e + h_i A_i} \tag{3}$$

We can define ratios as:

$$\beta_i = \frac{h_i}{h_{i,ref}} \tag{4}$$

$$\beta_e = \frac{h_e}{h_{e,ref}} \tag{5}$$

In heat transfer design, thermal resistances should be equal in order to gain an optimum design. Thus an approximation can be done as the following equation:

$$h_{e,ref} A_e = h_{i,ref} A_i \tag{6}$$

Since the Eq. (3) can be rewritten as:

$$\frac{UA}{UA_{ref}} = \beta_i\beta_e \frac{2h_{i,ref}A_i}{h_{e,ref}\beta_e A_e + h_{i,ref}\beta_i A_i} = \frac{2\beta_i\beta_e}{\beta_e+\beta_i} \quad (7)$$

The heat transfer coefficient for the fluid flowing inside helical tube can be computed by means of the Rogers and Mayhew's correlation (Rogers and Mayhew, 1964) as:

$$h_i = 0.023\mathrm{Re}^{0.85}\,\mathrm{Pr}^{0.4}\left(\frac{d}{D}\right)^{0.1}\frac{k}{d} \quad (8)$$

for Re ≤ 50000 (Hardik *et al.*, 2015) .

where Reynolds number and Prandtl number are, respectively:

$$\mathrm{Re} = \frac{\dot{m}d}{A_c\mu} \quad (9)$$

$$\mathrm{Pr} = \frac{c_p\mu}{k} \quad (10)$$

D and d are respectively the outer and inner diameters of the helical tube. $\dot{m}$, c_p, k, and μ are mass flow rate, specific heat, thermal conductivity, and dynamic viscosity, respectively.

Therefore, h_i can be rewritten as follows:

$$h_i = (0.023)\left(k^{0.6}\mu^{-0.45}\dot{m}^{0.85}c_p^{0.4}\right)\left(D^{-0.1}d^{-0.05}\right) \quad (11)$$

Heat transfer and pressure drop of cross-flow straight tube bundles can be used to model helical heat exchanger as shown in previous studies (Sogni and Chiesa, 2014; San *et al.*, 2012). Therefore, the shell side heat transfer coefficient is obtained from the Zukauskas's correlation (1000<Re<200000) for in-line tube bundles (Cengel, 2003):

$$h_e = 0.27\mathrm{Re}^{0.63}\,\mathrm{Pr}^{0.36}\left(\frac{\mathrm{Pr}}{\mathrm{Pr}_w}\right)^{0.25} F_N\frac{k}{d} \quad (12)$$

where F_N is a correction factor whose values are dependent on number of row of tube bundle.

Neglecting the influence of temperature-dependent properties, i.e. $(\mathrm{Pr}/\mathrm{Pr}_w)^{0.25} = 1$, the coefficient h_e can be rearranged by:

$$h_e = (0.27)\left(k^{0.64}\mu^{-0.27}\dot{m}^{0.63}c_p^{0.36}\right)\left(F_N d^{-0.37}\right) \quad (13)$$

As can be seen in the above equations the heat transfer coefficients are functions of three terms including constant coefficient, properties of fluid, and geometry of helical heat exchanger. The terms of constant coefficient, and geometry will be eliminated in the ratio of heat transfer coefficients.

Thus, the ratios of tube side and shell side heat transfer coefficients are given by, respectively:

$$\beta_i = \frac{h_i}{h_{i,ref}} = \left(\frac{k}{k_{ref}}\right)^{0.6}\left(\frac{\mu}{\mu_{ref}}\right)^{-0.45}\left(\frac{\dot{m}}{\dot{m}_{ref}}\right)^{0.85}\left(\frac{c_p}{c_{p,ref}}\right)^{0.4} \quad (14)$$

$$\beta_e = \frac{h_e}{h_{e,ref}} = \left(\frac{k}{k_{ref}}\right)^{0.64}\left(\frac{\mu}{\mu_{ref}}\right)^{-0.27}\left(\frac{\dot{m}}{\dot{m}_{ref}}\right)^{0.63}\left(\frac{c_p}{c_{p,ref}}\right)^{0.36} \quad (15)$$

The effectiveness and number of transfer unit (ε-NTU) relation of the helical heat exchanger is similar to that of a cross-flow heat exchanger (with one fluid mixed and the other unmixed) if number of turns of helical tube equal or more than six as pointed out (San *et al.*, 2012). Therefore, the relation is:

$$\varepsilon = \frac{1}{C}\left\{1-\exp\left[-C\left(1-\exp(-NTU)\right)\right]\right\} \quad (16)$$

C_{max} mixed, C_{min}unmixed

$$\varepsilon = 1-\exp\left\{-\frac{1}{C}\left[1-\exp(-NTU.C)\right]\right\} \quad (17)$$

C_{max} unmixed, C_{min} mixed

$$C = \frac{C_{\min}}{C_{\max}} \quad (18)$$

where C_{min} and C_{max} are the smaller and the larger of $\dot{m}_i c_{p,i}$ and $\dot{m}_e c_{p,e}$, respectively.

2.2 Pressure drop inside helical tube

Pressure drop inside a tube of length L and inner diameter d is given by:

$$\Delta p = 2f\frac{L}{d}\frac{\dot{m}^2}{\rho A_c^2} \quad (19)$$

The Fanning friction factor f inside a helical tube can be used correlation of Srinivasan *et al.* (Kakaç and Liu, 2002) as follows:

$$f\left(\frac{2R}{d}\right)^{0.5} = 0.084\left[\mathrm{Re}\left(\frac{2R}{d}\right)^{-2}\right]^{-0.2} \quad (20)$$

for $\mathrm{Re}\left(\frac{2R}{d}\right)^{-2} < 700$ and $7 < \frac{2R}{d} < 104$.

where R is curvature radius of helical coil.

Similar to Eqs (14) and (15), pressure drop ratio of operating conditions to reference conditions can be correlated as:

$$\frac{\Delta p}{\Delta p_{ref}} = \left(\frac{\mu}{\mu_{ref}}\right)^{0.2} \frac{\rho_{ref}}{\rho} \left(\frac{\dot{m}}{\dot{m}_{ref}}\right)^{1.8} \tag{21}$$

where ρ is density of working fluid.

a. Shell-side pressure drop

From Smith (2005), the Fanning friction factor of the fluid across helical tube bundle can be expressed as:

$$f = 0.26 P_y \,\mathrm{Re}^{-0.117} \tag{22}$$

where P_y is shell-side porosity which depends on geometry of the bundle. Finally, shell-side pressure drop ratio can be computed from the following relation:

$$\frac{\Delta p}{\Delta p_{ref}} = \left(\frac{\mu}{\mu_{ref}}\right)^{0.117} \frac{\rho_{ref}}{\rho} \left(\frac{\dot{m}}{\dot{m}_{ref}}\right)^{1.8883} \tag{23}$$

The key equations are Eqs (14), (15), (22), and (23). As can be seen they are independent on geometry of the exchanger. The above models should be programed by using a computer program. The system of equations has a lot of temperature-dependent properties. Therefore, EES software (Klein, 2013) is the pertinent candidate for the current study. The properties of fluids are evaluated at bulk temperature. Procedure for solving the system of equations is summarized as follows. From reference data the remaining temperatures and UA_{ref} can be calculated. Effectiveness is then assumed. Maximum thermal duty at operating conditions $\dot{Q}_{max}$ is evaluated in next step. From these parameters $\dot{Q}$ can be found. After that the outlet fluid temperatures at operating conditions are computed. Next β_i, β_e, UA, and NTU are calculated. Thereafter a new effectiveness is determined and compared to the assumed value. A new loop is carried out if error is greater than a given tolerance.

3 EXPERIMENTAL VALIDATION

The experimental apparatus shown in Figure 2 was performed to determine whether the present model could be validated. In the apparatus, water was used as the working fluids for both sides of the tested helical heat exchanger. Hot water is heated by a three-phase electrical heater in a hot water tank. The tank temperature can be adjusted to set various experiments. The hot water is pumped to tube-side of the exchanger. Here the hot water is decreased in temperature by cold water. The cold water at almost room temperature enters shell-side of the exchanger. The cold water rejects heat to the environment by an air-cooled heat exchanger right after the test section. It then travels to a large tank and mixes water in the tank. Water volumetric flow rates are measured by floating flow meters with 0.0028 L/s resolution. Inlet and outlet temperatures of both sides are measured by 4 thermocouples with 0.1ºC precision. Tube-side and shell-side pressure differences were processed by differential pressure transducers with an accuracy of ±0.075% of the measured value. Water flow rates were adjusted by ball valves.

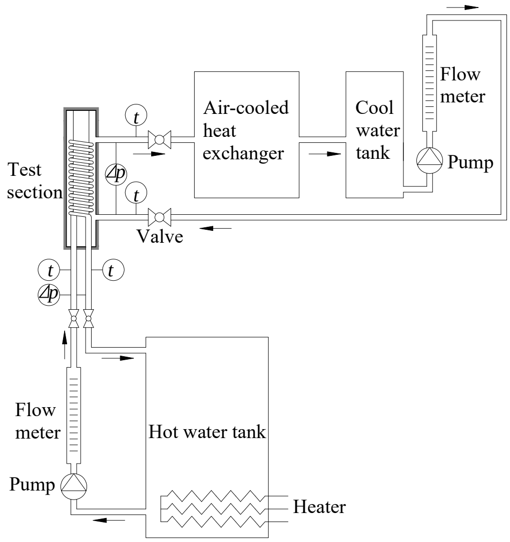

Fig. 2: Experimental apparatus

Table 1 presents reference data used in the current work. Figures 3-5 show the calculated results and experimental results of thermal duty and outlet water temperatures. Error of the thermal duty is less than ±5%. And it can be noted that errors of the outlet water temperatures is lower than ±1%. This confirmed that the heat transfer is good.

Table 1: Reference data

Heat transfer rate	$\dot{Q}_{ref} = 6.2$ kW
Tube-side pressure drop	$\Delta p_{i,ref} = 93$ kPa
Shell-side pressure drop	$\Delta p_{e,ref} = 20$ kPa
Tube-side flow rate	0.278 l/s
Shell-side flow rate	0.194 l/s
Inlet tube-side fluid temp.	$T_{i,in,ref}$ =59.5ºC
Inlet shell-side fluid temp.	$T_{e,in,ref}$ = 31.5ºC

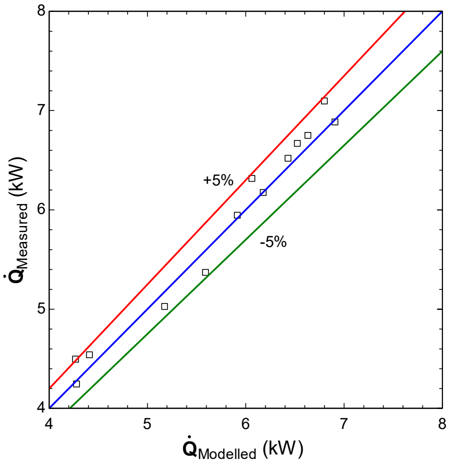

Fig. 3: Experimental vs. theoretical heat transfer rate

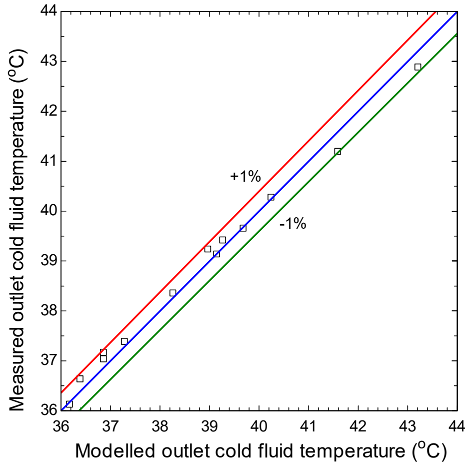

Fig. 4: Experimental vs. theoretical cold fluid temperature

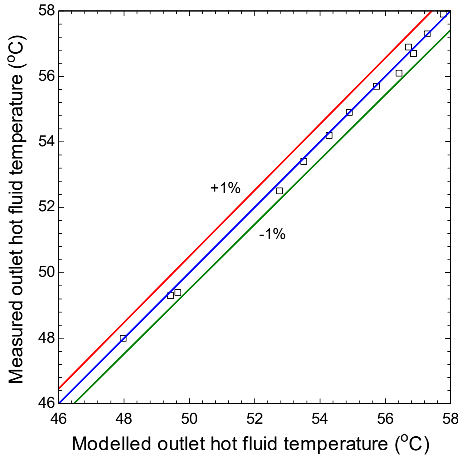

Fig. 5: Experimental vs. theoretical outlet hot fluid temperature

Figures 6 and 7 show the pressure drops between two approaches, modelling and experiment. It can be concluded that the results well coincide each other. The relative error of shell-side pressure drop is no greater than ±5%. The difference of tube-side pressure drop is within ±2% except the difference up to ±8% at low flow rate.

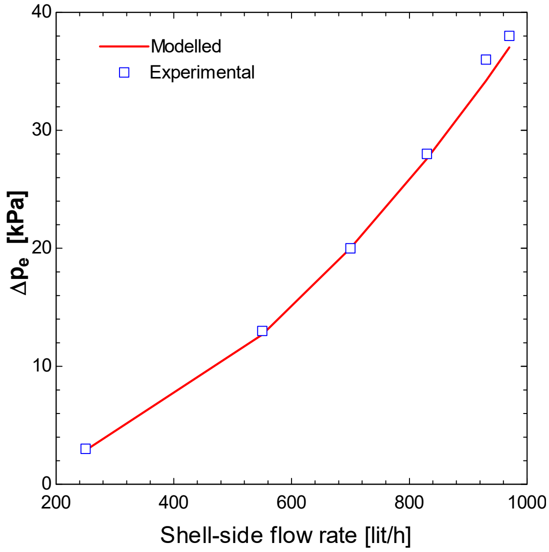

Fig. 6: Shell-side pressure drop

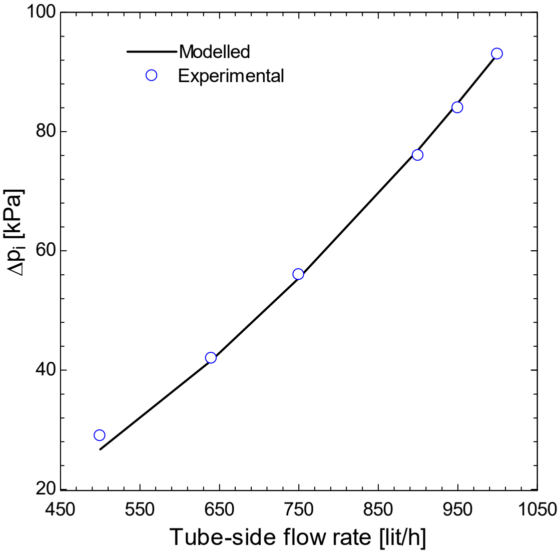

Fig. 7: Tube-side pressure drop

4 CONCLUSION

The single phase heat transfer model and pressure drop model were formulated to predict off-design conditions of the helical heat exchanger. The models could evaluate outlet fluid temperatures, thermal duty, and pressure drops for various operating conditions without geometrical information of heat transfer surface. An experiment was set-up to determine the reliability of the models. Results showed that the differences between calculation and experiment are from ±1 to ±5%.

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