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GENERAL CORRELATIONS FOR THE PERFORMANCE PREDICTION OF STRIP FIN PLATE-FIN SURFACES

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The paper presents a theoretical study approach to the Nusselt number and friction factor characteristics in strip fin surfaces. Extensive analyses were conducted on the thermal and hydraulic performances of off-set strip fin surfaces using the experimental data provided by Kays & London [1] and Shah and London [2,3]. General relationships were developed to predict the hydraulic and thermal performances giving accurate predictions across all flow regimes for a wide range of common strip geometries. The pressure drop encountered during flow through channels having off-set strip fins is affected by both flow disruption and flow blockage. Flow disruption appears to be the dominant factor in the laminar flow region. Under turbulent flow conditions, both effects were considered. Laminar flow heat transfer is well predicted if the Graetz Number is based on four-time strip fin length. Turbulent flow heat transfer is found to be proportional to the square root of the friction factor.

Keywords: plate-fin surfaces, strip fins, friction factor, j-factor, general correlations

1 INTRODUCTION

Enhanced heat transfer can be effectively achieved using plate-fin heat exchangers with favorable surface-area-to-volume ratios and good heat transfer characteristics. A high area density of small passages with relatively high fin thickness between the passes can be provided, enabling high design pressure limits. Strip fins have the advantage of design flexibility, which allows engineers to modify the geometry and arrangement of fins according to the specifications required. Furthermore, strip fins can reduce the risk of fouling and the accumulation of surface contaminants. This contributes to improved heat transfer efficiency and minimizes the need for frequent maintenance and cleaning in the long term. Also, more uniform temperature distribution across the strip fin surface can be achieved, reducing the risk of hot spots and thermal gradients. Therefore, strip fins are suitable for a wide range of heat exchange industrial applications, such as; automotive, aerospace, HVAC, electronics cooling, and more.

Conversely, thermal stresses pose a significant issue in plate-fin exchangers as a result of the parallel area between passes. Furthermore, the presence of any plate-fin configuration and complex flow distribution can cause a reduction in performance. The characteristics of these types of heat exchangers have been evaluated through various methods, including experimental, numerical and theoretical analyses. Several experimental works were implemented in vacuum-brazed fin channels. However, the performance of the plate-fin heat exchangers on a variety of working fluids with appropriate correlations is sparse in the available literature. Several research studies of Nusselt number and friction factor characteristics in strip fin surfaces with an extensive review on thermo-hydraulic performances and effectiveness of plate-fin heat exchangers adopted for single and two-phase fluid flows [4-6]. Furthermore, Vyas Advance CFD Analysis on Off-set strip fins was implemented [7]. Whereas, the performance of 2D/3D models of plate-fin heat exchangers was conducted [8]. Also, an extensive experimental analysis was carried out [9]. Polley and Abu-Khader [10] presented new correlations to evaluate the thermo-hydraulic performance of rectangular plain fin surfaces over laminar, transitional, and turbulent flow regimes. These design correlations can be used along with the available experimental data by Kays and London [1]. Recent work was conducted to develop simplified correlations for surfaces such as louvered, rectangular, and triangular fins [11].

Prasad [12] has stated that the absence of reliable generalized correlations for the prediction of plate-fin exchanger performance is a significant drawback to their application. At present designers depend on experimentally determined j-factor and friction factor data for a range of fin surfaces. This dependence on experimental data presents four difficulties. First, rather than engineer a surface for a specific application, the engineer is restricted to using a geometry for which data is available. Second, all experiments are subject to error. These errors are subsequently embedded into equipment design. Third, design algorithms become based on the examination of a database. This means that the development of new algorithms is time-consuming. Finally, the absence of a general correlation means little guidance regarding how finned surfaces can be improved. The development engineer moves forward based on experimental results and experience rather than understanding. Prasad [12] also points out that the only open literature source for extensive coverage of plate-fin surface performance is that published by Kays & London [1]. In this paper, the data of Kays & London are analyzed in an attempt to generate general correlations for friction factors and j-factors. Whilst the coverage made by Kays & London is wide, it is still insufficient for a truly general result. Consequently, attention has also been directed to identifying the conditions in which performance can be expected to diverge from that predicted using the correlations presented here.

The main goal of this study is to establish hydraulic and thermal performance relationships with a satisfactory level of accuracy that can be applied to various off-set strip fin configurations found in existing literature. The data source for this investigation is provided by Kays and London [1]. With regard to strip-fin surfaces, it is emphasized that the j-factor and the friction factor are influenced by the thickness of the fin. Furthermore, the manufacturing process results in the production of burs that have a substantial effect on pressure drop. They found it difficult to reproduce surfaces exhibiting the same performance. This suggests that strip-fin surfaces fabricated from materials that will bend and burr should be avoided. Alternate plates could have different pressure drop characteristics. This would result in unwanted flow mal-distribution in industrial exchangers.

2 THEORETICAL FRAMEWORKS

The analysis undertaken here seeks to build upon established heat transfer and friction factor correlations for simpler geometry. In the case of plain fin surfaces the correlation builds upon those accepted for circular tubes. Moving forward to other fin types, the aim has been to relate the performance of these surfaces to those of the plain fin. In each case, the plain fin is taken as a special case of the enhanced type. For instance, the plain fin surface can be considered to be a louvered fin with a louver gap of zero; a wavy surface with infinite wave pitch; or an off-set fin with a fin length equal to the exchanger length. Enhancement factors need to be reduced to unity for the plain fin case. Rather than applying statistical analysis to all of the data in order to account for the enhancements through a series of dimensionless groups effort has been directed at determining the effect of geometry on friction factor then using the correlation for friction factor in a general relationship for heat transfer coefficient. The performance of each individual surface is then examined. The key question becomes: does the correlation predict the observed behavior within the accuracy that can be expected for the experiment? Given the warnings issued by Kays & London the treatment of the data base as single entity is ill advised. For instance, in the case of off-set fins they state: “Fins of this type are generally constructed by a machine cutting process that inevitably leaves a slightly bent and scarfed fin edge that differs depending upon fin material and the character of the cutting tool. Since a few ten-thousandths of an inch of scarfing can have a considerable effect, it is difficult to either dimensionally describe this effect or exactly duplicate one of the test surfaces.” General correlations resulting from statistical analysis of the full database will have distortions resulting from inconsistency in manufacture and experimental inaccuracy embedded in the result.

3 RESULTS AND DISCUSSION

3.1 The Friction Factors

In their research work on wavy fin surfaces [2-3] the authors endeavored to develop a friction factor correlation that would have a limiting value equal to plain fin performance as the fin geometry approached that of the plain fin. They achieved this by examining data at the extremes of the Reynolds number range covered in the experiments. For fixed values of Reynolds number, they compared fin performance with that of a plain fin having the same plate spacing and fin density (and, hence, the same hydraulic diameter). The ratio of values was then related to fin geometry.

The performance of strip fins can be expected to be governed by two geometrical factors. Strip length and spacing are typically in millimeters as this is a common unit for precise measurement in engineering. The ratio of strip length to fin spacing will characterize the frequency at which the flow field is disrupted. The disruption is caused by the flock blockage introduced by the off-setting of the strips. The extent of this disruption will be characterized by the ratio of fin thickness to fin clearance (fin spacing minus fin thickness). Therefore, the correlation between friction factor (as characterized by the ratio of strip-fin to plain fin values) and these geometrical ratios in turn was considered.

Looking at the performance under laminar flow conditions, as shown in Fig. 1, it is clear that the increase over what could be expected correlates with the ratio of fin spacing to strip length. This ratio (f_{strip}/f_{plain}) follows an equation of the form shown in Eq. (1):

$$\left(\frac{f_{strip}}{f_{plain}}\right)_{lam} = 1 + k_l \left(\frac{F_s}{S_l}\right) = 1 + k_l\ S_l\ F_s \tag{1}$$

Whereas k_l is an equation constant, Fs is fin spacing and S_l is strip length. The strip density is the number of interruptions per meter run. For a plain fin surface, this is zero, and the equations yield the plain fin result. The introduction of the flow blockage term does not improve the correlation. Therefore, this correction was adopted.

Looking at the performance of these surfaces under turbulent flow conditions (Reynold No. (Re) = 6000), and this time initially relating performance to the flow blockage effect. The results shown in Fig. 2 were obtained. This time, the introduction of the factor relating to the rate of disruption does result in an improvement in the correlation as clearly shown in Fig. 3.

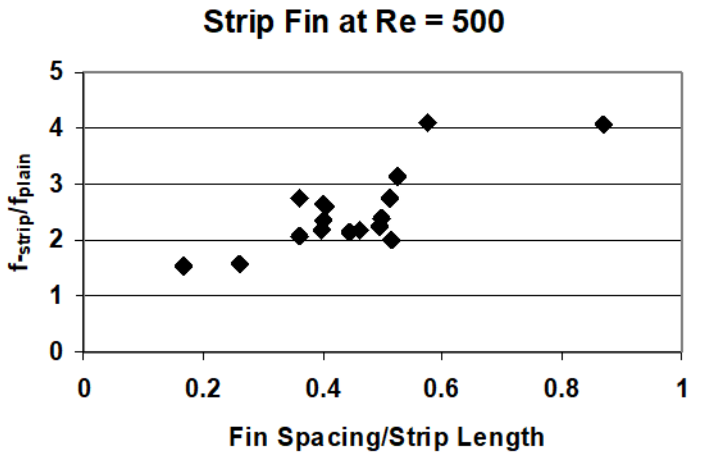


Fig. 1. Increase under laminar flow conditions

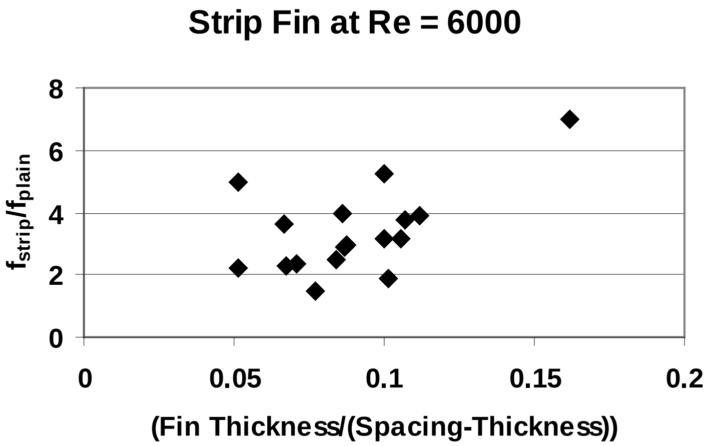


Fig. 2. Effect of Flow Blockage: Turbulent Flow Conditions

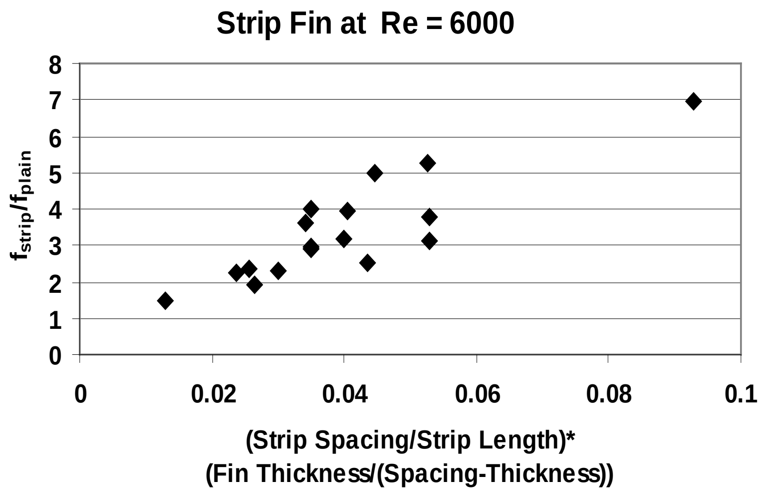


Fig. 3. Combined Effects of Disruption Rate and Flow Blockage

From both Fig. 2 and 3, it was observed that the increase in friction factor above the performance of plain fins follows the relationship presented in Eq. (2):

$$\left(\frac{f_{strip}}{f_{plain}}\right)_{turb} = 1 + \frac{K_T\ F_s}{S_l}\left(\frac{F_{th}}{F_s - F_{th}}\right) = 1 + K_T\ S_d\ F_s\left(\frac{F_{th}}{F_s - F_{th}}\right) \tag{2}$$

The best fit to the data appears to be achieved when $k_l = 2$ and $k_T = 60$. Whereas, S_d strip density and F_{th} is the fin thickness. A general equation is obtained by multiplying the plain fin friction factors by these correction factors (which again go to unity for the plain fin case) and substituting the results into the power law interpolation described earlier, as shown in Eq. (3):

$$f = \left[\left(\left(\frac{f_{strip}}{f_{plain}} \right)_{lam} \left(\frac{16}{\mathrm{Re}} \right) \right)^3 + \left(\left(\frac{f_{strip}}{f_{plain}} \right)_{turb} \left(\frac{0.078}{\mathrm{Re}^{0.25}} \right) \right)^3 \right]^{1/3} \quad (3)$$

Table 1 compares the predictions of this equation with the data of Kays and London [1], as shown in Figures 4 to 24 below.

Overall, the comparisons are very favorable. One exception is the 3/32-12.2 surface. It is interesting to note that this was the only test section fabricated from copper (the comments made by Kays and London [1] regarding the cutting of the strips may be relevant here). The other exceptions are the 1/8-16.00, 1/8-16.12 and 1/4-15.4 surfaces. The material of construction used in these test sections was aluminum, (which was the material used for most of the test sections). There does not appear to be any structural reason why the performance of these units should not be in line with that of the other test sections.

Table 1. Comparisons between the predictions of this equation and the data of Kays and London [1]
♦ experimental values; ─ predicted values;

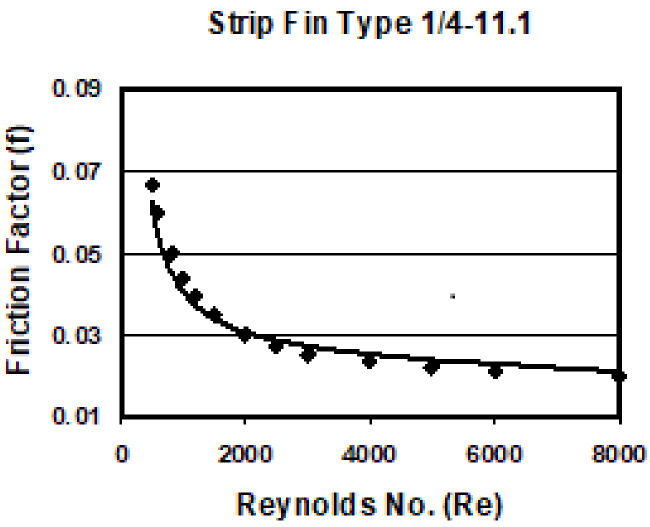


Fig. 4.

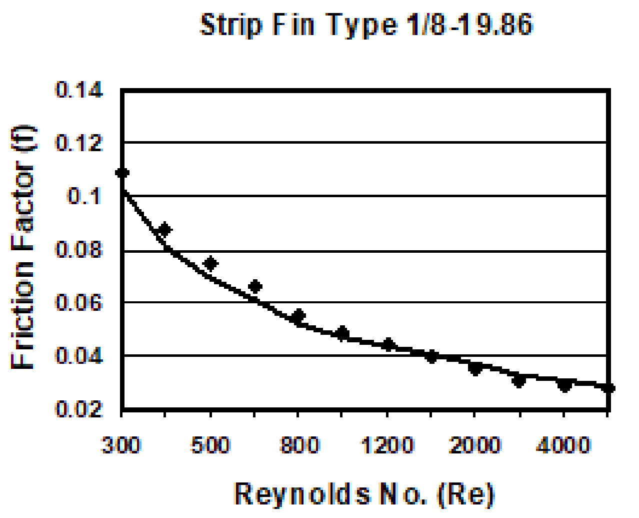


Fig. 5.

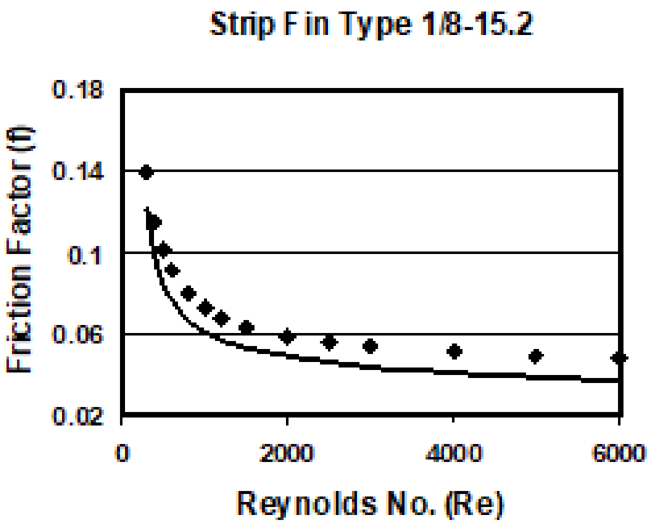


Fig. 6.

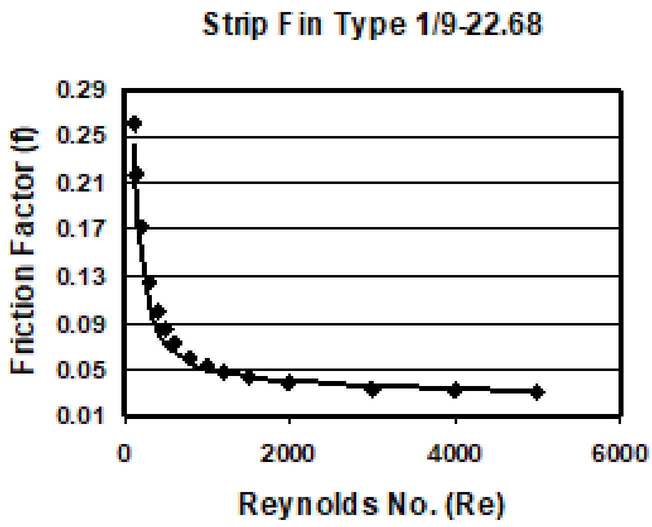


Fig. 7.

Table 1. Comparisons between the predictions of this equation and the data of Kays and London [1]
♦ experimental values; ─ predicted values;

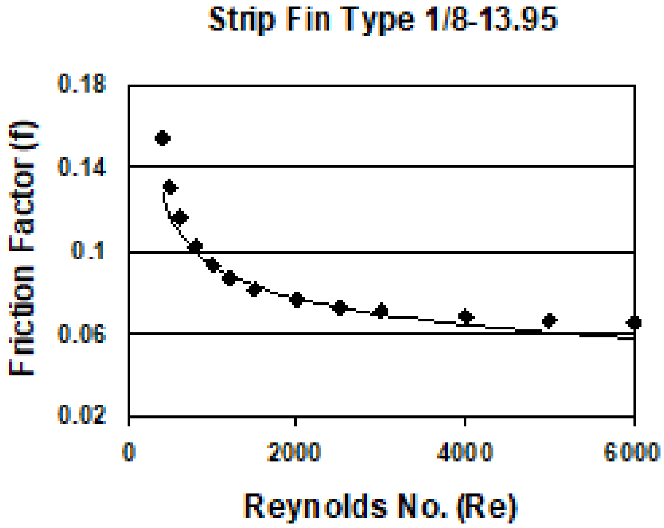


Fig. 8.

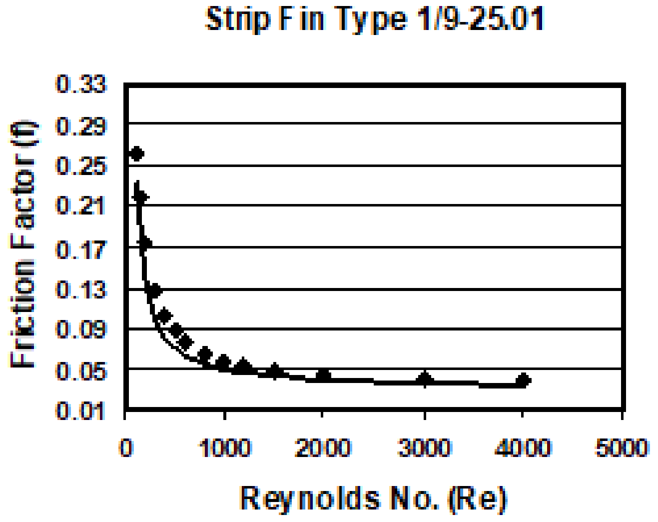


Fig. 9.

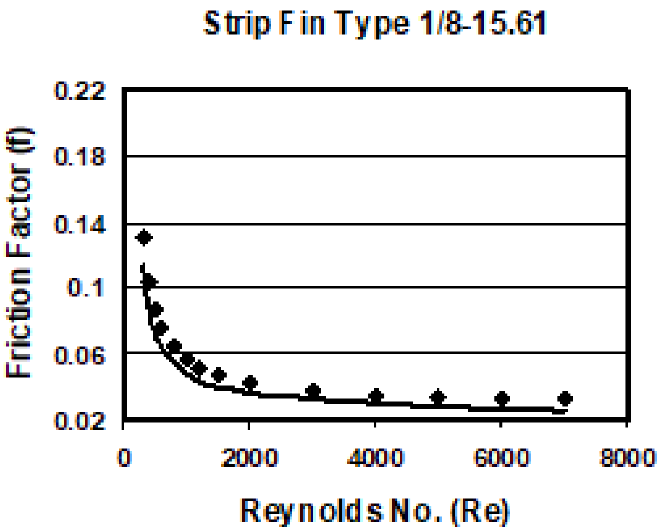


Fig. 10.

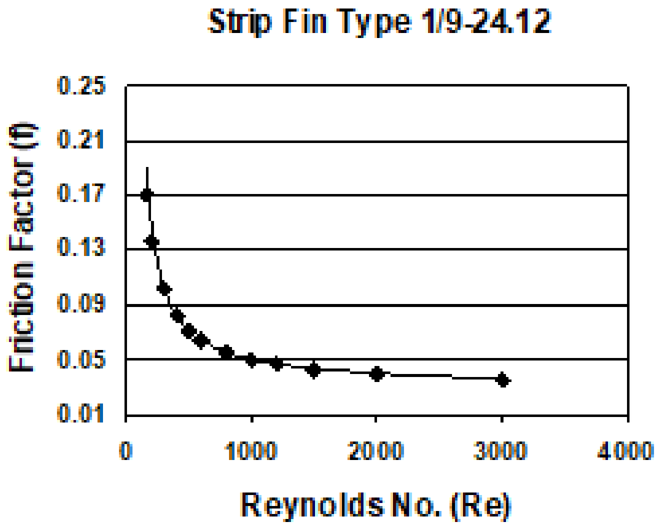
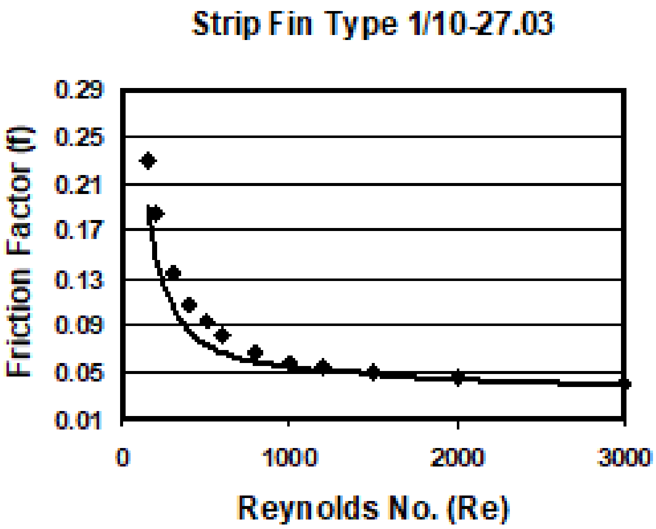


Fig. 11.



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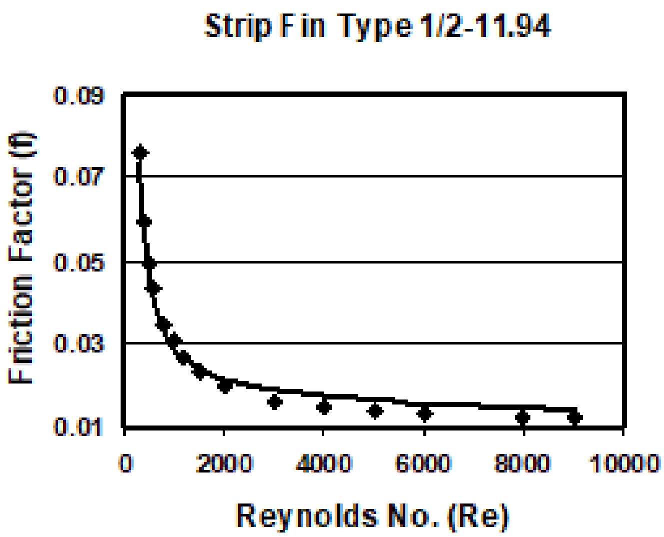


Fig. 13.

Table 1. Comparisons between the predictions of this equation and the data of Kays and London [1]
♦ experimental values; ─ predicted values;

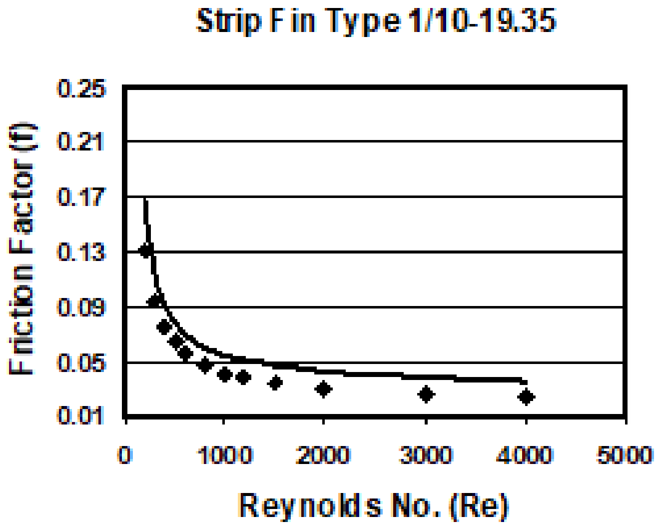


Fig. 14.

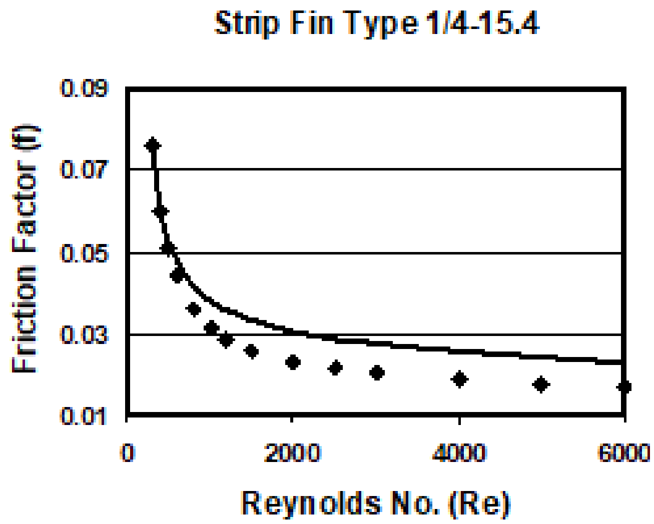


Fig. 15.

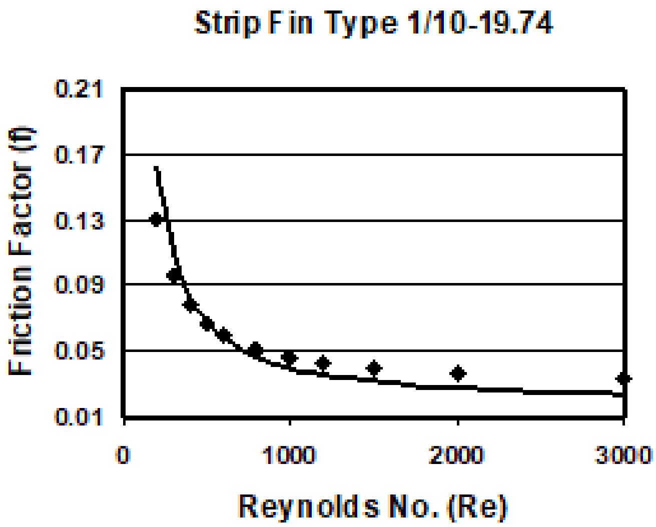


Fig. 16.

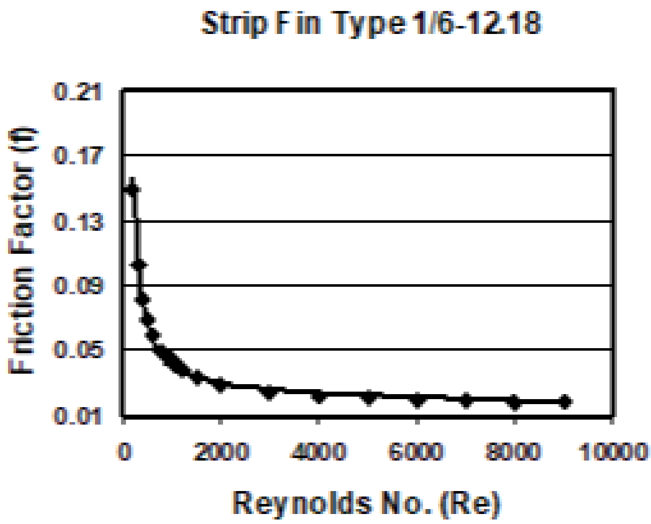


Fig. 17.

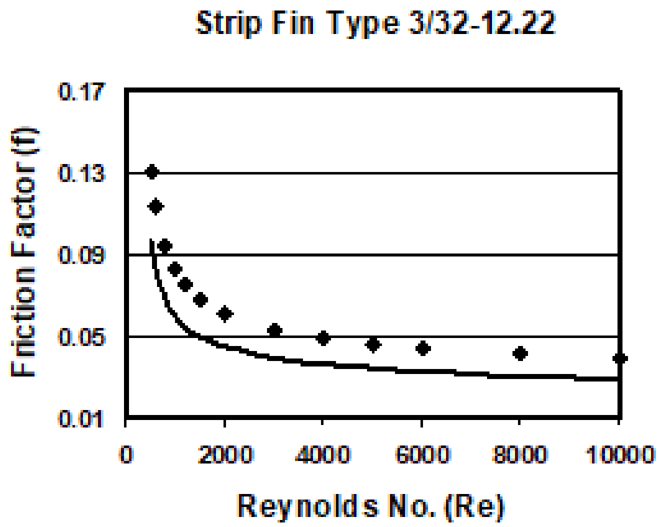


Fig. 18.

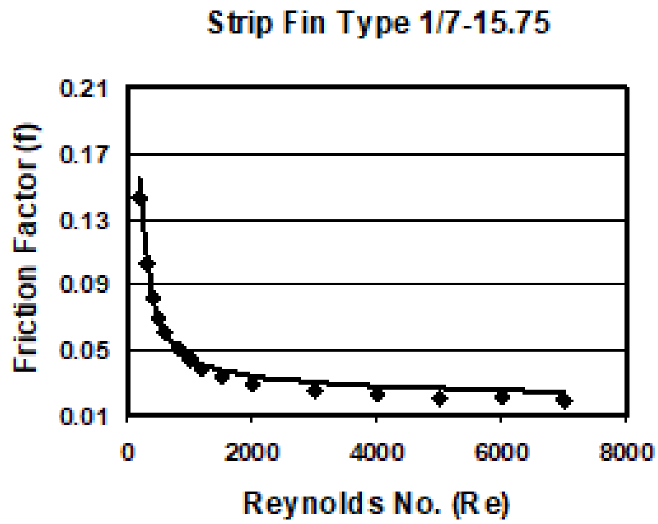


Fig. 19.

Table 1. Comparisons between the predictions of this equation and the data of Kays and London [1]
♦ experimental values; ─ predicted values;

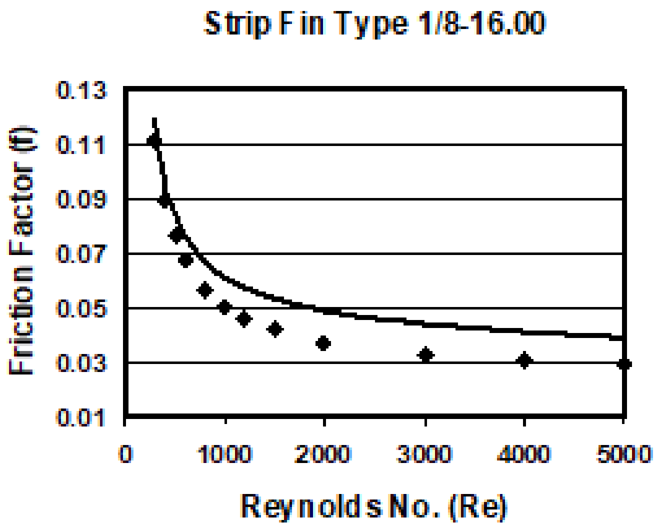
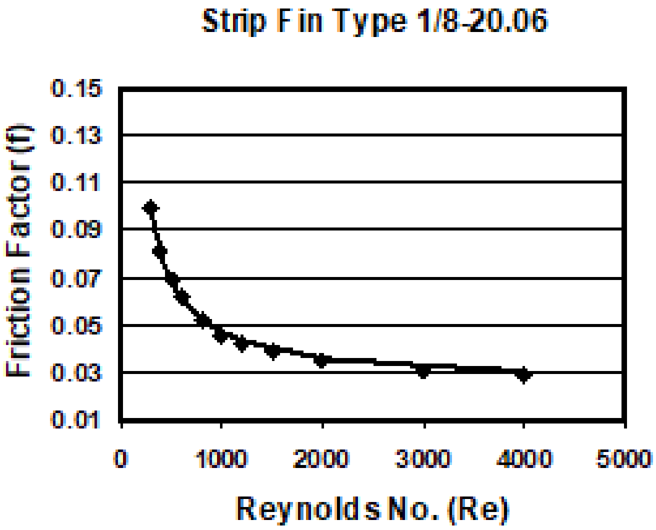


Fig. 20.



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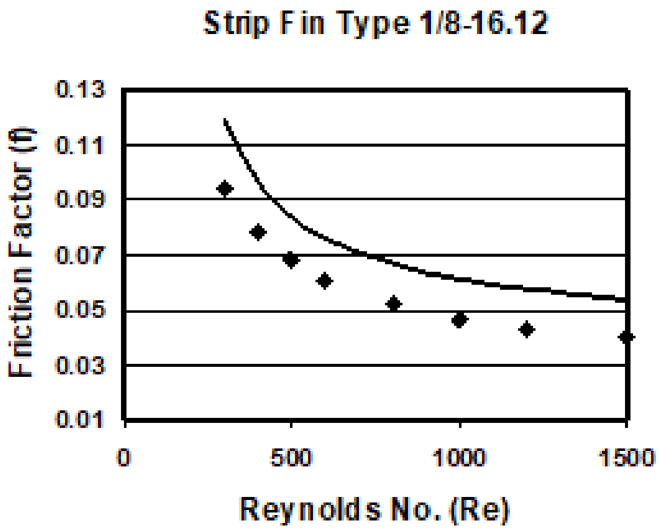
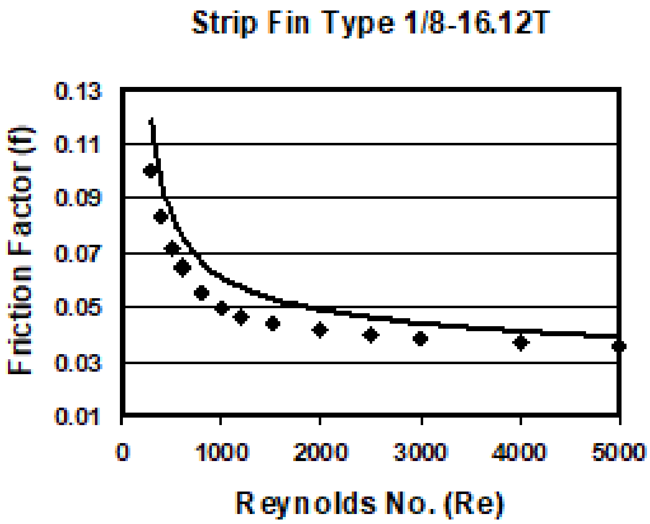


Fig. 22.



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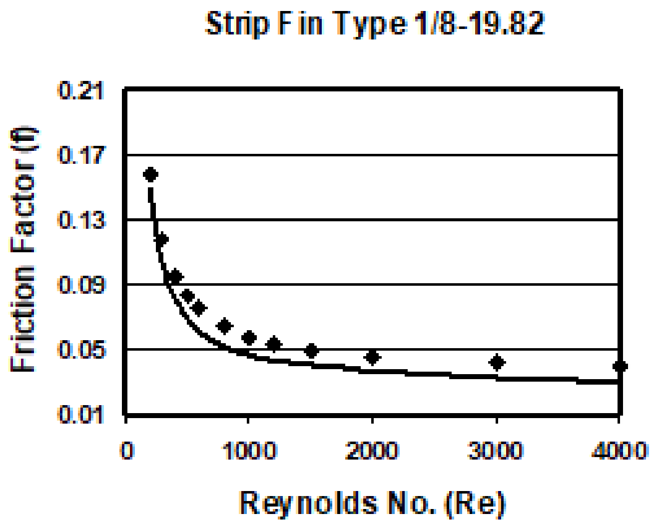

Fig. 24.

3.2 The j-Factors

First, the transition from laminar to turbulent flow occurred at a lower Reynolds number for wavy surfaces. Furthermore, the 'hump' that characterized this transition for heat transfer in round tubes and with plain fins was absent in the case of wavy fins. Consequently, rather than use the form of interpolation equation introduced by Churchill [13], a simple power law equation was used in terms of Nusselt number (Nu):

$$Nu = \left(Nu_{lam}^{n} + Nu_{turb}^{n} \right)^{1/n} \tag{4}$$

An exponent of (2) was found to be appropriate. A simple equation [1] was found to give good predictions in the turbulent flow region:

$$Nu = 0.0352\ \sqrt{f}\ \ \mathrm{Re}\ \ \mathrm{Pr}^{1/3} \tag{5}$$

Another difference was the 'characteristic length' appearing in the Graetz number (Gz) used for the prediction of heat transfer in the laminar flow regime. With plain fins, the characteristic length was the fin length. With wavy fins, the characteristic length was the wave pitch. Given this adjustment in characteristic length, heat transfer in the laminar flow regime for both plain rectangular and wavy fins was found to be well correlated by [11,14 -15]:

$$Nu = \left[\ 3.66^3 + 0.7^3 + \left(1.77\ Gz^{\ 1/3} - 0.7\right)^{\ 3} \right]^{\ 1/3} \tag{6}$$

$$j = Nu\ /\ \mathrm{Re}.\mathrm{Pr}^{1/3} \tag{7}$$

Whereas, j is colburn factor which is used to compare heat transfer efficiency across different systems. The j factor takes into account the mode of the fluid flow in Re and the impact of the physical fluid properties in Pr.

Finally, the same set of equations for strip fin surfaces were applied. Here, the characteristic length for use in the Graetz Number is four times the strip length. Table 2 provides comparisons between measured and predicted j factors, as presented in figures 25 to 45. Again, the comparison between prediction and measurement is good over the full Reynolds Number range. The poorest comparisons are found for the 1/8-13.95, 1/9-25.01 and 1/10-27.03 surfaces. The friction factor predictions for these surfaces were very good. The j-factors for which friction factors were poorly predicted are well predicted by the proposed general correlation.

Table 2. Comparisons between measured and predicted j factor

♦ experimental values; — predicted values;

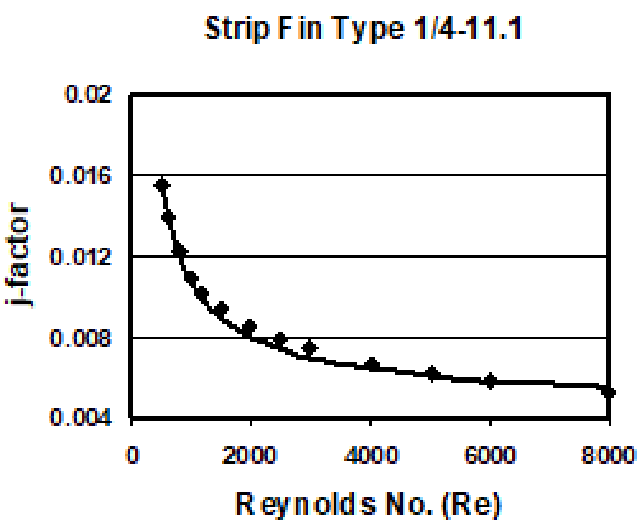

Fig. 25.

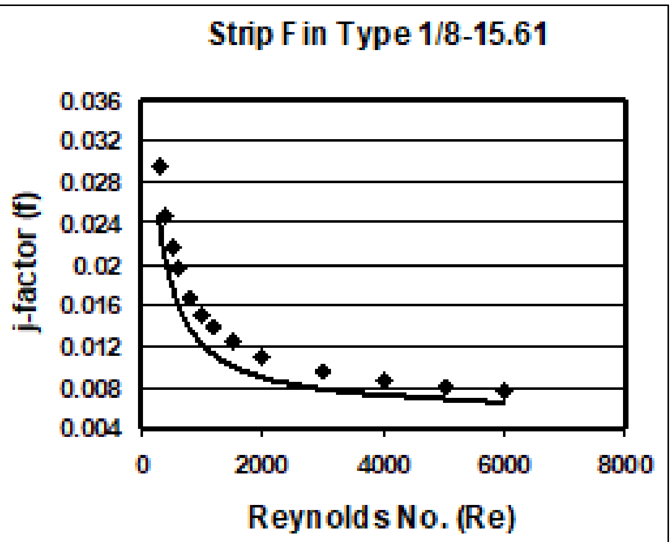

Fig. 26.

Table 2. Comparisons between measured and predicted j factor

♦ experimental values; — predicted values;

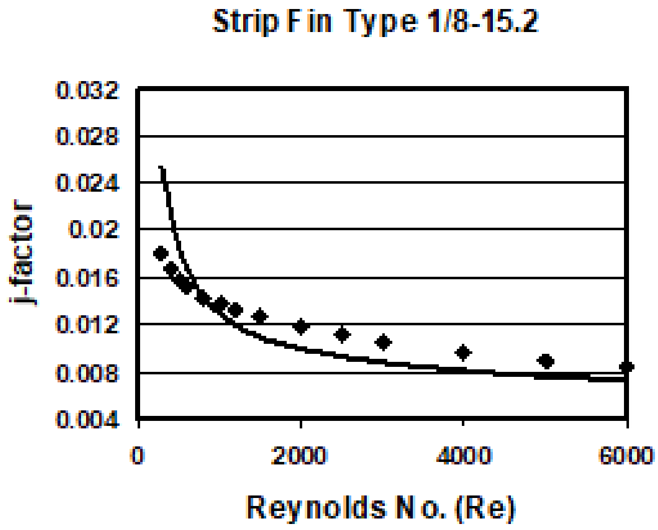

Fig. 27.

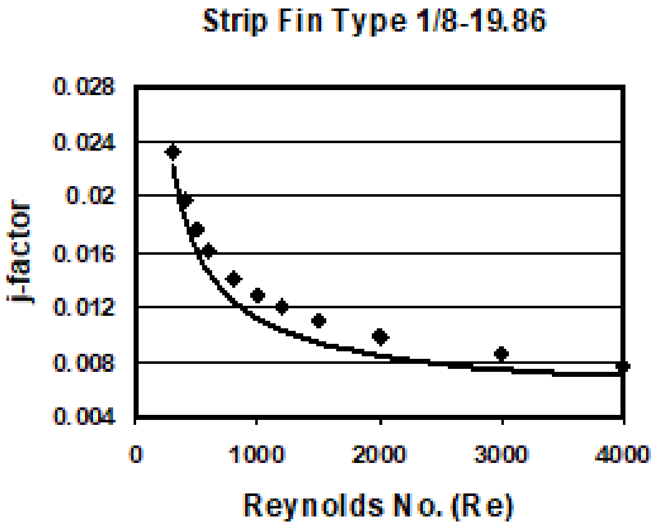

Fig. 28.

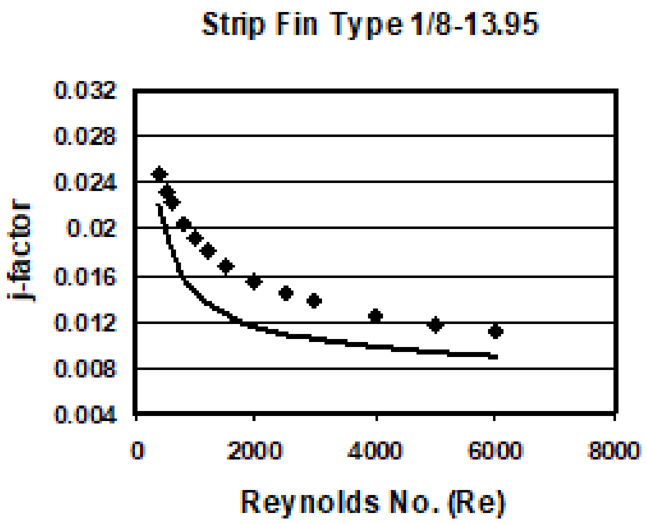

Fig. 29.

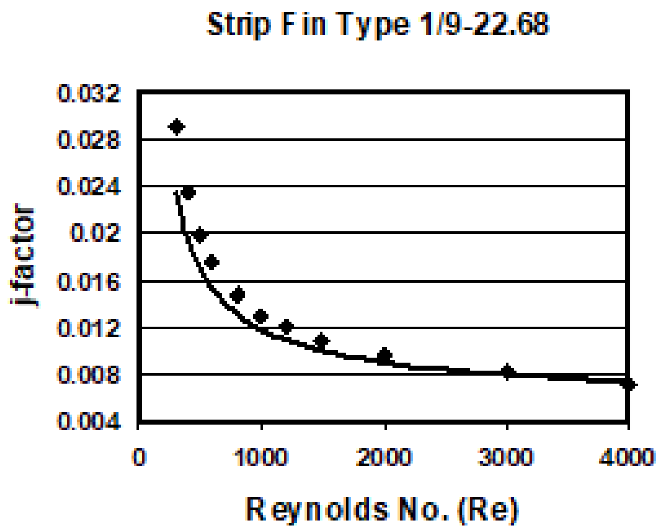

Fig. 30.

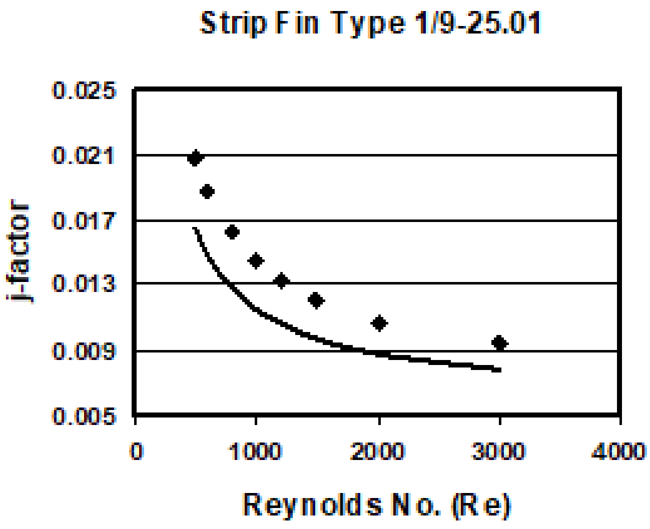

Fig. 31.

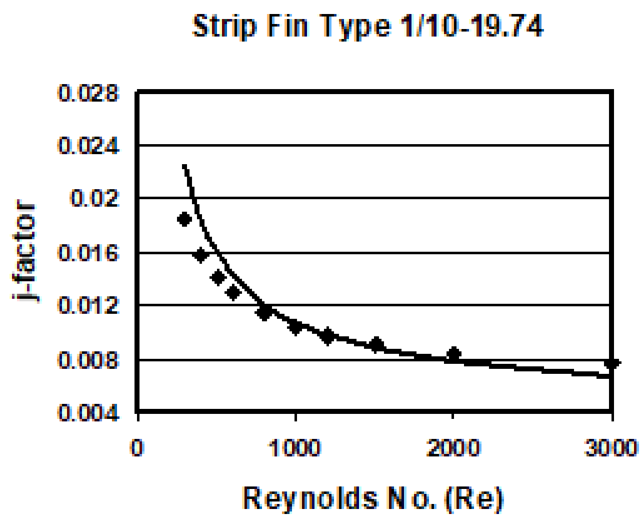

Fig. 32.

Table 2. Comparisons between measured and predicted j factor

♦ experimental values; — predicted values;

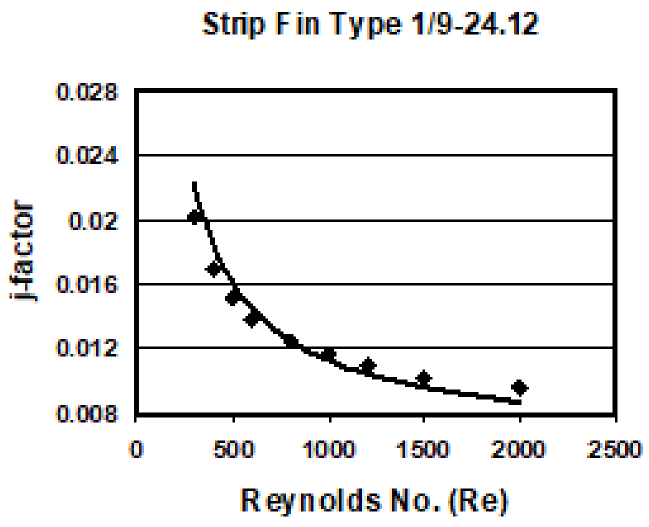

Fig. 33.

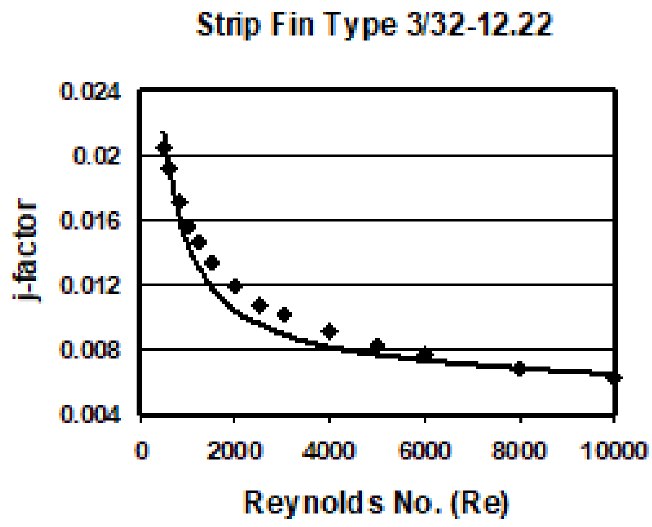

Fig. 34.

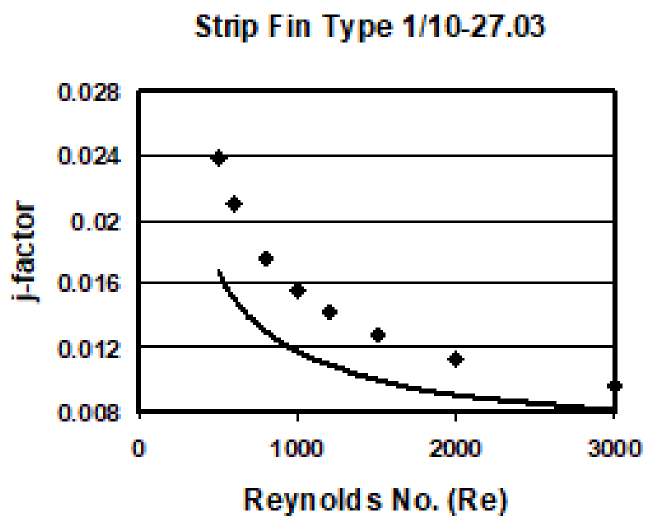

Fig. 35.

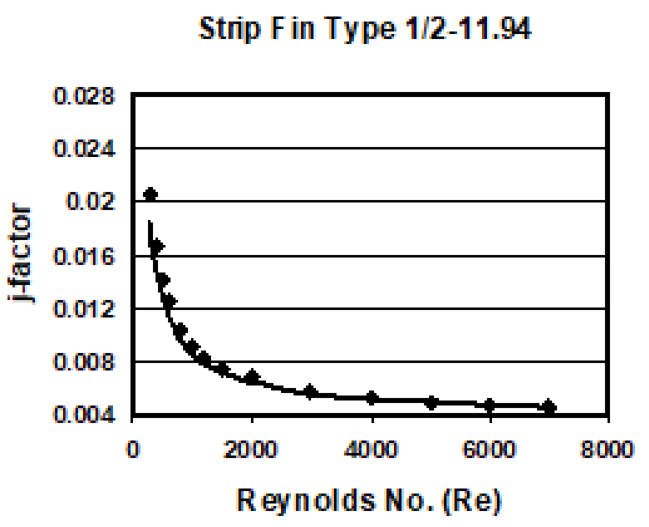

Fig. 36.

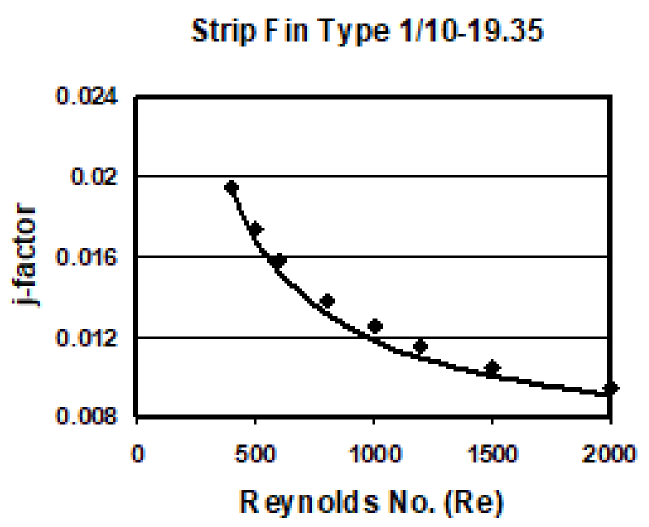

Fig. 37.

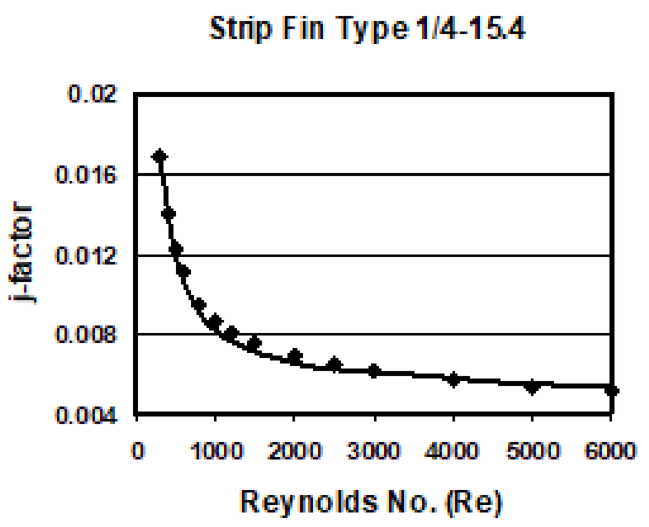

Fig. 38.

Table 2. Comparisons between measured and predicted j factor

♦ experimental values; — predicted values;

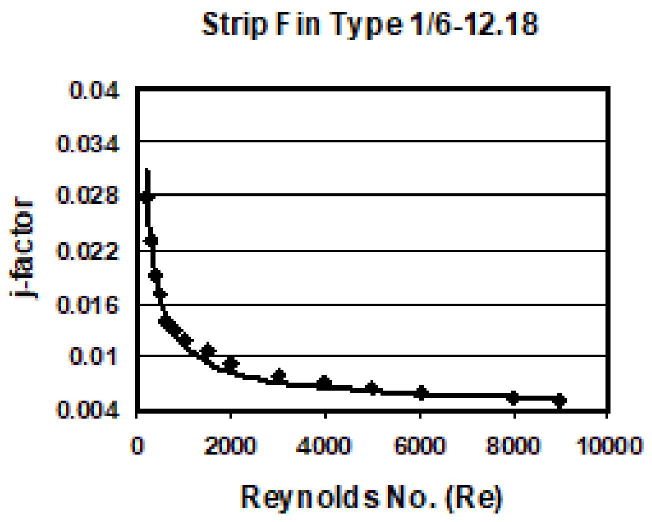

Fig. 39.

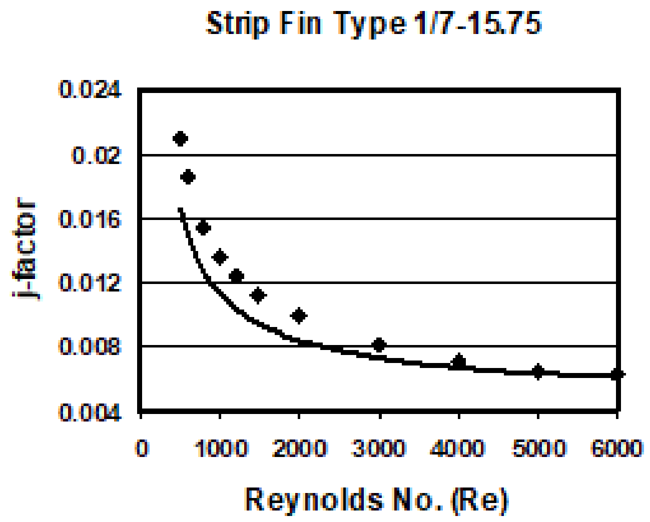

Fig. 40.

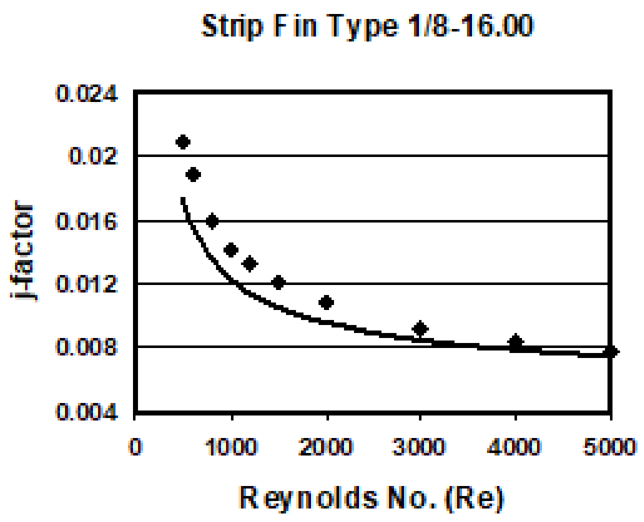

Fig. 41.

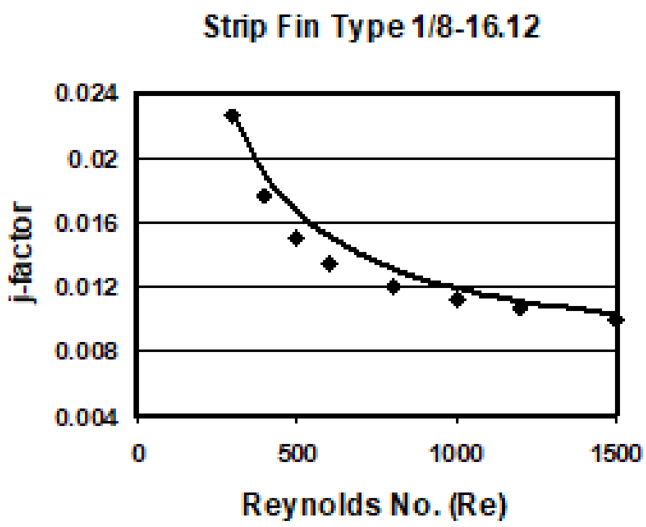

Fig. 42.

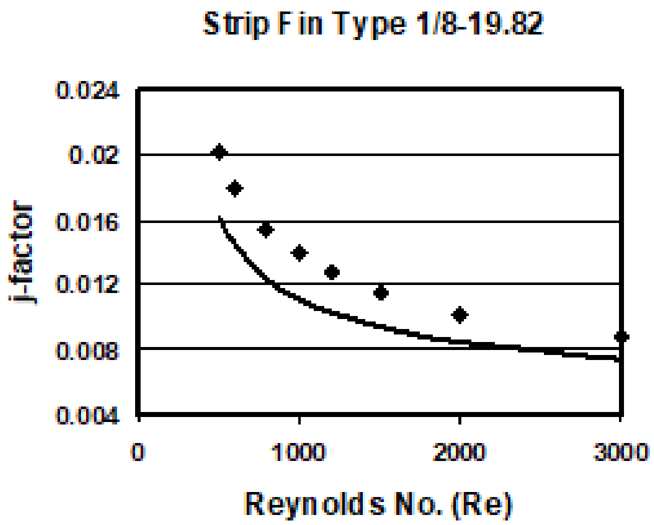

Fig. 43.

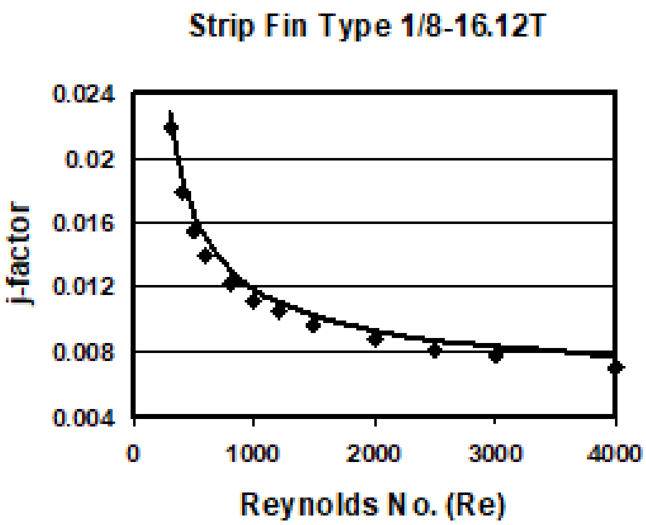

Fig. 44.

Table 2. Comparisons between measured and predicted j factor

♦ experimental values; ─ predicted values;

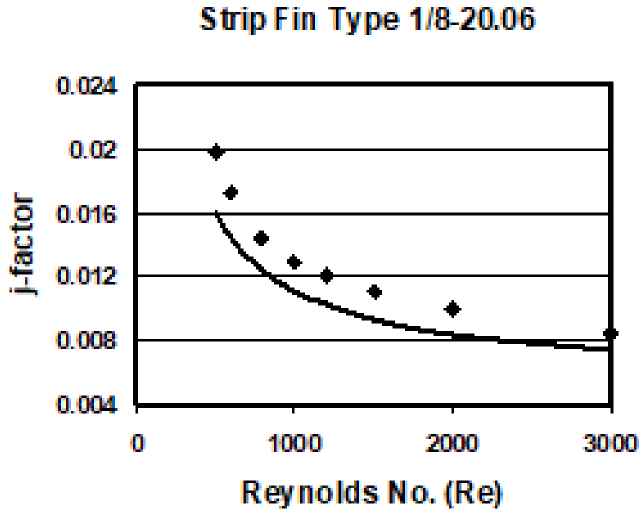

Fig. 45.

4 CONCLUSIONS

General correlations have been developed for the thermal-hydraulic performance of strip fin plate-fin surfaces. The ratio in both laminar and turbulent conditions was successful in presenting the fluid behavior. The correlations of equations (1-3) for friction factor benefit from a systematic move to plain fin performance as geometry approaches that limit. Also, 'characteristic length' appearing in the Graetz number (Gz) is used for the prediction of heat transfer and to calculate the j-factor from the Nu number through equations (4-7). Laminar flow heat transfer is well predicted if the Graetz Number is based on four-time strip fin length. Turbulent flow heat transfer is found to be proportional to the square root of the friction factor. The comparison between prediction and measurement is good. Any ability to derive a general correlation is testimony to the care and accuracy of the original experiments. The scope for further experimental work has been identified.

5 ACKNOWLEDGEMENT

The paper is dedicated to my late partner Professor Graham T. Polley. He graduated as a chemical engineer from the Loughborough University of Technology in 1969. As a PhD holder, he worked at the UK National Engineering Laboratory (NEL). In 1978, he was appointed UK representative on the International Energy Agency Executive Committee and joined UMIST as project director for the Centre for Process Integration in 1985. He retired from UMIST in 1996. In 1990, he was awarded the Moulton Medal by the IChemE for his work on oil refinery retrofit. Also, a past president of the UK Heat Transfer Society. He kept working as a visiting Professor in the Department of Chemical Engineering at the University of Guanajuato, Leon, Guanajuato, Mexico. He was involved in wide research works such as boiling and condensation heat transfer, two-phase flow, heat exchanger design and fouling, and heat recovery networks.

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7 NOMENCLATURE

d_h	Hydraulic diameter (m)
F_s	Fin spacing (m)
F_{th}	Fin thickness (m)
f	Friction factor
Gz	Graetz Number (dimensionless)
j	Colburn factor
k_l	Constant in Eq. (1)
k_T	Constant in Eq. (2)
Nu	Nusselt Number (dimensionless)
Pr	Prandtl Number (dimensionless)
Re	Reynolds Number (dimensionless)
S_d	Strip density (kg/m3)
S_l	Strip length (m)

Subscript

lam	laminar flow
turb	turbulent flow

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