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Optimizing Fin Geometry for condensation on Integral Fin Tubes

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***Abstract*--The overall aim of this report was based on the analysis of the enhancement ratio for heat transfer horizontal fin-tubes and wastobe confirming the validity of a semi-empirical model by comparison with experimental measurements. This model has been taken from a research study of Queen Mary University of London. Estimating the optimum fin spacing, height, thickness and tube diameter for different fluids to increase the heat transfer performance of shell-side condensers. The results have been split into the three different fluids, steam, R113 and glycol and three tube materials, copper, brass and bronze. Enhancement ratios rise gradually with fin height and diminish with fin thickness. The fin spacing has been demonstrated to very relevant factor to enhance the performance. This work has revealed that the optimum spacing for the refrigerant is 0.5 mm while for steam is 1 mm, for the same thickness and diameter.**

***Keyword*- Heat transfer, fin tubes geometry, optimization, condensers.**

I. INTRODUCTION

Condensation heat transfer has been studied extensively because of its importance in broad industrial applications from thermal power plants [1] to air-conditioners [2].Improvements of condensers thermal performance can entail increases in energy efficiency, reduction in the size of heat exchangers and its correspondingfinancial savings.

Raising the heat transfer ratio of smooth tubes by adding extended surfaces called fins has become popular since the end of 1970s [3]. **Error! Reference source not found.**, taken from [4], shows the cross sections of two different integral-fin tubes.

This enhancement is to some extent due to the boost of the effective exchange area. However, the area cannot be exceeded not only due to manufacture limits, but also because high fins entail an increase in the thermal resistance, the condensate layer [5]. Besides, the condensation process in the fins leads to a pressure gradient in the liquid which produces a drainage mechanism. This drainage needs fins to be low to pressure gradient and condensate surface curvature vary [6].

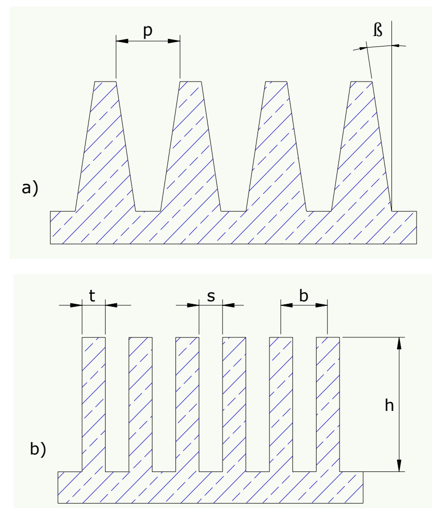

Figure 1. Schematic Of (A) A Trapezoidal Integral-Fin And (B) A Rectangular Integral-Fin Geometry.

Integral fin geometries have the drawback of retaining condensation at the lower part of the tube (flooding) by capillarity. Such a problem is caused by surface tension [6]. This disadvantage is one of the main concerns for the research.

Over the years, many analytical models regarding condensation on integral-fin tubes have been carried out [7] due to its numerous engineering applications, its mathematical complexity and the adoption of new working fluids. The heat transfer rate in shell-side condensers can be improved by increasing the performance of the condensing side of the tube.

This project deals with an analytical model and the geometry optimisation of some finned-tube surfacetobetter understand the condensation of shell-side condensers.

II. METHODOLOGY

This work is carried out in an Excel spreadsheet involving the following equations,[5].

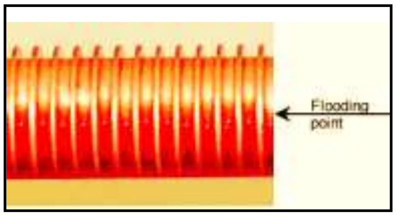

Figure 2. Model Accuracy.

Condensate retention is a critical factor in the heat transfer process for horizontal fin-tubes. The condensed liquid is retained by capillary forces at the lower part of the tube.

The flooding is widely expressed by an angle measured from the top of the tube to the inter-fin space, equation (1).

$$\theta_f = \cos^{-1}\left(\frac{4\sigma\cos\beta}{\rho g b d_0} - 1\right), \quad b < 2h\cos\beta / 1 - \sin\beta, \tag{1}$$

where β is the fin tip half angle. For rectangular fin sections this angle is equal to 0. The fraction of fin flank and the fraction of inter-fin tube surface have been also considered as follows, equation (2):

$$f_f = \frac{1-\tan(\beta/2)}{1+\tan(\beta/2)} \cdot \frac{2\sigma\cos\beta}{\rho g d h} \cdot \frac{\tan(\theta_f/2)}{\theta_f}$$
$$f_s = \frac{1-\tan(\beta/2)}{1+\tan(\beta/2)} \cdot \frac{4\sigma}{\rho g d s} \cdot \frac{\tan(\theta_f/2)}{\theta_f} \tag{2}$$

Other parameters considered have been the mean vertical fin height, equation (3),

$$h_v = \frac{\theta_f}{\sin(\theta_f)} \cdot h \qquad \theta_f \le \frac{\pi}{2}$$
$$h_v = \frac{\theta_f}{2-\sin(\theta_f)} \cdot h \qquad \frac{\pi}{2} \le \theta_f \le \pi \tag{3}$$

and the approximated equation given by, equation (4):

$$\xi(\theta_f) = 0.874 + 0.1991 \cdot 10^{-2} \cdot \theta_f - 0.2642 \cdot 10^{-1} \cdot \theta_f^{\,2} + 0.553 \cdot 10^{-2} \cdot \theta_f^{\,3} - 0.1363 \cdot 10^{-2} \cdot \theta_f^{\,4} \tag{4}$$

Finally, the enhancement ratio is calculated by equation (5) with all above assumptions:

$$\varepsilon = \left(\frac{d}{d_0}\right)^{3/4} \cdot t \cdot \left(0.281 + \frac{B_{tip}\sigma d_0}{t^3 \tilde{\rho} g}\right)^{1/4} + \frac{\theta_f}{\pi}$$
$$\left[\frac{1-f_f}{\cos\beta} \cdot \frac{d_0^2 - d^2}{2h_v^{1/4} d^{3/4}} \cdot \left(0.791 + \frac{B_{flank}\sigma h_v}{h^3 \tilde{\rho} g}\right)^{1/4} + \right. \tag{5}$$
$$\left. B_1 \cdot (1-\mathrm{f}_s) \cdot \mathrm{s}\left(\left(\xi(\theta_f)\right)^3 + \frac{B_{\text{int}}\sigma d}{s\tilde{\rho} g}\right)^{1/4}\right] / 0.728(\mathrm{b}+\mathrm{t})$$

Therefore, tubes geometry, the working fluid and fin features affect to enhancement ratio of the tube. This proportion can be defined as the heat transfer coefficient of the finned tube divided by the heat transfer of the smooth tube [6].

Optimization:

For this section, all possible combinations for the fin thickness, height, spacing and tube diameters have been introduced. The values are within the following range according to the data given, table 1:

Table 1.
Combinations, Thickness, Height, Spacing And Tube Diameters.

d: 12,7 mm and 19,7mm	0,5mm < t < 1,5mm
0,5mm<s<4mm	0,5mm < h < 1,5mm

III. DEVELOPMENT

A. Part 1. Model verification

Agreement between the model and with the experiment. 1): The results have been split into the three different fluids in order to compare clearly the experimental data with the model. As can be seen in Figure 3, Figure 4 and Figure 5, such relation have been plotted with its corresponding regression line as well as with the line x=y.

Figure 3 shows the calculated enhancement ratio against the experimental one. The results follow considerably the identity line, so one can say that the model gives a good agreement with the experiment in the case of steam.

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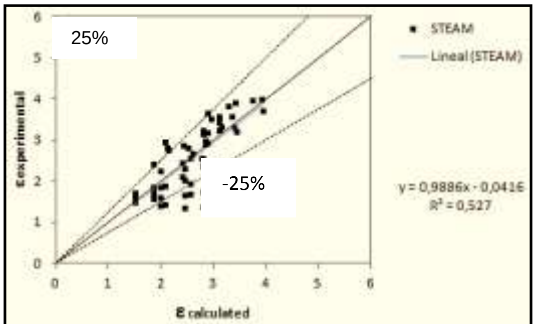

Figure 3. Comparison Of Theoretical Model With The Experimental Model For Steam.

However, the coefficient of determination R^2 indicates that the data points fit lowly their regression line, accounting for 0.527. Furthermore, the outcomes vary 0.389 from the experimental data (Standard deviation between both ratios).The Standard Deviation (SD) has been computed according to Equation 6, the standard deviation of the difference of experimental data minus theoretical data.

$$SD = \sqrt{\frac{\sum_{i=1}^{n}(\varepsilon_{experimental} - \varepsilon_{calculated})^2}{n-1}} \qquad (6)$$

Regarding the refrigerant, the data points adapt slightly better that the stream to the identity line. In fact, all the results are within the range ±25 % of the expected values, **Error! Reference source not found.**. Also, the statistics adjust greater to the linear trend line with the coefficient of determination of 0.861. As far as standard deviation concerns, it is 0.365.

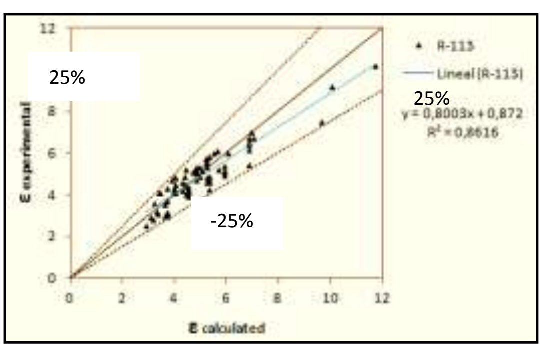

Figure 4. Comparison Of Theoretical Model With The Experimental Model For R-113

In the case of Glycol, the number of figures is significantly lower, but it is apparent from **Error! Reference source not found.** that the calculated model adapts also to great extent to the identity line and all the values are within the expected range. Concerning the determination coefficient is nearly one, which would be the ideal situation. The data is spread out conforming to standard deviation 0.22.

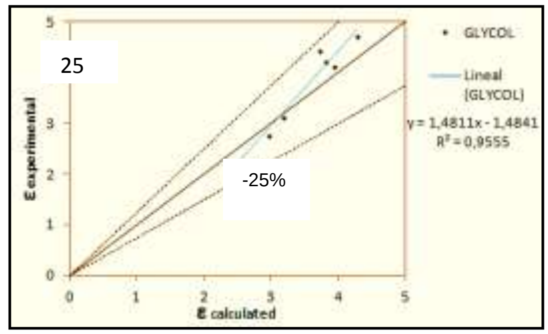

Figure 5. Comparison Of Theoretical Model With The Experimental Model For Glycol.

In view of the above results, it may be concluded that that the model satisfies largely the measurements in general.

Comparison of tube materials and working fluids. 2): It is worthwhile to make a distinction between the tube materials since the theoretical model does not include the features of any of them, Equation 5.

The **Error! Reference source not found.** and **Error! Reference source not found.** can thus be used to predict that model is more suitable for copper than for brass and bronze for any working fluid. Furthermore, the overall outcomes for refrigerant R-113 are more favourable since the data points for brass and bronze fit more properly in the analysis that in the steam case, as below figures depict.

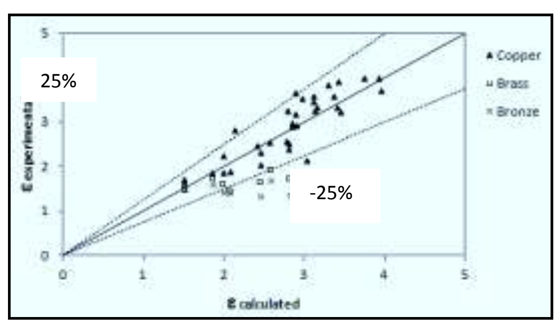

Figure 6. Comparison Of Enhancement Ratios According To The Tube Materials For Steam.

Regarding Glycol, it is difficult to find similarities with the other liquids considering that there is not too many data and the unique studied material is copper.

The reason for these variations among the materials might be due to their thermal conductivity. The shapes of the charts demonstrate that the copper is the most convenient material for the tubes. The thermal conductivity of pure copper is three times higher than brass and fifteen times higher than bronze, table 2 [8]. In fact, brass and bronze are copper alloys.

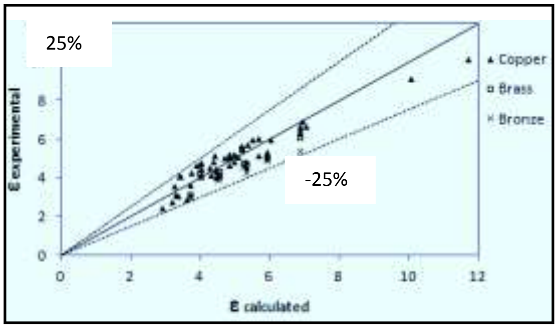

Figure 7. Comparison Of Enhancement Ratios According To The Tube Materials For R-113.

Table 2.
Thermal Conductivities For Different Metals.

Metal	Thermal Conductivity [W/mK]at 100^0C
Copper	401
Brass	133
Bronze	26

It is interesting to point out that the model works better in the R-113 example, particularly for brass and bronze tubes. The refrigerant has the lowest surface tension of the working fluids presented in this work which leads to reduce condensate flooding. The surface tension implies a pressure gradient in the liquid along the fin flanks.

Finally, we must take into accounttwofactors which make some figures not acceptable.

- It has been found out that for 0.25 mm and 0.5 mm spacing between fins at fin roots the Equation 1 (the flooding angle) makes an error in steam and glycol analysis. Six data points of steam and one of glycol have been removed. Arccosine of their corresponding values are not within -1 and +1 range.
- Consistent to Equation 1, the spacing at fin tip has to be lower than twice the radial fin height when the fin is rectangular. If it is trapezoidal, the fin tip half angle is considered. $b < 2h\cos\beta / 1 - \sin\beta$. This occurs mainly for 4 mm fin spacing. It has been obtained eight cases for steam, seven for R-113 and one for glycol.

Variation of enhancement ratio with different parameters. 4): Details of the fin height, thickness and spacing effect can be found below.

illustrates how calculated enhancement ratio boost gradually with fin height for all fluids as expected. The bigger is the height, the bigger is the heat transfer surface. Thus, the rate of heat transfer may increase several folds.

However, the length of the height is principally restricted by production processes (joining large size fins or decreasing the inter-fin spacing). Also, the heat transfer rate would diminish because of the reduction of the most effective part of the fin (tube surface). Consequently, the less effective part (parallel to the flow) would increase.

For the purpose of drawing this trend, It has been necessary to remain the fin root diameter, fin thickness and fin spacing constant. As can be seen for the graph the highest enhancement ratio corresponds to R-113.

Error! Reference source not found. shows the enhancement ratio variation with fin thickness. In this case, diameter, spacing and height are the constant values. For this parameter, the general trend is opposite to the height one. The enhancement ratio decreases gradually with the fin thickness.

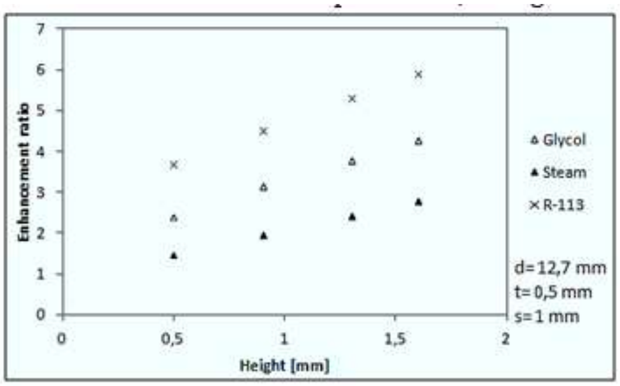

Figure 8. Variation Of Enhancement Ratio With Fin Height.

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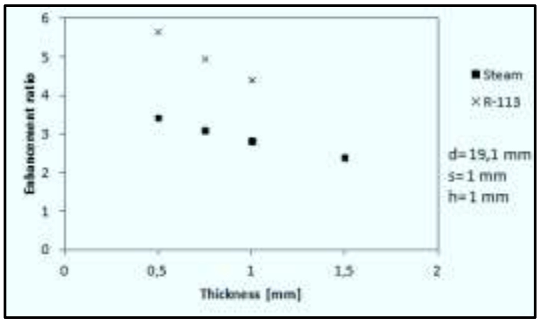

Figure 9. Variation Of Enhancement Ratio With Fin Thickness.

The graph below describes how the enhancement ratio changes with spacing at fin root. According to this figure, one can say that there is downwards trend. However, looking more closely at **Error! Reference source not found.** there is a peak at some point. That is why is really relevant to find the optimum fin spacing. For the case of the following specific diameter, thickness and height, this peak accounts for 0.5 mm spacing.

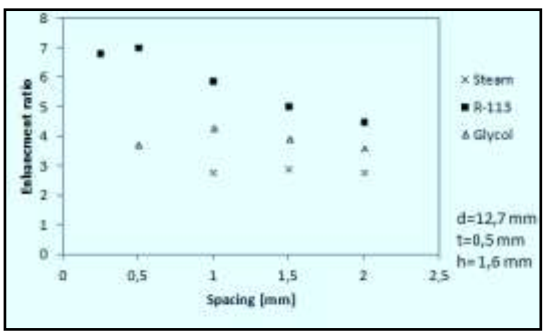

Figure 10. Variation Of Enhancement Ratio With Inter-Fin Spacing.

B. Part 2. Optimization.

In order to tackle the optimum tube geometry for steam and R-113 as working fluids, 195 combinations have been calculated and plotted. Rectangular fins and thickness higher than 0.5 mm have been fixed.

As explained in, the highest is the height; the highest is the enhancement ratio. Therefore, to represent the variation of fin spacing by including the change in the thickness and in the tube diameter, the height has been set at 1.5 mm. This is the highest value of the range given in data base.

Error! Reference source not found. shows that the optimum tube geometry for steam as working fluid is: s= 1 mm, t= 0.5 mm, h= 1.5 mm and d= 19.1 mm. For this combination, the ratio is equal to 4.18. Best ratios have been obtained for low fin thickness and high diameters.

As far as R-113, **Error! Reference source not found.**, the most favourable geometry is: s=0.5 mm, t=0.5 mm, h=1.5 and d= 19.1. The enhancement ratio for the R-113 is approximately twice as many as the ration for the steam, accounting for 8.25.

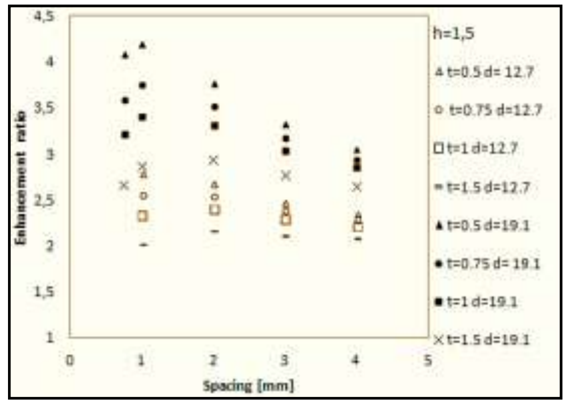

Figure 11. Variation Of Enhancement Ratio With Spacing According Todiameter And Thickness For Steam.

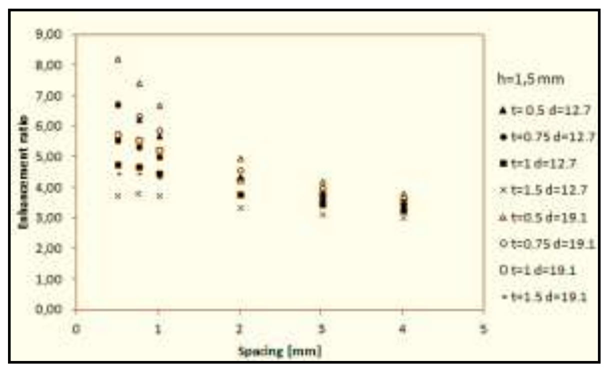

Figure 12. Variation Of Enhancement Ratio With Spacing According Todiameter And Thickness For R-113.

IV. Conclusions

Although the model agrees with the experiment properly, the outcomes have fitted more suitable for copper as tube material and the refrigerant as working fluid. The reasons for that may be the high thermal conductivity of copper and the low surface tension of fluids.

In view of the results, enhancement ratios rise gradually with fin height cause the increasing area. However, this length cannot be as much long as one wanted. There are manufacture requirements as well as rate heat transfer loss. The theoretical model cannot predict the limit of the fin height length.

Enhancement ratios diminish with fin thickness since the density is reduced.

Fin spacing has been demonstrated to very relevant factor to enhance the performance of horizontal finned tubes. The optimum fin spacing can be identified with the mathematical model in question. It depends substantially on the working fluid. This work has revealed that the optimum spacing for the refrigerant is 0.5 mm while for steam is 1 mm, for the same thickness and diameter, table 3. As a result, the tube has more quantity of fins and higher rate heat transfer.

Table 3.
Optimum Choice Of Fin Geometry

Optimum choice					
Material	Working fluid	Spacing (mm)	Thickness (mm)	Height	Diameter (mm)
Copper	R-113	0.5	0.5	1.5	19.1

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