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Correlations for heat transfer and flow friction characteristics of compact plate-type heat exchangers

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Abstract—Correlations for heat transfer and flow friction coefficients are provided for plane parallel plates and offset strip-fin plates over the ranges used in compact heat exchangers. Closed form expressions have been used to present these correlations. The proposed correlations allow one to adequately predict experimental data available for the heat exchanged and pressure losses in compact plate-type heat exchangers. The correlations cover continuously the full range from laminar to turbulent flow, for both short and long pipes. Suggestions to extend the correlations to other flow conditions are provided.

1. INTRODUCTION

PLATE-TYPE heat exchangers are widely used in engineering devices because of the need for small size, lightweight heat exchangers. This need is particularly important in automobiles and aircraft. In order to adequately predict the performance of water/engine oil compact heat exchangers, a literature survey was made to find heat transfer correlations for both parallel plates and offset strip-fin plates. A classical reference for fully developed turbulent flow in smooth pipes is that of Dittus and Boelter [1]. A similar correlation was proposed by Sieder and Tate [2], and accounted for fluid property variations. A somewhat simpler empirical correlation for laminar flow in pipes was also proposed. Analytical solutions exist for round tubes, rectangular ducts and parallel plates [3–5]. These solutions include the developing thermal boundary layers with either uniform heat flux or constant temperature, although they are sometimes a little cumbersome to use.

In engineering applications, it is common to find a transitional regime between laminar flow and fully turbulent flow. This transitional regime is characterized by a Reynolds number between 2000 and 10 000. In addition the pipes or passages are short, in the sense that they have length-to-diameter ratios smaller than 50. In the case of long passages with turbulent flow, the correlations valid for circular pipes may be used for non-circular cross-sections, if the Nusselt and Reynolds numbers are based on the hydraulic diameter. In the case of laminar flow in long non-circular cross-sections, the use of the hydraulic diameter is not a good approach, particularly if the section involves sharp corners. Different values of the Nusselt number have to be used depending on the type of passage cross-sections. These values of the Nusselt number appear tabulated in, for instance ref. [6], for both constant temperature and constant heat flux boundary conditions, always referred to fully developed laminar flow in long passages. Few references discuss the correlations for laminar flow in short passages. The situation is worse for the transitional Reynolds number range, even for long pipes.

A number of workers [3] summarize heat transfer and pressure drop experimental data for rectangular ducts, offset strip-fin plates and other different shapes of fin-plates design. Offset strip-fin plates are frequently used in heat exchangers, as shown in Fig. 1. The fins are offset to prevent fully developed flow and thereby to take advantage of the increased heat transfer characteristics due to the entry length effect. A prediction model for a water/engine oil compact heat exchanger has been developed by the authors. The heat exchanger performance can thus be predicted and the effects on performance of the various geometric parameters can be assessed. In order to do this good correlations for the Nusselt number and the flow friction coefficient have to be implemented in the model. Two of the problems that the authors found have been that the compact heat exchanger flow passages are short and the flow regime is sometimes transitional. (Since this type of heat exchanger is used in automotive engines, the operating conditions are very variable, particularly the flow rates.)

The purpose of this paper is to provide empirical correlations for heat transfer and flow friction for the conditions found in plate heat exchangers. The correlations for offset strip-fin plates should be considered particularly for the geometry tested in the present work. However, the correlations for plane parallel plates are of general applicability, especially the ones for the Nusselt number. These can be easily extended to, for instance, circular pipes, as suggested below. The mathematical expressions cover the laminar, transitional and turbulent regimes continuously.

NOMENCLATURE

a	free flow area [m^2]
A	heat transfer area [m^2]
C	constant
D	hydraulic diameter, $(4a/p)$ [m]
f	friction factor
$f(Re_D)$	correction function for transitional flow ($2000 < Re < 10\,000$)
h	heat transfer coefficient [W m^{-2} K^{-1}]
j	Colburn factor, $(St\, Pr^{2/3})$
k	thermal conductivity [W m^{-1} K^{-1}]
L	length in the flow direction [m]
$\dot{m}$	mass flow rate [kg s^{-1}]
Nu	Nusselt number
p	wetted perimeter [m]
Pr	Prandtl number
Re	Reynolds number
St	Stanton number
U	overall heat transfer coefficient [W m^{-2} K^{-1}].

Greek symbols

α	constant
δ_1	correction coefficient for short pipes when $Re < 2000$
Δp	pressure losses in flow passages [Pa]
Δx	plate thickness [m]
ε_1	correction coefficient for short pipes when $Re > 10\,000$
η	coefficient in expression (6)
μ	fluid viscosity [Pa s]
ρ	fluid density [kg m^{-3}]
τ	dimensionless variable in expression (8).

Subscripts

1	water side
2	oil side
D	based on the hydraulic diameter
I	intermediate (transitional) regime
L	laminar regime
S	short pipes ($L/D < 50$)
T	turbulent regime
w	plate wall.

They are valid for short pipes and in the limit for long pipes they reduce to the more standard expressions usually found in the literature. Globally the expressions proposed have been found acceptable when comparing the results of the model with experimental data [7].

2. HEAT TRANSFER CORRELATIONS

Mathematical expressions are proposed for heat transfer coefficients over three Reynolds number ranges: laminar ($Re_D < 2000$), transitional ($2000 < Re_D < 10\,000$) and turbulent ($Re_D > 10\,000$). The correlations for the Nusselt number are written in terms of Reynolds and Prandtl numbers and other dimensionless variables. These correlations are presented first for plane parallel plates, in a form that can be regarded as general. Secondly, correlations for a particular geometry of offset strip-fin plates are presented.

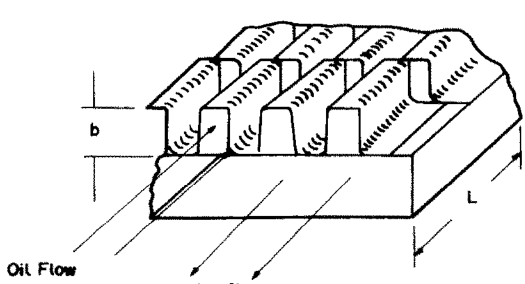

FIG. 1. Offset strip-fin plates heat exchanger geometry.

2.1. *Short passages between plane parallel plates*

For passages between plane parallel plates with small length-to-hydraulic diameter ratios (less than 50), the following relationships have been used.

(i) *Turbulent flow* ($Re_D > 10\,000$). In the fully turbulent region, the Nusselt number expression for long passages, Nu_T, takes the form of the standard correlation of Dittus and Boelter [1]

$$Nu_T = 0.021 Re_D^{0.8}\, Pr^{0.43}. \tag{1}$$

In the present study this correlation is corrected by a coefficient ε_1 to account for short pipe conditions as follows.

$$Nu_{TS} = Nu_T \varepsilon_1 \tag{2}$$

where Nu_{TS} is the Nusselt number based on the hydraulic diameter for short pipes ($L/D < 50$), and ε_1 is the correction coefficient for short pipes in the range $Re_D > 10\,000$. Vsórov [8] provides tabulated values of ε_1 in terms of the length-to-hydraulic diameter ratio and the Reynolds number. A plot of the values of ε_1 is shown in Fig. 2. For instance, for a length-to-diameter ratio of about 20, a regression fit of ε_1 in terms of Reynolds number gives

$$\log \varepsilon_1 = 0.05723 - 0.05175 \log (Re_D/10^4) + 0.06182[0.3705 \log (Re_D/10^4) + 0.2589]^2. \tag{3}$$

If the length-to-diameter ratio is smaller than 20, higher values of ε_1 are obtained, showing the increased

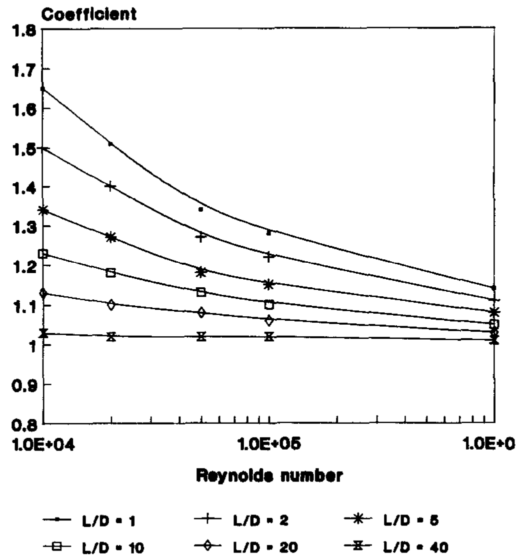

FIG. 2. Correction coefficient ε_l for turbulent flow in short pipes for several values of the length-to-diameter ratio.

influence of the entry length. On the other hand, when the length-to-diameter ratio approaches the value 50, the correction factor ε_l goes to 1 and expression (2) then reduces to expression (1).

(ii) *Laminar flow* ($Re_D < 2000$). In the present study the Nusselt number for a short duct in the laminar regime, Nu_{LS}, is assumed to be the one corresponding to the same type of duct with infinite length Nu_L, but corrected with a factor δ_l :

$$Nu_{LS} = Nu_L\,\delta_l. \tag{4}$$

In the present case of a passage formed by plane parallel plates, Nu_L equals 7.54 [6]. For a circular pipe, Nu_L adopts the value 3.66. In all cases δ_l (dimensionless) is the correction coefficient for short pipes when $Re_D < 2000$ and depends on the length-to-diameter ratio and the Reynolds and Prandtl numbers. The expression proposed by Sieder and Tate for circular pipes [2] can be generalized for non-circular cross-sections accounting for the reference value $Nu_L = 3.66$, thus giving

$$\delta_l = 0.508\left[\frac{D}{L}Re_D\,Pr\right]^{1/3}. \tag{5}$$

Expression (4), with δ_l defined by (5), provides the mean value of the Nusselt number from the entry section up to a distance L. The thermal entry length in which expression (5) applies depends on the value of the variables inside the brackets. As stated by Kays and Crawford [5], the correction coefficient for thermal entry effects should be equal to one if $L/D > 0.1$ $(Re_D\,Pr)$.

If the temperature difference between the wall and liquid bulk is higher than 6°C (60°C for gases), Sieder and Tate suggested to account for the variation of the transport properties by modifying the correction factor as follows:

$$\delta_l = 0.508\left[\frac{D}{L}Re_D\,Pr\right]^{1/3}\left[\frac{\mu}{\mu_w}\right]^{0.14} \tag{5'}$$

where μ and μ_w are the fluid viscosities evaluated at the bulk temperature and wall temperature respectively. A similar viscosity correction factor could be introduced in expression (1).

(iii) *Transitional regime* ($2000 < Re_D < 10\,000$). In this transitional regime the authors' aim has been to find an expression to provide a smooth transition between the laminar and the turbulent regimes. One of the main justifications to do that is the need for an expression providing continuous values in the full Reynolds number range and that can be used in a computer. Otherwise the values of Nusselt number have a gap between the correlations valid for laminar flow (up to $Re_D = 2000$) and those valid for turbulent flow (above $Re_D = 10\,000$).

Vsórov [8] provides a clue to cover the gap. This researcher proposes the use of the following expression for the Nusselt number in the intermediate (transitional) regime for long circular passages:

$$Nu_l = Nu_T(Re_D = 10^4)\eta \tag{6}$$

where Nu_T is given by expression (1) particularized for $Re_D = 10^4$. The coefficient η depends on the Reynolds number and is given graphically, decreasing from unity when $Re_D = 10^4$ up to a value close to 0.6 when $Re_D = 5\times10^3$. However, expression (6) lacks the flexibility needed to account for other flow conditions. In addition, it is difficult to extrapolate expression (6) down to $Re_D = 2000$ while guaranteeing continuity with the Nusselt number corresponding to laminar flow.

In order to overcome these problems the authors propose to use the following expression for the Nusselt number in the intermediate (transitional) regime in short passages:

$$Nu_{lS} = Nu_{LS}(Re_D = 2\times10^3)+[Nu_{TS}(Re_D = 10^4) - Nu_{LS}(Re_D = 2\times10^3)]f(Re_D) \tag{7}$$

where $f(Re_D)$ is the function of the Reynolds number that ensures a smooth transition between the Nusselt numbers corresponding to the limits of laminar and turbulent flows ($Re_D = 2000$ and $Re_D = 10\,000$, respectively). A power-exponential function adapts well for this. The expression used is as follows:

$$f(\tau) = 2.54[1-\exp(-0.5\tau^{1.5})] \tag{8a}$$

$$\tau = (Re_D-2000)/8000. \tag{8b}$$

In expressions (8a) and (8b), τ has the meaning of a non-dimensional value related to the Reynolds number. For $Re_D = 2000$, $\tau = 0$ and $f(\tau) = 0$, while $\tau = 1$ gives $f(\tau) = 0.9994 \simeq 1.0$.

Expressions (2), (4) and (7) appear plotted together in Fig. 3. The graph represents the Nusselt number for a rectangular duct (aspect ratio of 5) as a function of Reynolds number for the three flow regimes discussed above. The values are computed for length-to-diameter ratios of 10, 20 and 50, and a Prandtl number of 2 (for water). The effects of the length of the duct are more important in the laminar regime, with the Nusselt number decreasing as the length increases. This effect presents the same trend in the turbulent regime, although it is less notorious. In the intermediate range, the values provided by expression (7) adapt to provide the continuity sought for all lengths. It should be recalled that in turbulent flow, long pipes are assumed to be those for which $L/D > 50$, and then $\varepsilon_1 = 1$. For laminar flow with a Prandtl number of 2, a value of at least $L/D > 100$ is needed to assume a long pipe condition ($\delta_1 = 1$). Expression (7) is still valid if the appropriate values of Nusselt number for long pipes at $Re_D = 2000$ and $Re_D = 10\,000$ are used.

2.2. *Offset strip-fin plates*

The offset strip-fin geometry that has been studied is characterized by the following quantities: core plate spacing $b = 1.95$ mm, core length $L = 44$ mm, hydraulic diameter $D = 0.93$ mm, and the heat transfer surface area $A = 0.012$ m^2. The total number of fluid passages varies from $N = 4$ to $N = 12$. As commonly done when referring to this type of plate, the heat transfer correlation is written in terms of the Colburn factor j. In this work, the following non-dimensional correlation for the Colburn factor j has been obtained:

$$j = 0.0944 Re_D^{-0.353}. \tag{9}$$

This correlation was obtained from the heat transfer data available for a set of heat exchangers with different numbers of fluid passages and varying operating conditions (flow rates especially). For comparison purposes, the Nusselt number can be calculated as

$$Nu = j\, Re_D\, Pr^{1/3} \tag{10}$$

which gives:

$$Nu = 0.0944 Re_D^{0.647}\, Pr^{1/3}. \tag{11}$$

Expression (11) is plotted in Fig. 4 giving the Nusselt number as a function of Reynolds number for the offset strip-fin plates with Prandtl numbers of 1, 2, 10 and 100. The higher values of the Prandtl number are representative of viscous fluids like lubricating oils, as in the authors' case. The length-to-diameter ratio of the oil passages in the heat exchanger is about 48, although that value should be interpreted differently from the case of plane parallel plates.

3. FRICTION FACTOR

As in the case of the heat transfer correlations, suitable expressions for the friction factor were sought in order to predict the performance of the heat exchangers for which experimental data have been available. The prediction model considers the pressure losses separately for each fluid and accounts for the entrance and exit losses, and the core losses. The friction factor expressions shown below refer to the latter type of pressure losses, which depend on the geometry of the flow passages and Reynolds number. Once the friction factor is known, the core pressure losses can be calculated as

$$\Delta p = \tfrac{1}{2} f \rho c^2 \tag{12}$$

where ρ is the fluid density and c the flow speed through the plate passages.

The expressions used for the friction factor f (dimensionless) are the following.

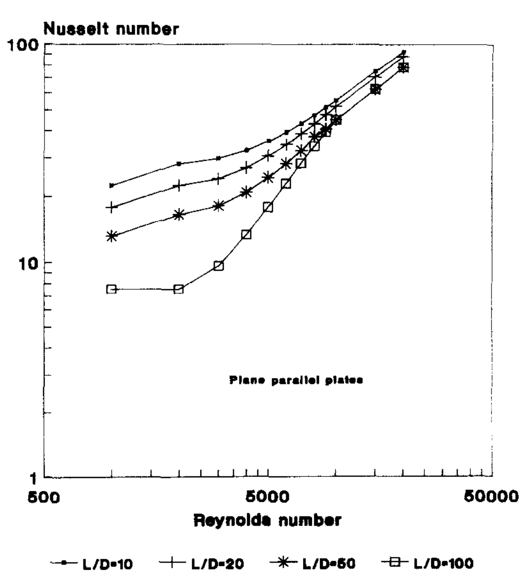

FIG. 3. Nu_{DL} vs Re_D for several L/D ratios.

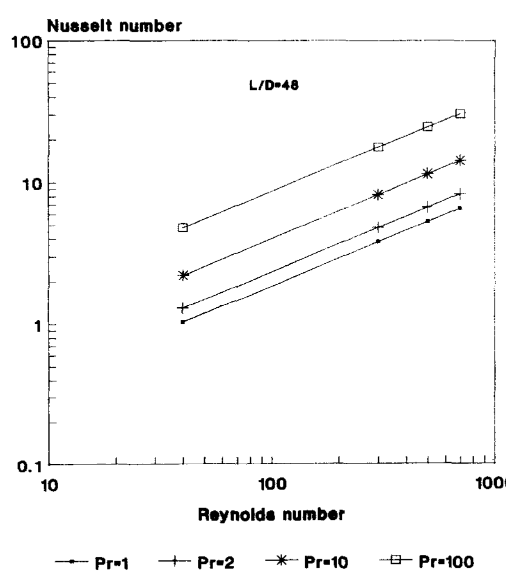

FIG. 4. Nu_{DL} vs Re_D for oil passages.

3.1. *Plane parallel plates*

$$f = (2 \log Re_D - 2.322)^{-2}. \quad (13)$$

This relation has been obtained by slightly modifying the coefficients of an expression given by Rohsenow *et al.* [6], in order to get an adequate match with the available experimental data for the heat exchanger pressure losses in the water side. As pointed out by Rohsenow, expressions similar to expression (13) have the virtue of explicitness, thus simplifying the implementation in a computer program. The Reynolds number in expression (13) is valid for $Re_D > 4000$. When needed a linear variation between the value for $Re_D = 4000$ ($f = 0.042$) and the value for laminar flow at $Re_D = 2000$ ($f = 64/Re_D = 0.030$) has been performed.

3.2. *Offset strip-fin plates*

For the particular geometry adopted, an expression that fits the core pressure losses in the oil side is the following:

$$f = 8Re_D^{-0.3} \quad (14)$$

where equation (14) is valid for Reynolds numbers between 1 and 1000. The friction factor of this equation has been devised empirically in order to get the best fit of the results. It has been found to be in good agreement with the expressions found by other authors for similar geometries. This aspect will be considered below with more detail.

4. APPLICATION OF CORRELATIONS TO HEAT EXCHANGER CALCULATIONS

In the present study, the fluid flow and heat transfer in a compact plate-type heat exchanger are studied, using a prediction model that assumes a number of pairs of parallel plates. Each plate has at one side a smooth surface and at the other side an offset strip-fin surface. Water or engine coolant circulates through the passages formed by the smooth parallel plates, while engine oil circulates through the channels formed by the rough surfaces. The flow of the two fluids inside the heat exchanger develops in two parts. In the first part the flow is parallel, while in the second part the flow is in counter-flow. The overall heat transfer coefficient U is computed from the following expression:

$$U = 1/(1/h_1 + \Delta x/k_w + 1/h_2) \quad (15)$$

where h_1 refers to the heat transfer coefficient for the fluid flow between the parallel plates, computed from the Nusselt number given by expressions (2), (4) or (7)

$$h_1 = Nu\, k_1 D_1^{-1} \quad (16)$$

and h_2 refers to the heat transfer coefficient for the fluid flow between the offset strip-fin plates, computed from the Nusselt number given by expression (11)

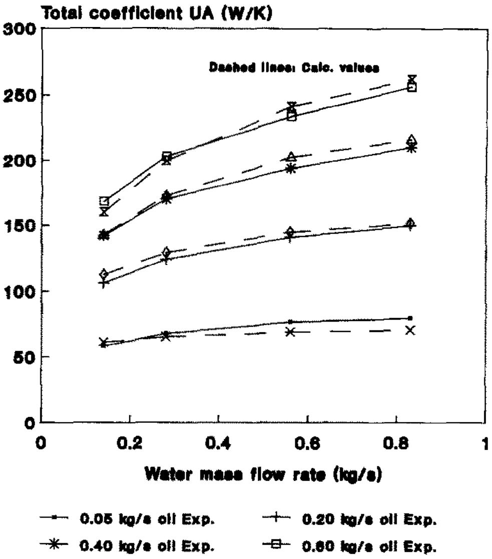

FIG. 5. The product UA vs the water flow rate.

$$h_2 = Nu\, k_2 D_2^{-1}. \quad (17)$$

Finally, Δx and k_w are, respectively, the thickness and thermal conductivity of the metallic plates.

The values of the total heat transfer coefficient UA for a heat exchanger with eight fluid passages have been plotted in Fig. 5 as a function of the mass flow rates of both fluids (water and engine oil). The graph shows the performance of the water/engine oil plate heat exchanger with entering temperatures of 85°C (water) and 120°C (oil) for different oil mass flow rates ranging from 0.05 to 0.6 kg s^{-1}. These values appear compared with the corresponding values of UA obtained from the experimental data (exit fluid temperatures) obtained in a test rig. As shown, the calculated values are in very good agreement with the experimental data over the full range of flow rates.

Figure 6 shows the values of the pressure drop

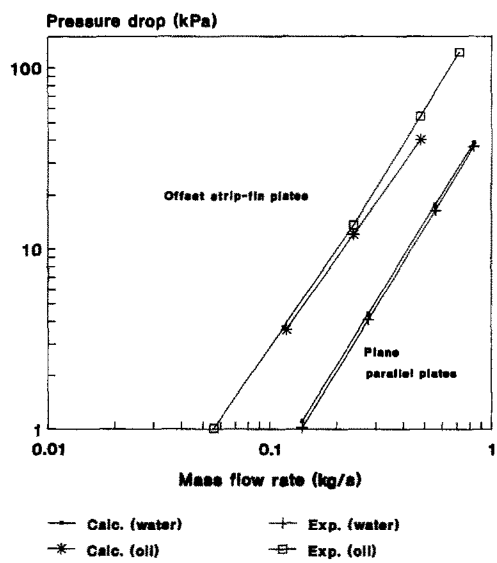

FIG. 6. Pressure drop in water and oil passages.

(expression (12)) in both the water passages (formed by plane parallel plates) and the oil passages (formed by offset strip-fin plates), compared with the experimental data. The conditions of the two fluids are the same as in Fig. 5. The pressure drop is plotted as a function of the mass flow rate on a log–log scale. Since the data follow an almost straight line, this implies that the overall pressure drop can be correlated by a power function relationship of the general form:

$$\Delta p = C\dot{m}^{\alpha} \qquad (18)$$

where C is a constant for a particular plate configuration, $\dot{m}$ the mass flow rate and α an exponent of order 1.3–1.8 that depends on the number of flow passages. These values of exponent α agree well with those that can be obtained from expressions (12) and (14)

$$\Delta p = \tfrac{1}{2} f \rho c^2 \simeq Re_D^{-0.3}\,\rho c^2 \simeq (\rho c)^{-0.3}(\rho c)^2 \rho^{-1} \simeq \dot{m}^{1.7}. \qquad (19)$$

5. DISCUSSION AND ACCURACY OF THE CORRELATIONS

The ability of the model to predict the heat exchanger performance by using the correlations has been evaluated for a number of operating conditions. The values of the product UA for the whole heat exchanger have been compared with experimental data as shown in Fig. 5. The average percentage error for all values of oil flow rates is between 3% and 1.9% when the water flow rate varies from 0.14 to 0.83 kg s^{-1}.

In order to further explore the validity of the correlations, especially the ones corresponding to the offset strip-fin plates, several comparisons with other results found in the literature have been performed. Figure 7 shows a composite plot of heat transfer (j) and friction factor (f) coefficients as functions of Reynolds number for typical operating conditions. Instead of the Nusselt number, the corresponding j (Colburn factor) has been chosen to establish the comparison. The graph indicates the correlations proposed in this paper for both water passages, formed by plane parallel plates with a length-to-diameter ratio of 20, and the oil passages formed by offset strip-fin plates as defined above in Section 2.2. The Reynolds number range for oil is from 20 to 1000, while for water it is from 1000 to 20 000.

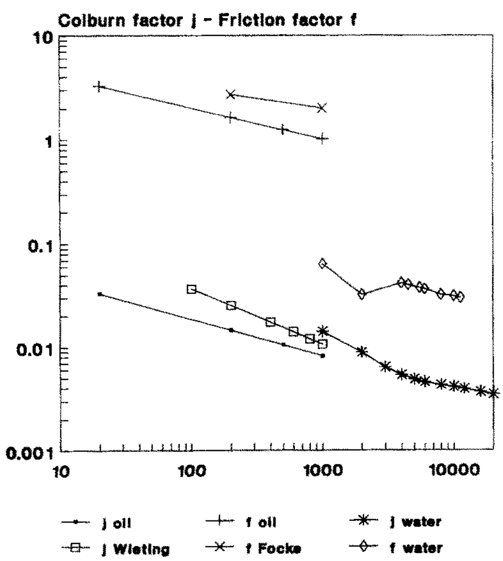

FIG. 7. Colburn j and friction factor f vs Re_D.

Figure 7 also contains the Colburn factor correlation found by Wieting [9] for offset strip-fin plates similar to those of the present study. The smooth line through the range of Reynolds numbers between 100 and 1000 represents the linear regression of the experimental data for 22 geometric configurations found by Wieting. The trends of the present correlations for offset strip-fin plates and Wieting's correlation are the same, although the authors' correlation provides values about 35% less between $Re_D = 200$ and $Re_D = 1000$. It should be noted that the working fluid used by Wieting was air and that of the present study is engine oil, and the discrepancy could be due to different values of Prandtl number for the air and the oil. In addition, Wieting has noted that dispersions of up to 40% can be found when comparing data from different sources.

The evaluation of expression (14) has been achieved by comparing it with the results of other workers for similar plate geometries [10–12] and has been found to be quite similar. For instance, the friction factor found by Focke [10] for asymmetrically corrugated plates is $f = 7.4Re_D^{-0.19}$ for a Reynolds number range of 300–1000. The difference between the latter correlation and the proposed correlation is again of the order of 40% in the range of interest. This difference can be explained, since the plate geometry, although similar, is not the same, especially if the border effects are considered.

6. CONCLUSIONS

The expressions provided for the heat transfer coefficient allow the computation of the Nusselt number for short ducts and for all values of Reynolds number, including the transitional regime. These expressions are well formed in the laminar and fully turbulent regimes, since they can be considered as the standard expressions modified by correction factors. The values obtained from these expressions are in agreement with the literature data. In the transitional regime, the solution adopted can be regarded as a convenient expression to obtain a closed-form equation. The correlations for the friction factor have been found to be good by comparing with other references. The application of heat transfer and friction factor expressions to a plate-type heat exchanger (with both parallel plates and offset strip-fin plates) gives very good agreement with experimental data.

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FORMULES POUR LES CARACTERISTIQUES DE TRANSFERT THERMIQUE ET DE FROTTEMENT FLUIDE DANS LES ECHANGEURS THERMIQUES COMPACTS PLATULAIRES

Résumé—Des formules sont proposées pour les coefficients de transfert thermique et de perte de charge entre plaques parallèles et entre plaques munies d'ailettes offset pour les domaines correspondant aux échangeurs de chaleur compacts. Des expressions analytiques sont utilisées pour présenter ces formules. Celles-ci permettent de prédire correctement les données expérimentales disponibles pour la chaleur transférée et la perte de pression dans les échangeurs de chaleur compacts platulaires. Les formules couvrent le domaine complet de l'écoulement depuis le laminaire jusqu'au turbulent, à la fois pour les tubes courts et longs. On donne aussi des suggestions pour les étendre à d'autres conditions d'écoulement.

KORRELATIONEN FÜR WÄRMEÜBERGANG UND DRUCKABFALL IN KOMPAKTEN PLATTENWÄRMEAUSTAUSCHERN

Zusammenfassung—Es werden Korrelationsgleichungen für Wärmeübergangskoeffizienten und Widerstandsbeiwerte in Kompaktwärmeaustauschern aus ebenen parallelen Platten und Platten mit Streifenrippen vorgeschlagen. Diese Korrelationen werden als geschlossene Ausdrücke angegeben. Sie erlauben eine angemessene Wiedergabe von verfügbaren Versuchsdaten für die ausgetauschte Wärme und den Druckabfall in kompakten Plattenwärmeaustauschern. Die Korrelationen decken kontinuierlich den gesamten Bereich laminarer und turbulenter Strömung ab, und zwar für kurze wie auch für lange Rohre. Es werden Vorschläge zur Erweiterung der Korrelationen auf andere Strömungsbedingungen unterbreitet.

ОБОБЩАЮЩИЕ СООТНОШЕНИЯ ДЛЯ ХАРАКТЕРИСТИК ТЕПЛОПЕРЕНОСА И ГИДРОДИНАМИЧЕСКОГО ТРЕНИЯ КОМПАКТНЫХ ПЛАСТИНЧАТЫХ ТЕПЛООБМЕННИКОВ

Аннотация—В замкнутом виде представлены обобщающие соотношения для коэффициентов теплопереноса и гидродинамического трения в случаях плоскопараллельных пластин и оребренных пластин с перекрытием. Исследуемые параметры изменяются в диапазонах, используемых в компактных теплообменниках. Предложенные соотношения позволяют с достаточной точностью описать имеющиеся экспериментальные данные по теплопереносу и потерям давления в компактных пластинчатых теплообменниках. Эти соотношения охватывают весь диапазон течений от ламинарного до турбулентного в случаях как с короткими, так и с длинными трубами. Даны предложения по распространению указанных соотношений на другие условия течения.