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ORIGINAL RESEARCH ARTICLE

OPTIMIZATION OF FIN GEOMETRY FOR CONDENSATION ON INTEGRAL FIN TUBES

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ABSTRACT

This paper reports an optimization aimed at verifying the accuracy of a semi-empirical condensation heat transfer model in calculating the enhancement ratio of a rectangular- finned tube. Three condensing fluids (ethylene glycol, refrigerant-R113 and steam) and three tube materials (copper, brass and bronze) were used to evaluate the response of the model at different fluid properties. Optimum tube geometries (diameter, fin thickness, height and spacing) were calculated and later compared with those geometries of a real condenser. The verification was achieved by comparing the condensation results of the model based on numerical calculations using Excel spread sheet with an established experimental data from the measurements of condensation heat transfer. The optimization results showed that the model accurately predicted the observed trends in the experimental data for the condensing fluids of steam, R113 and ethylene glycol. Condensation of refrigerant-R113 on copper tube material was found to be the best predicted model. Calculated optimum fin spacing for steam as a condensing fluid corresponded reasonably to those of a real condenser. Optimum fin spacing increased with increased surface tension.

1.0 Introduction

Extended surfaces (fins) are mostly applicable in condensation such as those in refrigeration and air conditioning. Upon their usage, enhanced heat transfer and provision of additional drainage systems may be obtained due to surface tension effects (Fitzgerald, 2011). Surface tension greatly affect heat transfer due to capillary retention of condensate between fins (Chen and Lin, 2009). These fins can be rectangular, triangular or trapezoidal in nature. Enhancement ratio of the fin tubes is generally defined as ratio of the heat-transfer coefficient of the enhanced tube (based on the pin- or fin-root diameter) to that of a plain tube with diameter equal to the root diameter of the enhanced tube provided the vapour-side temperature difference is the same (Ali and Briggs, 2013). Figure 1 shows a schematic diagram of an idealised fin-tube geometry.

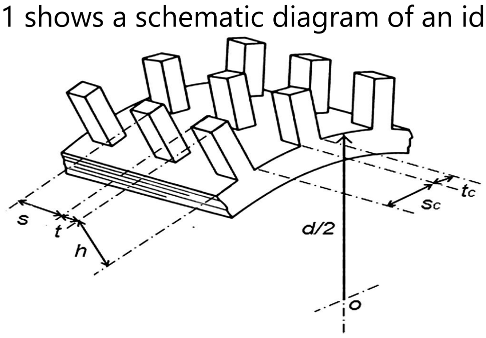

Figure 1: Schematic diagram of idealized rectangular fin-tube (Ali and Briggs, 2013)

Enhancement ratio varies with the shapes and sizes of the various fin geometries as well as nature of the condensate fluids and materials with which the fin tube is made. Effect of pin fin rotational angle on the enhancement ratio as a function of vapour velocity has recently been reported (Haseeb, 2020). Research into the effects of the interdependence of these factors on the enhancement ratio has received substantial attention for the past few decades. For example, Honda et al. (1983) reported the highest enhancement ratio of 10.2 for refrigerant-R113. Honda et al. reported that low surface tension refrigerant-R113 resulted in reduced condensate flooding. Wanniarachchi et al. (1985) reported enhancement ratio of slightly over 4 while condensing steam. Briggs et al. (1992) reported enhancement ratio of 4.7 for ethylene glycol. Ethylene glycol was reported to have ratio of surface tension to density ratio between the extremes of water and other refrigerants. Briggs et al. (1995) systematically studied the effect of condensing R-113 and steam on a rectangular fin tube made up of brass, bronze and copper. Fin thickness and fin height were also varied. Briggs et al. (1995) found that the enhancement ratio while condensing R-113 varied strongly with fin height and thickness and; weakly dependent on the thermal conductivity of the tube materials. Enhancement ratio for condensing steam appeared to have lesser effect on the fin height and thickness, but strongly dependent on the thermal conductivity. More recently, Jan et al. (2017) experimentally investigated the enhancement of condensation of heat transfer rate of the air-steam mixture on a passive condenser using annular fins. Jan et al. reported that the condensation heat transfer was enhanced by a factor of 1.54 when the area for the heat transfer increased by 84%.

This paper is aimed at verifying the accuracy of a semi-empirical model by Rose (1994) in calculating the enhancement ratio of a rectangular- finned tube. The methods employed in calculating the enhancement ratio is described as follows.

2. Methodology

2.1 Model Equations

The enhancement ratio (ε) of the Rose (1994) model is given by Eqn.1. Three condensing fluids (Ethylene Glycol, Refrigerant-R113 and Steam) and tube materials (copper, brass and bronze) were used to evaluate the response of the model at different fluid properties. Details of the model are summarized in Eqns. (1), (2), (3), (4), (6) and (8). The verification was achieved by comparing the results of the model based on calculations from Excel spread sheet with an established experimental data from Briggs (2012) obtained from the measurements of condensation heat transfer. Optimum tube geometries (diameter, fin thickness, height and spacing) were calculated and compared with those in manufacturing a real condenser.

The model was implemented on an excel spread sheet using relevant fluid properties shown in Table 1. Tube geometry parameters from the experimental data base of Briggs (2012) obtained from the Queen Mary university of London, United Kingdom was used to calculate the enhancement ratio for all the condensing fluids (ethylene glycol, R113 and steam) and on all the different tube materials (copper, brass and bronze). The definition of each parameter and numerical values of constants are shown on the nomenclature page. Piece-wise calculations of the enhancement ratio in Eqn.1 and all other subsidiary model equations from Eqn.2 to Eqn.8 were computed on an excel spreadsheet. The input parameters with their respective SI units were strictly employed.

Table 1: Properties of condensing fluid (Briggs, 2012 and EES NASA Equations, 2019)

S /No	Condensing fluids	Temperature (°C)	Density (kg /m^3)	Surface Tension (kg /s^2)
1	Ethylene Glycol	197	970	0.03268
2	R113	48	1510	0.01642
3	Steam	100	970	0.0588

$$\varepsilon = \left(\left(\frac{d_o}{d}\right)^{3/4} t\left\{0.281 + \frac{B_{tip}\sigma d_o}{t^3\tilde{\rho}g}\right\}^{1/4} + \frac{\phi_f}{\pi}\left[\frac{1-f_f}{\cos\beta}\,\frac{{d_o}^2-d^2}{2{h_v}^{1/4}d^{3/4}}\right]\left\{0.791 + \frac{B_{flank}\sigma h_v}{h^3\tilde{\rho}g}\right\}^{1/4} + B_1(1-f_s)s\left\{(\xi(\phi_f))^3 + \frac{B_{int}\sigma d}{s^3\tilde{\rho}g}\right\}^{1/4}\right]\Bigg)/0.728(b+t) \quad (1)$$

Although Eqn. (1) is a model for trapezoidal fins, adequate care has been taken during the computation to use the requirements for rectangular fins only. The fin-tip half angle (β) in Eqn.1 differentiates the model for different fin geometry (β=0) for rectangular fin geometry.

ϕ_f, f_f ,f_s , h_v and $\xi(\phi)$ in the model are further defined in the following equations:

$$\phi_f = \cos^{-1}\left[\frac{4\sigma}{\rho g b d_o} - 1\right] \quad \text{for } b < 2h \quad (2)$$

where ϕ_f is the retention or flooding angle of the retained condensate from the top of the fin

f_f and f_s are the fractions of the fin flank and inter-fin tube surface “blanked" by condensate "wedge” respectively. According to Masuda and Rose (1987), f_f and f_s are respectively given by Eqns. (3) and (4) for a rectangular fin.

$$f_s = \frac{2\sigma}{\rho g d h} \cdot \frac{\tan(\phi_f/2)}{\phi_f} \quad (3)$$

$$f_s = \frac{4\sigma}{\rho g d s} \cdot \frac{\tan(\phi_f/2)}{\phi_f} \quad (4)$$

According to Rose (1994), the mean vertical height of the fins (h_v) is given by Eqn. (5) and Eqn. (6). However, only Eqn. (6) corresponds to the condition of ϕ_f for all the data points during computation on an Excel spreadsheet.

$$h_v = \frac{\phi_f}{\sin(\phi_f)} h \quad \text{for } \phi_f \leq \frac{\pi}{2} \quad (5)$$

$$h_v = \frac{\phi_f}{2-\sin(\phi_f)} h \quad \text{for } \frac{\pi}{2} < \phi_f \leq \pi \quad (6)$$

Active area enhancement $\xi(\phi_f)$ as defined by Masuda and Rose (1987) for a rectangular fin is given by Eqn. (7).

$$\xi(\phi_f) = \frac{\text{Unblanked area of finned tube}}{\text{area of smooth tube}} = \frac{R_r b\phi_f(1-f_t)+({R_o}^2-{R_r}^2)\phi_f(1-f_f)+\pi R_o t}{\pi R_r(b+t)} \quad (7)$$

where: $R_r b\phi_f(1-f_t)$ = area of a fin top; $({R_o}^2-{R_r}^2)\phi_f(1-f_f)$ = area of unblanked part of fin flank and $\pi R_o t$ = area of unblanked part of inter-fin tube surface

Rose (1994) calculated the active area enhancement as a function of flooding angle based on Nusselt (1916) theory and is given by Eqn. (8).

$$\xi(\phi) = 0.874 + 0.1991 \times 10^{-2}\phi - 0.2642 \times 10^{-2}\phi^2 + 0.5530 \times 10^{-2}\phi^3 - 0.1363 \times 10^{-2}\phi^4 a \quad (8)$$

The standard deviation for a sample of data between the calculated and experimental values is given by Eqn. (9) as reported by Freedman et al. (1998).

$$\text{Standard deviation (Std. dev.)} = \sqrt{\frac{\sum_{i=1}^{N}(\varepsilon_c-\varepsilon_e)^2}{N-1}} \quad (9)$$

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Where: N= total number of data points.

However, three (3) standard deviations were calculated to reflect the effect of removing some data points due to their inconsistencies with the model. They are as shown below:

Std. dev. 1 = $\sqrt{\frac{\sum_{i=1}^{148}(\varepsilon_c - \varepsilon_e)^2}{148-1}}$ = 0.573 for the entire 148 data points

Std. dev. 2 = $\sqrt{\frac{\sum_{i=1}^{146}(\varepsilon_c - \varepsilon_e)^2}{146-1}}$ = 0.553 for 146 data points (2 data points where the fin tip half angle is not zero removed)

Std. dev. 3 = $\sqrt{\frac{\sum_{i=1}^{139}(\varepsilon_c - \varepsilon_e)^2}{139-1}}$ = 0.549 for 139 data points (9 data points that could not predict the model removed).

3.0 Results and Discussion

3.1 Verification of the model with experimental data

Figure 2 shows a graph of calculated enhancement ratio (ε_c) from the model against the experimental enhancement ratio (ε_e) for the three condensing fluids (steam, R113 and ethylene glycol) in the three tube materials (bronze, brass and copper). The ε_e was adopted from Briggs (2000), while ε_c was calculated from the model. The line of best fit in Fig.2 has an equation of y = 0.9817x from the origin. The other two lines showed the deviation of +25% and -25% from the line of best fit.

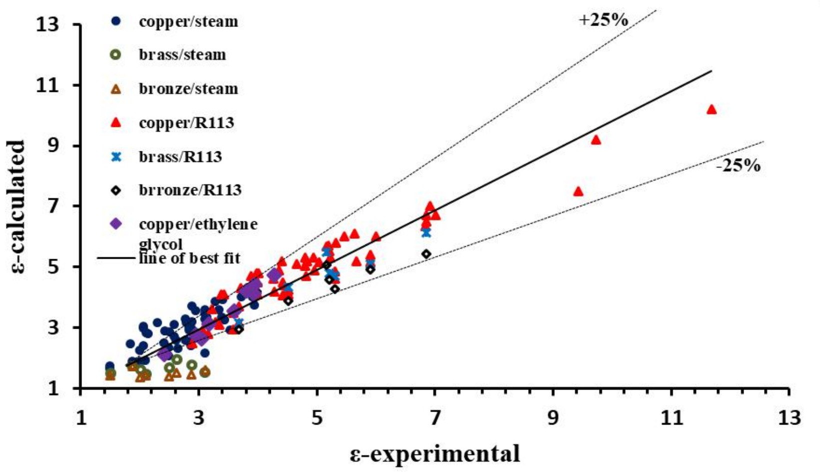

Figure 2: Calculated and experimental enhancement ratios of different working fluids and tube materials

It can be seen from Figure 2 that most of the data points for the condensation of refrigerant-R113 in copper tube material fall within the standard deviations of ±25% of the line of best fit, and therefore predicted the model best. Similar observations could be seen on the condensation of R113 in brass and bronze and the condensation of ethylene glycol in copper. Condensation of steam in copper was partly predicted by the model, because some of the data points fall outside the ±25% standard deviations. However, condensation of steam in brass and bronze did not to predict the model well, since none of the data point fall within the ±25% standard deviations. It is noteworthy that condensation of R113 in copper did not only predict the model correctly, but also provided the highest calculated and experimental enhancement ratios. Generally, the model predicts the observed trend in the data for condensation of steam, R113 and ethylene glycol. The calculated enhancement ratio response varies in the same manner for all the fluids. Therefore, the trend on the enhancement ratio is found to be dependent on the fluid properties.

Generally, the model can be said to be in good agreement with the experimental data. This is evident from Figure 1 where most of the data points were having standard deviations of ±25% to the line of best fit on the graph. Although an assumption of zero flooding angles was made for seven data points for which the component $(\frac{4\sigma}{\rho g b d_o} - 1)$ from Eqn. 2 is greater than unity. Nevertheless, the model reasonably predicted these points to a high degree of accuracy. This indeed is a fair approximation, since, the seven points were assumed to be completely flooded within the inter-fin spaces at zero vapor velocity (Fitzgerald, 2011). Two data points designate points where the fin tip half angle is not zero (i.e., β = 4.50 and β = 50), which is obviously not true for rectangular fins. Hence, the reason for the discrepancy of the points with the rest of the data.

It can be seen clearly from the 3 values of the standard deviations calculated from Eqn. 9 that, Std. dev. 2 seemed to be more reasonable, since it excludes the 2 points on which the model is not valid and includes the fair approximation for the 7 data points. Furthermore, it is quite justifiable to remove some data points from the data-base since the model could not provide reasonable predictions. The two data points where the fin tip half angle is not zero were therefore removed. The assumption that the model is rectangular does not work for these points. Due to differences in the prediction of condensation for steam on brass and bronze tube materials, Briggs and Rose, (1994) modified the model to include conduction in the rectangular fins. The subsequent section discuss how the change in the fluid and tube materials influence the enhancement ratio.

3.2 Influence of condensing the same fluid on different tube materials

3.2.1 Condensation of steam on copper, brass and bronze

Figure 3 shows the response of experimental enhancement ratio (ε_e) against the calculated enhancement ratio (ε_c) for steam condensing on copper, brass and bronze. A fair response for steam condensing on copper tube material was observed from Figure 3. However, the prediction of condensation of steam on brass and bronze tubes was totally inconsistent with the model. Good prediction of the model by condensing steam on copper tube materials is believed to be due to high thermal conductivity of copper. On the other hand, condensing steam on brass and bronze tubes is inconsistent with the model primarily due to temperature fall for large $\alpha h^2/tk_w$ explained by Briggs, (2012) and differences in surface tension of steam on brass and bronze (Rose, 2004).

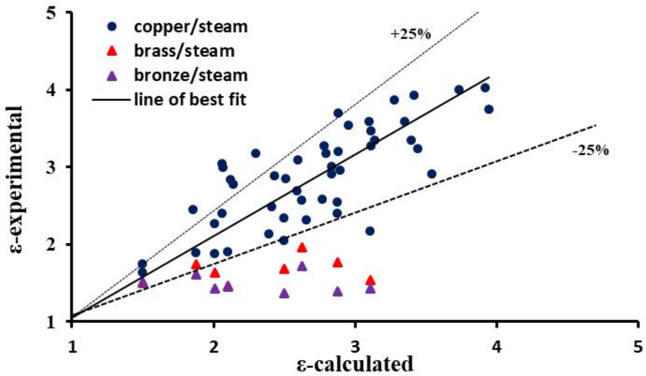

Figure 3: Calculated and experimental enhancement ratios for condensation of steam on copper, brass and bronze tubes.

3.2.2 Variation of enhancement ratio with fin geometry

Three fin geometry were considered (spacing, height and thickness) and the response of the calculated enhancement ratio was determined at a fixed fin root diameter (d = 12.7 mm) while also keeping two out of the three fin geometries constants. Figure 4 show variations of the calculated enhancement ratio (ε_c) with fin spacing (s) (h =1.6mm and t = 0.5mm). It can be observed from the Figure 4 that the calculated enhancement ratio (ε_c) decreases with increasing fin spacing for steam, R113 and ethylene glycol. Figure 5 presents variations of enhancement ratio (ε_c) with fin thickness (t) (h = 0.9 mm and t = 1 mm). It can be seen from the graph that the calculated enhancement ratio also decreases with increasing fin thickness. Figure 6 shows the relationship between ε_c and fin height (h) (s = 1mm and t = 0.5 mm). It can be deduced from Figure 6 that ε_c increases with increasing fin height for all the condensing fluids. Wanniarachchi et al. (1985) also observed similar trend of increased enhancement ratio with increased fin height.

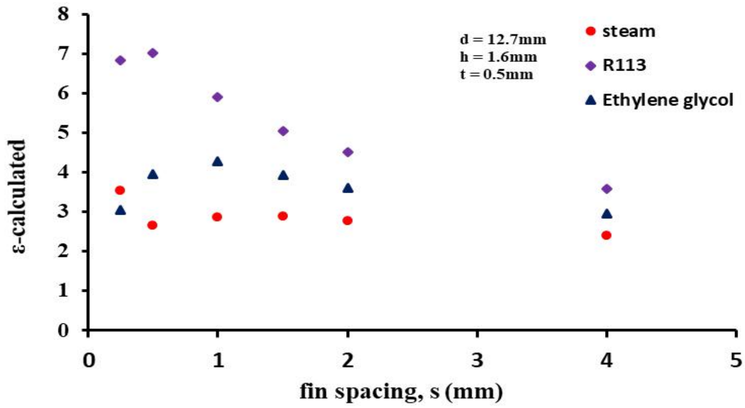

Figure 4: Variation of calculated enhancement ratio with fin spacing (s)

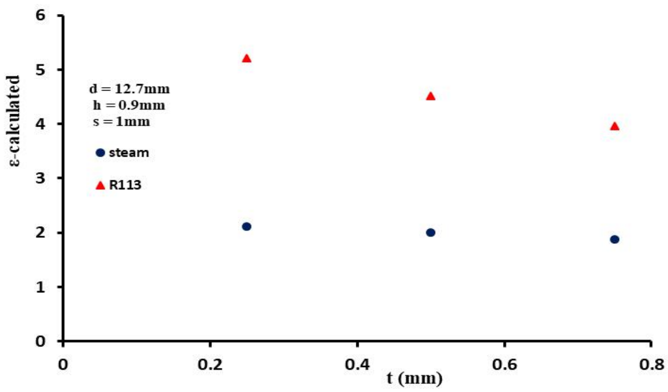

Figure 5: Variations of calculated enhancement ratio with fin thickness

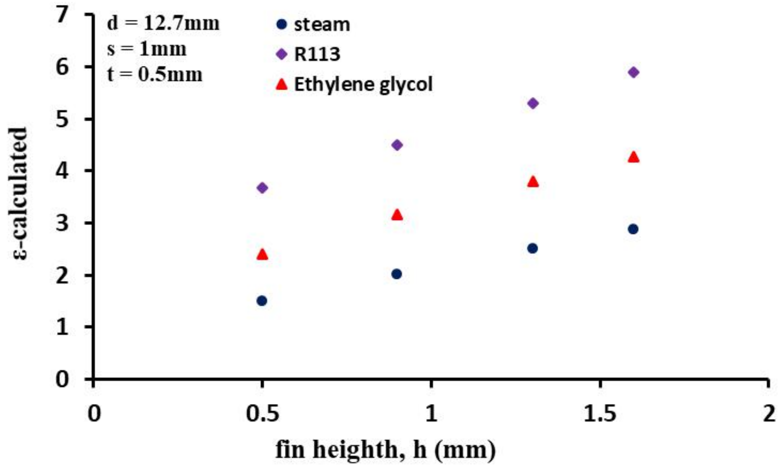

Figure 6: Variations of calculated enhancement ratio with fin height (h)

These variations have been associated with surface tension effects and capillary retention of the condensate (Briggs, 2012). Pravin, (2011) listed Surface of the condensing fluids (σ), Density (ρ) and Surface tension to density ration (σ / ρ) as the basic factors that could affect the trend of the model. These parameters are all included in the model which help to predict the trends reasonably well. This observation could be underpinned using earlier models such as those of Beatty and Katz, (1948) and Rudy and Webb, (1981) in Rose, (1994). These two models neglected the effects of surface tension and thus, suffer limitations of interpretations and under-predictions (Briggs, 2012).

3.2.3 Influence of tube geometries on the model performance

Tube geometries from the data base were optimized for condensation of steam and refrigereant-R113 while making the following parameters constants (Maximum fin root diameter d = 19.1 mm, Fin thickness t ≥ 0.5 mm and Rectangular fins θ = 0). Results for the optimum tube geometries are presented in Figures. 4, 5 and 6. The results are then compared with that of a commercial real condenser shown Table 2. It can be seen from Table. 2 that optimum fin spacing for steam is 1.5mm which is within the specified range of 1- 2 mm for a commercial condenser obtained from Briggs, (2012) while the optimum fin spacing for R113 is 0.5mm which is out of the range and therefore, not highly recommended. However, these correspond with values obtained by Patten, (1988) using the same diameter. Optimum fin height and thickness for both steam and R113 are 2mm and 0.5mm respectively.

Table. 2: Comparison of calculated optimum tube geometries with commercial tube geometry for practical tube manufacture of a real condenser (Briggs, 2012)

Tube geometry parameters	Calculated optimum tube geometries for condensing fluids at d = 19.1mm		Values from the commercial tube geometries for condensing fluids	
	Steam	Refrigerants	Steam	Refrigerants
Fin spacing, s (mm)	1.5	0.5	1 - 2	1 – 2
Fin thickness, t (mm)	0.5	0.5		
Fin height, h (mm)	2	2	1	0.2 – 0.4
Fin root diameter, d (mm)	19.1	12.7		
Fin tip diameter, d_o (mm)	23.1	16.7		

4. Conclusion

The accuracy of a semi-empirical model of Rose (1994) has been verified and the following conclusions could be deduced:

The model clearly predicted an observed trend in the experimental data for most of the condensing fluids (steam, R113 and ethylene glycol) on brass, bronze and copper.

Condensation of refrigerant- R113 on copper tube predicted the model with the best results while condensation of steam in brass and bronze were poorly predicted by the model.

Standard deviations between the calculated and experimental enhancement ratio data was found to be 0.553

Calculated enhancement ratio appeared to increase with fin height but decreased with increasing fin thickness and fin spacing regardless of the condensing fluid and the tube material.

Calculated optimum fin spacing for steam corresponded reasonably to those of a real condenser and the optimum fin spacing increased with surface tension increase.

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