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Performance characteristics of industrial finned tubes presented in dimensional form

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Abstract—A method for presenting the heat transfer and pressure drop characteristics of industrial finned tubes is proposed. Dimensional heat transfer and pressure drop parameters are defined. This form of data presentation eliminates uncertainties inherent in the evaluation of the fin-side heat transfer coefficient, fin efficiency, thermal contact resistance, etc. Existing performance correlations are transformed to the proposed form of presentation and are compared to new experimental data obtained from wind-tunnel tests.

1. INTRODUCTION

THE PERFORMANCE characteristics of many different types of industrial finned tubes have been reported in the literature. Extensive summaries of some of these results have been prepared [1]. Although numerous empirical correlations based on these results have been proposed, discrepancies do occur, even when applied only to a limited range of operating conditions and to specified finned tube geometries. In part, these discrepancies may be due to unreliable experimental data or the incorrect interpretation of test results. Recent, carefully conducted experiments by Eckels and Rabas [2] on 'T' foot and overlapped 'L' foot finned tubes tend to confirm this statement.

In this investigation the performance characteristics of a four-row bundle of closely packed, extruded, helically finned, bi-metallic tubes are determined experimentally and the results are presented in terms of the conventional Colburn and friction factors and in the form of dimensional heat transfer and pressure drop parameters.

2. APPARATUS

The performance characteristics of different types of finned tubes employed in industrial heat exchangers are determined experimentally in wind-tunnels designed specifically for this purpose. One example of such a wind-tunnel is shown schematically in Fig. 1 [3].

The heat exchanger bundle (1) is preceded by a rounded inlet section that ensures an essentially turbulence-free, uniform stream of ambient air enters the bundle. The air is then drawn through an insulated connecting section (2) and two rows of mixing louvres (4) by means of a centrifugal fan (8) that has adjustable inlet vanes for flow control. After passing through a perforated plate (6) the air flows through one or more elliptic flow nozzles mounted in a suitable plate (7). By measuring the pressure differential across the plate, the air flow rate can be determined. Both the wet- and dry-bulb temperatures of the inlet air stream are measured before the bundle while the outlet air temperature is measured by a sampling probe (5) located

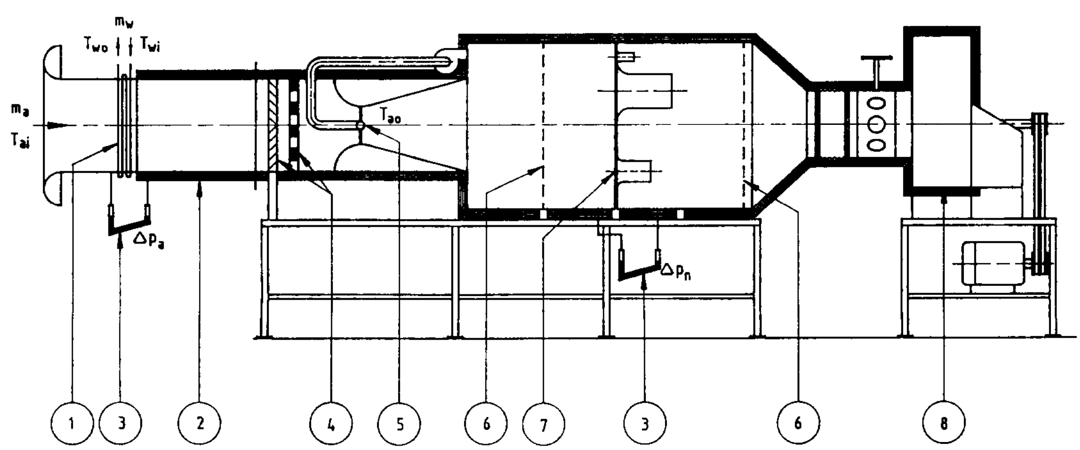

FIG. 1. Test wind-tunnel.

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NOMENCLATURE

A	area [m^2]
a	coefficient
b	exponent
C_r	row correction factor
c_p	specific heat [J kg^{-1} °C^{-1}]
d	diameter [m]
de	equivalent diameter [m]
Eu	Euler number
Ey	characteristic pressure drop parameter [m^{-2}]
F_t	temperature correction factor
G_c	mass velocity through minimum flow area [kg m^{-2} s^{-1}]
h	heat transfer coefficient [W m^{-2} °C^{-1}]
k	thermal conductivity [W m^{-1} °C^{-1}]
L	length [m]
m	mass flow rate [kg s^{-1}]
Nu	Nusselt number
Ny	characteristic heat transfer parameter [m^{-1}]
n	number
P	pitch [m]
Δp	pressure differential [N m^{-2}]
Pr	Prandtl number
Q	rate of heat transfer [W]
R	thermal resistance [m^2 °C W^{-1}]
Re	Reynolds number
Ry	characteristic flow parameter [m^{-1}]
St	Stanton number
T	temperature [°C]
t	thickness [m]
ΔT_{lm}	logarithmic mean temperature difference [°C]
U	overall heat transfer coefficient [W m^{-2} °C^{-1}]
V	volume [m^3].

Greek symbols

ε	effectiveness or surface roughness
η	efficiency
σ	contraction ratio.

Subscripts

a	air
c	contact or minimum through flow area
f	fin
fr	frontal or free
i	inlet or inside
l	longitudinal
o	outlet or outside
r	root or rows of tubes
t	transverse or tube
w	water.

after the mixing louvres. The static pressure drop across the bundle is measured with an inclined manometer or pressure transducer.

Warm water is pumped through the tubes at prescribed rates. Due to relatively small changes in water temperature great care is taken in measuring these differences.

Details of the finned tube tested are shown in Fig. 2. The effective finned length of each tube was 500 mm. The fins are tapered having a tip thickness of 0.22 mm and a mean thickness of 0.4 mm. The mean surface roughness ε is less than 1 μm. The thermal contact resistance between the core tube outside surface and the aluminium muff was determined experimentally on a number of randomly selected tubes [4]. Based on the contact surface area a mean value of 3.5×10^{-5} m^2 °C W^{-1} was obtained for this resistance.

Water at approximately 55°C was introduced into

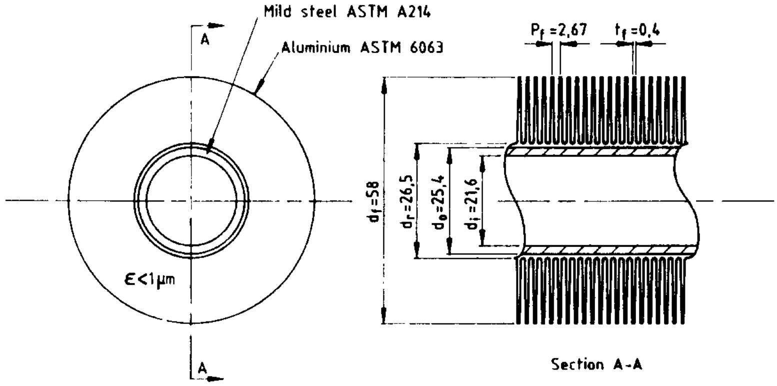

FIG. 2. Extruded, bi-metallic, finned tube.

the tubes through a manifold located above the last row of tubes. To ensure that the flow through each tube path was uniform, flow throttling plugs were located at the water inlet end. The circuiting was such that essentially counterflow conditions were maintained. Errors due to bundle end effects were minimized by having thermally active half-tubes located at the ends of the tube rows. This was accomplished by casting the particular tube into a trough containing a predetermined quantity of plaster which was subsequently allowed to harden as shown in Fig. 3.

During the actual tests, the degree of turbulence in the inlet air stream was measured and found to be consistently less than 1%. Since free-stream turbulence tends to influence performance, particularly when the number of tube rows is reduced, control thereof is desirable. The mean inlet air temperature during testing was 25°C.

3. ANALYSIS

In the absence of heat losses from the bundle to the environment the reduction of enthalpy of the water stream is equal to the increase of enthalpy of the air stream or

$$Q = m_w c_{pw}(T_{wi} - T_{wo}) = m_a c_{pa}(T_{ao} - T_{ai}). \quad (1)$$

It is however also possible to express the heat transfer rate in terms of the overall heat transfer coefficient and the logarithmic mean temperature difference:

$$Q = U_a A_a F_t \Delta T_{lm} \quad (2)$$

where F_t is the temperature correction factor in the case of cross flow [12].

The overall heat transfer coefficient may be expressed as

$$U_a = \left(\frac{1}{h_a \varepsilon_f} + \frac{A_a}{h_w A_w} + \sum_n \frac{A_a R_n}{A_n}\right)^{-1} \quad (3)$$

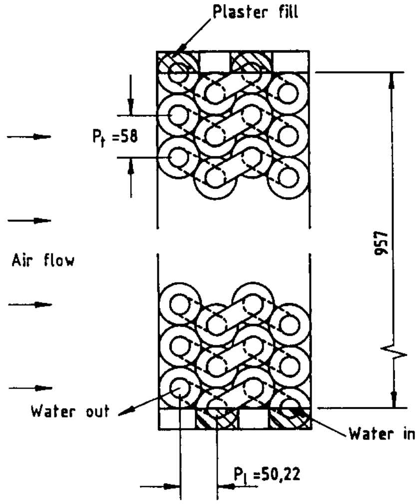

FIG. 3. Details of tube bundle layout.

where the summation term represents all thermal resistances other than the air- and water-side values including those due to the tube wall, fouling and the thermal contact between the tube and the fin when they are not integral.

Part of the heat is transferred from the prime surface between the fins directly to the air stream, while most of the heat is transferred through the fins having a certain fin efficiency. The effectiveness of such a finned surface may be expressed in terms of the fin efficiency:

$$\varepsilon_f = 1 - \frac{A_f}{A_a}(1 - \eta_f) \quad (4)$$

where A_f is the fin surface area and A_a is the total air-side area.

Substitute equation (3) in equation (2), re-arrange and find

$$h_a = \left[\varepsilon_f A_a\left(\frac{F_t \Delta T_{lm}}{Q} - \frac{1}{h_w A_w} - \sum_n \frac{R_n}{A_n}\right)\right]^{-1}. \quad (5)$$

If all parameters in the denominator of equation (5) are either specified or available from experimental data, except ε_f which is also a function of h_a, then the air-side heat transfer coefficient under particular test conditions can be found by an iterative procedure.

According to Colburn [5] this coefficient can be presented in dimensionless form as follows:

$$j_a = St_a Pr_a^{0.67} = \frac{Nu_a}{Re_a Pr_a^{0.33}} = \frac{h_a Pr_a^{0.67}}{G_{ca} c_{pa}} = f(Re_a) \quad (6)$$

where j_a is known as the Colburn j-factor. The mass velocity G_{ca} is evaluated on the basis of the minimum free flow area A_{ca}, i.e. $G_{ca} = m_a/A_{ca}$. The Reynolds and Nusselt numbers are based on the equivalent or hydraulic diameter defined as $de = 4A_{ca}L/A_a$ where L is the flow length of the heat exchanger and A_{ca} is the minimum flow cross-sectional area. For flow normal to tube banks, L is an equivalent flow length measured from the leading edge of the first tube row to the leading edge of a tube row that would follow the last tube row, were another tube row present [6].

The Reynolds number is defined as

$$Re_a = 4G_{ca}A_{ca}L/\mu_a A_a = 4m_a L/\mu_a A_a. \quad (7)$$

With h_a obtained from equation (5) and the corresponding value of G_{ca} known, the j-factor and the Reynolds number can be determined.

The air-side pressure drop through the finned tubes can be similarly expressed in terms of a dimensionless friction factor during isothermal operation

$$f_{a,iso} = \frac{2\rho_a \Delta p_{a,iso}}{G_{ca}^2}\frac{A_c}{A_a}. \quad (8)$$

An Euler number may also be defined as

$$Eu_{a,iso} = \frac{\rho_a \Delta p_{a,iso}}{G_{ca}^2} = \frac{f_{a,iso} A_a}{2A_c} = f(Re_a). \quad (9)$$

Numerous empirical correlations based on extensive experimental data are available for the fin-side Nusselt and Euler numbers [7–10]. If applied correctly within specified limits these correlations are of considerable value to designers of heat exchangers. It should however be stressed that to date no correlation exists that accurately predicts the performance over a wide spectrum of finned tube geometries and operating conditions. Due to factors such as fin surface roughness, free-stream turbulence, poor flow distribution, bypassing of the air stream, different testing temperatures and uncertainties concerning thermal contact resistance, poorly defined or unreliable data exist in the literature.

In view of the above considerations, the final design of very costly, large air-cooled systems that have to operate satisfactorily under specified conditions, cannot be based on approximate correlations. In such cases specific performance tests should be conducted on the tubes to be used. The range of test air flow rates should cover the proposed actual operating velocities while fluid temperatures should also be close to those found in the planned cooling system. Instead of presenting the test data in the form of j- and f-factors a more meaningful form, based on the method presented by Kern [11] is proposed.

From equation (5) it follows that

$$\frac{1}{h_a \varepsilon_f A_a} + \sum_n \frac{R_n}{A_n} = \frac{F_t \Delta T_{lm}}{Q} - \frac{1}{h_w A_w}$$

or

$$h_{ae} A_a = \left(\frac{1}{h_a \varepsilon_f A_a} + \sum_n \frac{R_n}{A_n}\right)^{-1} = \left(\frac{F_t \Delta T_{lm}}{Q} - \frac{1}{h_w A_w}\right)^{-1} \tag{10}$$

where h_{ae} is an effective heat transfer coefficient based on the air-side surface area.

According to equation (6) the heat transfer coefficient h_a expressed in dimensionless form, may be expressed in terms of the Reynolds number as

$$Nu_a / Pr_a^{0.33} = a_1 Re_a^{b_1}. \tag{11}$$

Because the effectiveness ε_f is related to the fin efficiency η_f as shown in equation (4), it is also a function of the flow Reynolds number. The thermal resistances appearing in the summation term in the denominator of equation (10) are essentially independent of the Reynolds number and may be assumed to be constant for all practical purposes.

Both the Nusselt and the Reynolds numbers in equation (11) contain an equivalent or hydraulic diameter. Because of the relatively arbitrary nature of the definition of this quantity for finned surfaces, different definitions are found in the literature. In practice this often leads to confusion and makes any comparison of different types of finned surfaces meaningless.

In the absence of an equivalent diameter and taking into consideration the above mentioned interdependence of variables, the following relation holds:

$$\frac{h_{ae}}{k_a Pr_a^{0.33}} = a_2 Ry^{b_2} \tag{12}$$

where $Ry = m_a / A_{fr} \mu_a$ is known as a characteristic flow parameter.

The effective finned surface area and the heat exchanger frontal area play a major role in comparing and optimizing heat exchangers. These geometric parameters may be introduced into equation (12) such that

$$Ny = \frac{h_{ae} A_a}{A_{fr} k_a Pr_a^{0.33}} = a_{Ny} Ry^{b_{Ny}} \tag{13}$$

where Ny is known as a characteristic heat transfer parameter.

The value of $h_{ae} A_a$ can be determined directly from equation (10) in terms of measured and specified values. Uncertainties concerning fouling and in particular the magnitude of the thermal contact resistance between the tube and the fin are eliminated since these values are included in the effective heat transfer coefficient h_{ae}. In practice the actual values of h_a and ε_f are of no particular interest. From equation (10) it follows that in the absence of fouling, but taking into consideration contact resistance

$$\begin{aligned} h_{ae} A_a = [&1/h_a \varepsilon_f A_a + \ln(d_o/d_i)/2\pi k_t L_t \\ &+ \ln(d_r/d_o)/2\pi k_f L_t + R_c/A_o]^{-1} \end{aligned} \tag{14}$$

hence, from equation (13)

$$\begin{aligned} Ny = {} & (h_a / A_{fr} k_a Pr_a^{0.33}) \{1/\varepsilon_f A_a + h_a \\ & \times [\ln(d_o/d_i)/2\pi k_t L_t + \ln(d_r/d_o)/2\pi k_f L_t + R_c/A_o]\}^{-1} \end{aligned} \tag{15}$$

where L_t is the total tube length.

The characteristic heat transfer parameter is related to the fin-side Nusselt number. According to Mirkovic [7] the latter may be expressed as:

$$\begin{aligned} Nu_a = \frac{h_a de_t}{k_a} = {} & 0.224 Pr_a^{0.33} Re_a^{0.662} C_r \left(\frac{P_t - d_r}{d_r}\right)^{0.1} \\ & \times \left(\frac{d_r}{P_l - d_r}\right)^{0.15} \left[\frac{2(P_f - t_f)}{d_f - d_r}\right]^{0.25} \end{aligned} \tag{16}$$

where de_t is the equivalent thermal diameter and C_r is a correction factor depending on the number of tube rows.

Similarly a general heat transfer correlation is given by Briggs and Young [8], for bundles having approximately six tube rows

$$\begin{aligned} Nu_a = \frac{h_a de_t}{k_a} = {} & 0.134 Pr_a^{0.33} Re_a^{0.681} \\ & \times \left[\frac{2(P_f - t_f)}{d_f - d_r}\right]^{0.2} \left(\frac{P_f - t_f}{t_f}\right)^{0.1134}. \end{aligned} \tag{17}$$

Another correlation is that of Schulenberg [9]

$$Nu_a = 0.586 Pr_a^{0.33} (0.0547 Re_a)^{0.7-0.0118(A_a/A_r)}. \tag{18}$$

Although many other similar correlations exist, all the relations may in general be written in the form

$$\frac{h_a}{k_a Pr_a^{0.33}} = \frac{a_{Nu}}{de_t} Re_a^{b_{Nu}} \tag{19}$$

where

$$Re_a = \frac{G_{ca} de_t}{\mu_a} \quad \text{and} \quad G_{ca} = \frac{m_a}{A_{ca}}$$

or

$$\frac{h_a}{k_a Pr_a^{0.33}} = \frac{a_{Nu}}{de_t^{1-b_{Nu}}} \left(\frac{G_{ca}}{\mu_a}\right)^{b_{Nu}} \tag{20}$$

where a_{Nu} depends on the geometry of the heat exchanger.

It now follows from equation (15) that

$$Ny = \frac{a_{Nu}}{A_{fr} de_t^{1-b_{Nu}}} \left(\frac{A_{fr}}{A_c}\right)^{b_{Nu}} \left(\frac{m_a}{A_{fr}\mu_a}\right)^{b_{Nu}}$$
$$\left\{\frac{1}{\varepsilon_f A_a} + h_a \left[\frac{\ln(d_o/d_i)}{2\pi k_t L_t} + \frac{\ln(d_r/d_o)}{2\pi k_f L_t} + \frac{R_c}{A_o}\right]\right\}^{-1}$$
$$= a_{Ny} R_y^{b_{Ny}} \tag{21}$$

with

$$b_{Ny} = b_{Nu}$$

and

$$a_{Ny} = \frac{a_{Nu}}{A_{fr} de_t^{1-b_{Nu}} \sigma^{b_{Nu}}}$$
$$\times \left\{\frac{1}{\varepsilon_f A_a} + h_a \left[\frac{\ln(d_o/d_i)}{2\pi k_t L_t} + \frac{\ln(d_r/d_o)}{2\pi k_f L_t} + \frac{R_c}{A_o}\right]\right\}^{-1}$$

where $\sigma = A_c/A_{fr}$.

The values of a_{Nu}, b_{Nu} and de_t are as follows for the different correlations.

Mirkovic [7]:

$$a_{Nu} = 0.224 C_r \left(\frac{P_t - d_r}{d_r}\right)^{0.1}$$
$$\times \left(\frac{d_r}{P_l - d_r}\right)^{0.15} \left[\frac{2(P_f - t_f)}{d_f - d_r}\right]^{0.25}$$

$$b_{Nu} = 0.662, \quad de_t = \frac{A_{al}}{\pi(d_f - d_r + P_f - t_f)}$$

where the subscript 1 refers to the surface area of a single fin plus the root area between two fins.

Briggs and Young [8]:

$$a_{Nu} = 0.134 \left[\frac{2(P_f - t_f)}{d_f - d_r}\right]^{0.2} \left(\frac{P_f - t_f}{t_f}\right)^{0.1134}$$

$$b_{Nu} = 0.681, \quad de_t = d_r.$$

Schulenberg [9]:

$$a_{Nu} = 0.586 (0.0547)^{0.7-0.0118(A_a/A_r)}$$

$$b_{Nu} = 0.7\text{–}0.0118(A_a/A_r), \quad de_t = \frac{d_r}{(A_a/A_r)}.$$

The Euler number as given by equation (9) may also be expressed in terms of the characteristic flow parameter

$$Eu_{a,iso} = \frac{\rho_a \sigma^2 \Delta p_{a,iso}}{\mu^2 Ry^2} = f(Ry) \tag{22}$$

or a characteristic pressure drop parameter may be defined as

$$Ey_{iso} = \frac{Eu_{a,iso} Ry^2}{\sigma^2} = \frac{\rho_a \Delta p_{a,iso}}{\mu^2} = a_{Ey} R_y^{b_{Ey}}. \tag{23}$$

However, during non-isothermal operation there is, in addition to the frictional loss, a further term due to acceleration effects [6]

$$\Delta p_{acc} = \frac{G_{ca}^2}{2}(1+\sigma^2)\left(\frac{1}{\rho_{ao}} - \frac{1}{\rho_{ai}}\right) \tag{24}$$

or in terms of the characteristic flow parameter

$$\frac{\rho_a \Delta p_{acc}}{\mu^2} = Ry^2 \left(\frac{1}{\sigma^2} + 1\right)\left(\frac{\rho_{ai} - \rho_{ao}}{\rho_{ai} + \rho_{ao}}\right) \tag{25}$$

where

$$\rho_a = \frac{2}{(1/\rho_{ai} + 1/\rho_{ao})}.$$

The non-isothermal characteristic pressure drop parameter is thus defined as

$$Ey = a_{Ey} Ry^{b_{Ey}} + Ry^2 \left(\frac{1}{\sigma^2} + 1\right)\left(\frac{\rho_{ai} - \rho_{ao}}{\rho_{ai} + \rho_{ao}}\right). \tag{26}$$

According to Mirkovic [7] the dimensionless isothermal pressure drop across a finned tube heat exchanger can be expressed as

$$Eu_{a,iso} = \frac{\Delta p_a \rho_a}{G_{ca}^2} = \frac{3.96 n_r}{Re_{ah}^{0.31}} \left(\frac{P_t - d_r}{d_r}\right)^{0.14}$$
$$\times \left(\frac{d_r}{P_l - d_r}\right)^{0.18} \left[\frac{d_f - d_r}{2(P_f - t_f)}\right]^{0.2}$$
$$= a_{Eu} Re_{ah}^{-0.31}$$

where

$$a_{Eu} = 3.96 n_r \left(\frac{P_t - d_r}{d_r}\right)^{0.14} \left(\frac{d_r}{P_l - d_r}\right)^{0.18}$$
$$\times \left[\frac{d_f - d_r}{2(P_f - t_f)}\right]^{0.2}$$

$$de_h = \frac{4 V_{fr}}{A}, \quad Re_{ah} = \frac{G_{ca} de_h}{\mu_a} = \frac{Ry\, de_h}{\sigma}.$$

From equation (23) with the above relations find

$$Ey_{iso} = a_{Ey} Ry^{b_{Ey}}$$

where

$$a_{Ey} = a_{Eu}/de_{h}^{0.31}\sigma^{1.69} \quad \text{and} \quad b_{Ey} = 1.69.$$

Robinson and Briggs [10] propose the following correlation for the Euler number:

$$Eu_{a,iso} = \frac{\Delta p_a \rho_a}{G_{ca}^2} = \frac{18.93 n_r}{Re_a^{0.316}}\left(\frac{d_r}{P_t}\right)^{0.927}\left(\frac{P_t}{P_l}\right)^{0.515}$$

$$= a_{Eu} Re_a^{-0.316}$$

where

$$a_{Eu} = 18.93 n_r \left(\frac{d_r}{P_t}\right)^{0.927}\left(\frac{P_t}{P_l}\right)^{0.515}$$

$$Re_a = \frac{G_{ca} d_r}{\mu_a} = \frac{Ry\, d_r}{\sigma}.$$

It is readily shown that

$$Ey_{iso} = a_{Ey} Ry^{b_{Ey}}$$

where

$$a_{Ey} = a_{Eu}/d_r^{0.316}\sigma^{1.684} \quad \text{and} \quad b_{Ey} = 1.684.$$

Although the thermophysical properties of the fluids are temperature dependent, no correction is required if the range of operating temperatures is limited [6].

4. RESULTS

The test results for the four-row heat exchanger bundle are shown in Fig. 4. In this particular presentation the Reynolds number is defined as $Re_a = G_{ca} d_r/\mu_a$ and $f_{a,iso} = \Delta p_{a,iso}\rho_a/2n_r G_{ca}^2$ as proposed by Eckels and Rabas [2]. These definitions differ from those given respectively by equations (7) and (8).

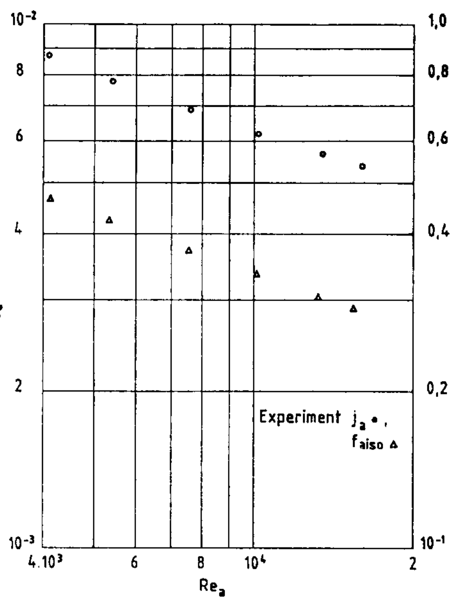

FIG. 4. Heat transfer and pressure drop in a four-row heat exchanger bundle.

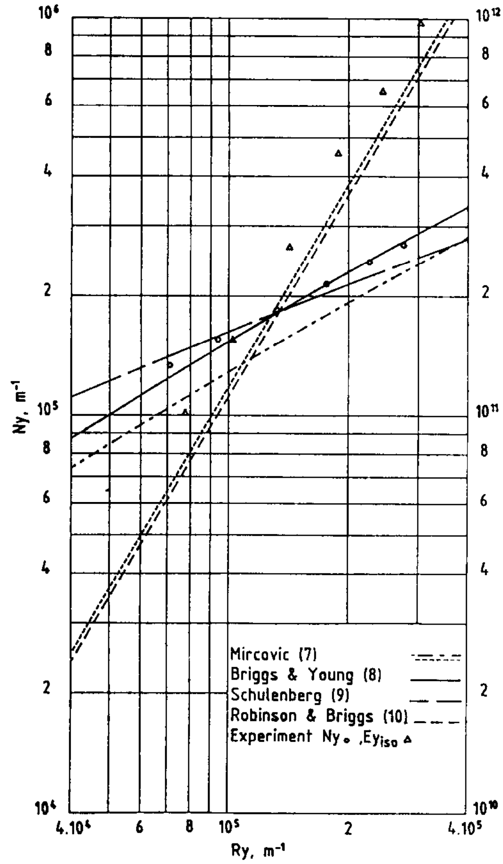

FIG. 5. Heat transfer and pressure drop in a four-row heat exchanger bundle.

The test results in the form of characteristic heat transfer and pressure drop parameters are also shown in Fig. 5 and are compared to various empirical correlations. The heat transfer correlation due to Briggs and Young [8] is in good agreement with the experimental data. Existing correlations however tend to underpredict the pressure drop in the closely packed four-row bundle.

5. CONCLUSION

A method for presenting performance characteristics of finned tube bundles in the form of dimensional heat transfer and pressure drop parameters is proposed for application in practice. The method eliminates potential sources of error in well conducted experiments. Existing empirical correlations are transformed to the new parameters and are compared to experimental results obtained on a closely packed four-row finned tube bundle.

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CARACTERISTIQUES DE PERFORMANCE DES TUBES AILETES INDUSTRIELS PRESENTES SOUS FORME DIMENSIONNELLE

Résumé—On propose une méthode de présentation des caractéristiques de transfert thermique et de perte de charge des tubes ailetés industriels. Des paramètres dimensionnels sont définis. Cette forme de présentation des données élimine les incertitudes inhérentes à l'évaluation du coefficient de transfert de chaleur du côté des ailettes, de l'efficacité d'ailette, de la résistance thermique de contact, etc. Des formules existantes sont transformées dans la forme proposée et sont comparées à des nouvelles données expérimentales obtenues à partir d'essais en soufflerie.

DIE LEISTUNGSCHARAKTERISTIK VON INDUSTRIELLEN RIPPENROHREN, DARGESTELLT IN DIMENSIONSBEHAFTETER FORM

Zusammenfassung—Eine Methode zur Darstellung des Wärmeübertragungs- und Druckabfallverhaltens von industriellen Rippenrohren wird vorgeschlagen. Dimensionsbehaftete Parameter für Wärmeübergang und Druckabfall wurden definiert. Diese Darstellungsform der Daten eliminiert die Unsicherheiten bei der Bestimmung des rippenseitigen Wärmeübergangskoeffizienten, des Rippenleistungsgrades, des thermischen Kontaktwiderstandes usw. Vorhandene Leistungskorrelationen werden in die vorgeschlagene Darstellungsform transformiert und mit neuen experimentellen Daten, die aus Windkanalversuchen ermittelt wurden, verglichen.

ПРЕДСТАВЛЕНИЕ РАБОЧИХ ХАРАКТЕРИСТИК ПРОМЫШЛЕННЫХ ОРЕБРЕННЫХ ТРУБ В РАЗМЕРНОМ ВИДЕ

Аннотация—Предложен метод представления характеристик теплообмена и падения давления в промышленных оребренных трубах. Определены размерные параметры теплообмена и падения давления. Эта форма представления информации устраняет неопределенности, свойственные оценке коэффициента теплообмена ребра, теплового сопротивления контакта и др. Известные соотношения для коэффициентов преобразованы в предложенную форму представления и сравниваются с новыми экспериментальными результатами, полученными в аэродинамической трубе.