Semi-Empirical Model for Forced-Convection Condensation on Integral Finned-Tubes

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Abstract—Main reason why integral-fin tubes are enhanced over smooth tubes is because of the added surface area presented by the fins that increases the area for heat transfer. The major objective of this project is to develop a semi-empirical model that can prove all the data collections of previous studies are in the mean deviation of ±25% using engineering and scientific formulas and methods.

Keywords—Semi-Empirical Model, Forced Convection Condensation, Finned Tubes, Condenser, Heat Exchanger

1. Introduction

The use of horizontal, integral-fin tubes as shown in Figure 1 to enhance condensation is quite common in the design of surface condensers in the refrigeration and power generation industries such as steam turbine. Integral-fin tube is a tube with circumferential fins on its outside, manufactured by machining the material between the fins away. Main reason why integral-fin tubes are enhanced over smooth tubes is because of the added surface area presented by the fins that also increases the area for heat transfer as given by equation 1. [1]

\[ Q = hA\Delta T \]  

Figure 1: Horizontal Integral-Fin Tube [2]

Figure 2 shows how the condensation on integral-fin tube works where high temperature steam will make contacts with the fin-tube that contains low temperature of condensate. The temperature differences will result for the condensation process and liquid will be formed. A bundle of horizontal integral-fin tube is used in heat exchanger of a steam turbine such as the shell and tube heat exchanger. As the efficiency of condensation process is increasing, thus more liquid can be formed in a certain time and the heat exchanger efficiency will also increase that can also results for the increasing efficiency of a steam turbine.

Figure 2: How Integral-Fin Tube Works

2. Methodology

Cavallini et al. (1996) developed a semi-empirical model for forced-convection condensation on integral-fin tubes. The following equation is proposed for the vapour-side, heat transfer coefficient.

\[ \alpha_v = \left[ \alpha_{ss}^n + \alpha_{fc}^n \right]^{\frac{1}{n}} \]  

where,

\[ \alpha_{ss} = C R e_{eq}^{0.8} \rho \gamma \]  

and

\[ C = 0.03 + 0.166 \frac{s}{h} + 0.07 \frac{h}{p} \]  

The values of C and n were obtained from the best fit procedure using the data of Bella et al. (1993) and Cavallini et al. (1996) for R-11 and R-113 condensing on 3 tubes.

Comparison was made between the model of R-11 and R-113 data of Cavallini with R-113 data of Honda et al. (1989, 1991). The model is able to predict the experimental data for forced-convection condensation of R-11 and R-113 occurring on horizontal integral-fin tubes with a margin of ±20%. In conclusion, Cavallini et al. successfully predicts forced-convection condensation data for R-11 and R-113 condensing on 3 tubes.

For this project, a semi empirical model involving assumptions, approximations, or generalizations designed to simplify calculation. Dimensional analysis and best fit procedure will be used to generate the model where it reduces the difficulties of having many variables by simplifying the expressions into dimensionless product which means the results will be independent from the system of units. There are some factors that need to be considered such as the selection of variables, geometry, material properties and external effects.

This semi-empirical model will be developed for the heat transfer of the condensation process involving dimensionless Nusselt number using dimensional analysis techniques involving some dimensionless parameters which are Reynold’s numbers and Prandtl numbers which represent the factors affecting the condensation which are the vapour velocity and condensate properties inside the finned-tubes respectively.

For Reynol number, the number is dimensionless relating the condensate density, viscosity, diameter of the tube and vapour city as shown in Equation 6 below.

$$Re = \frac{\rho d \nu}{\mu}$$  (6)

Another dimensionless parameter to be consider is Prandtl number where the number relates the heat capacity and condensate viscosity.

$$Pr_t = \frac{C_p \mu}{k}$$  (7)

There is also a ratio of the geometry of the fins to be consider which are the ratio of fin spacing and fin height where

$$X = \frac{t}{h}$$  (8)

Finally, the dimensionless Nusselt number equation considering most of the important properties for the heat transfer inside the fin tube.

$$Nu_{h/d} = 0.64[1+(1+1.69F)^{0.5}]^{0.5}$$  (9)

Where,

$$F = \frac{\rho gd h_w}{k U} \Delta T$$  (10)

The dimensionless parameter $F$ quantifies the relative importance of gravity and vapour velocity for the motion of the condensate film.

Generally, the semi-empirical model should be looked like the equations below:

$$Nu = Ce^{0.3}Pr^{0.5}X$$  (11)

3. Model Development

The method to develop the model is similar with Cavallini et al. (1996) First of all; sets of data from previous experiments are extracted. For this project, data is taken from Namasivaym (2006) [1]. There are sets of experiments using three different types of condensate which are steam at low pressure, steam at near atmospheric pressure and ethylene glycol at low pressure. The experiments are tested with different vapour velocities and different geometries of the fins. Figure 3 shows one of the experiments with fin spacing of 0.25 mm for steam at near atmospheric pressure tested with different values of vapour velocities.

Figure 3: Variation of Heat Flux with Vapour-Side Temperature Difference for Fin Tubes (Set A, $h = 1.6$ mm, $t = 0.25$ mm, $d = 12.7$ mm) – Effect of Vapour Velocity (steam at near atmospheric pressure)

After the data is extracted, data correlation is done and values for dimensionless properties which are Nusselt number, Reynol number and Prandtl number are calculated from the theoretical formulas. The values are then plotted into the graph of Nusselt number vs combination of Reynold number, Prandtl number and geometry ratio of fin height and fin spacing based on previous studies and best fit procedure is done to get the equation of the graph to complete the empirical model of this project as shown in Figure 4 below.

From the linear model, the equation relating all the dimensionless number which is the semi-empirical model developed is

$$Nu = 2.2(Re^{1.2}Pr^{0.3})(\frac{d}{h})^{0.3}$$  (12)

4. Results and Analysis

After the model is developed, it is important to validate the model. In this project, the model is required to give the data within 25% range of deviation.

4.1. Model Validation

From the graph, it is found that the model is able to predict 1328 points from 1560 points tested which means that the model can predict for 85.1% from the whole data. It is also noted that the points above the +25% is where the steam experiment is conducted with high vapour velocity.

5. Error Analysis

The Kline and McClintock technique was used in this model development. It was found that for a flow rate of 2l/min, there is a ±0.05l/min error. This would affect the outcome of the model development.

6. Conclusions

Heat transfer measurement for forced-convection condensation is influenced by vapour velocity, condensate properties and fin geometry. The semi-empirical model considering all this properties is able to predict the collections of the experimental data within the range of 25%. This model is also based on some engineering assumptions that can be modified for the improvement of the model.

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References