A comparative study of elliptical and circular sections in one- and two-row tubes and plate fin heat exchangers

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In this work, a two-dimensional (2-D) heat transfer analysis is performed in one- and two-row tubes and plate fin heat exchangers (circular and elliptical sections), using experimentally determined heat transfer coefficients from a heat and mass transfer analogy. The temperature distribution on the fin and air free stream, and the fin efficiency are determined for heat exchangers, with eccentricity 0.5 and 0.65, as a function of the Reynolds number. For tubes and plate fin heat exchangers, new numerical results of fin efficiency for elliptical tubes are compared with published results for circular tubes. A relative fin efficiency gain of up to 18% is observed in the elliptical arrangement, as compared to the circular one. The efficiency gain, combined with the relative pressure drop reduction of up to 25% observed in previous studies (Brauer 1964; Bordalo and Saboya 1995) show the elliptical arrangement has the potential for a considerably better overall performance than the conventional circular arrangement.

Keywords: fin efficiency gain; pressure drop reduction; heat transfer correlations

Introduction

The subject investigated in this paper was inspired by the increasing need for optimization in all engineering applications, aiming to rationalize the use of available energy and reduction of lost work. Tubes and plate fin heat exchangers are widely employed in such commercial applications as air conditioning systems, heaters, and radiators.

The elliptic tube geometry has a better aerodynamic configuration than the circular one; therefore, it is reasonable to expect a reduction in total drag force and an increase in heat transfer when comparing the former to the latter, when they are submitted to a cross-flow free stream. Guided by this observation, Brauer (1964) reported a survey of experimental results comparing elliptic and circular arrangements for the heat transfer and pressure drop points of view. Later, Schulemberg (1966) analyzed the potential of the application of elliptic tubes in industrial heat exchangers, showing experimental heat transfer and pressure drop results. Recently, Bordalo and Saboya (1995) reported pressure drop measurements comparing the two configurations, with one-, two-, and three-row arrangements. The conclusion of those studies based on experimental evidence is that the elliptic tube configuration performs better than the circular one.

In a pioneering study, Shepherd (1956) analyzed one-row circular tubes and plate fin heat exchangers determining global heat transfer coefficients as a function of the Reynolds number, assuming isothermal fins. Saboya (1974), using the naphtalene sublimation technique and the heat and mass transfer analogy, experimentally obtained local and global heat and mass transfer coefficients, for one- and two-row circular tube and plate fin heat exchangers. Saboya and Sparrow (1976) extended the study for three-row heat exchangers. The results show low mass transfer coefficients behind the tubes, as compared with the fin average. Ximenes (1981) reported experimental results for mass transfer coefficients in one- and two-row elliptical tube and plate fin heat exchangers. In the elliptic configuration, it was observed that the mass transfer coefficients drop less dramatically behind the tubes than in the circular configuration. Rosman et al. (1984) experimentally determined local and global heat transfer coefficients, using the heat and mass transfer analogy, for one- and two-row circular tubes and plate fin heat exchangers, followed by a numerical computation of the fin temperature distribution and efficiency, and free stream bulk temperature along the fin. The

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results show that the two-row configuration is more efficient that the one-row configuration.

Stanescu et al. (1996) studies the optimal spacing of circular cylinders in free-stream cross-flow forced convection, and Bejan et al. (1995) presented a companion study for natural convection. Both studies relied on the hypothesis of a fixed volume constraint. In these configurations, the design allowed for the use of a two-dimensional (2-D) numerical model in the simulations. On the other hand, for tubes and plate fin heat exchangers, a three-dimensional (3-D) numerical model based on the Navier–Stokes and energy equations would be required to obtain heat transfer results and perform a similar optimization. Before investing in such extensive computations, it is interesting to access the efficiency potential of the tubes and plate fin arrangement through a simpler approach.

The objective of this paper is to use a simple 2-D numerical model to study one- and two-row circular and elliptic tubes and plate fin heat exchangers to compare performance of the two configurations. The implementation of the model is possible only because the required local heat transfer coefficients by the conduction formulation on the fin were determined experimentally with the naphtalene sublimation technique through the heat and mass transfer analogy (Ximenes 1981). This simple approach makes it possible to test the fin efficiency potential of an elliptic configuration as compared with a circular one, thus avoiding expensive 3-D computations in a first numerical study.

Theoretical model

Figure 1 is a simple sketch of the problem configuration. Only one- or two-row heat exchangers are considered in this study. The governing equation for steady conduction on the fin is (Bejan 1993)

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} - \frac{2h}{Kt}(T - T_b) = 0 \qquad (1)$$

where T and T_b are the unknowns.

The second equation for completing the problem comes from an energy balance between the fin and the air free stream, so that:

$$mc_p(T_b - T_{b,i}) = \int_0^x \mathrm{d}\xi \int_0^{S/2} h(T - T_b)\,\mathrm{d}y \qquad (2)$$

Because $\partial T_b/\partial x \gg \partial T_b/\partial y$, the bulk air temperature T_b is a function of x alone.

Note that, in Equation 2, it has been assumed that the heat exchange between the tubes and air is negligible in the presence of the heat exchange between the air and fin. The accuracy of this assumption will be investigated further when computing the fin efficiency.

Notation

A	total heat exchange area in a unit cell, $A_F + (A_p/4)$, m^2
A_c	minimum air flow area in a unit cell, $(S - 2b)\delta/4$, m^2
A_c^*	dimensionless air flow area in a unit cell, A_c/L^2
A_F	fin heat exchange area in a unit cell, $(SL - r\pi ab)/2$
A_f	frontal flow area of a unit cell, $S\delta/2$, m^2
A_p	lateral area of the elliptical tube in a unit cell, $rp\delta$, m^2
a	bigger semi-axis of the elliptical tube section, m
b	smaller semi-axis of the elliptical tube section, m
c_p	specific heat at constant pressure, J/(kg.K)
D	circular tube diameter, m
Di	mass diffusivity, m^2/s
D_h	hydraulic diameter, $4A_cL/A$ (Kays and London 1968), m
D_h^*	dimensionless hydraulic diameter, D_h/L
e	ellipses eccentricity, b/a
f	force vector
h	local heat transfer coefficient, W/(m^2.K)
h_m	local mass transfer coefficient, m/s
k	fin thermal conductivity, W/(m.K)
k_{air}	air thermal conductivity, W/(m.K)
K	stiffness matrix
L	length of the one-row arrangement, m
m	heat and mass transfer analogy exponent
$\dot{m}$	air mass flow, kg/s
n	number of equations
Nu	local Nusselt number, $h\,D_h/k_{air}$
p	ellipses perimeter, $2\pi[(1/2)(a^2 + b^2)]^{1/2}$
Pr	Prandtl number, $\mu c_p/k_{air}$
$\dot{Q}$	actual air-fin rate, Equation 10, W
$\dot{Q}_i$	air-fin heat rate if the fin were isothermal, Equation 11, W
r	number of rows, 1, 2
Re	Reynolds number, $\dot{m}D_h/(A_c\mu)$
S	tube-to-tube center distance, m
Sc	Schmidt number, ν/Di
Sh	local Sherwood number, h_mD_h/Di
t	fin thickness, m
t^*	dimensionless fin thickness, t/L
T	temperature, K
x, y	Cartesian coordinates, m
x^*, y^*	dimensionless coordinates, x/L, y/L

Greek

δ	fin-to-fin distance, m
η	fin efficiency
θ	dimensionless temperature, $(T - T_{b,i})/(T_t - T_{b,i})$
μ	viscosity, kg/(m·s)
ν	kinematic viscosity, m^2/s
ξ	dummy variable
ρ	density, kg/m^2
$\lvert\cdot\rvert_2$	Euclidean norm

Superscripts

$(\ )^i$	isothermal fin conditions
$\overline{(\ )}$	fin-averaged quantities
$(\ )^j$	solution iteration
$(\ )^k$	mesh refinement iteration

Subscripts

$(\ )_b$	bulk conditions
$(\ )_{b,e}$	bulk conditions at the fin exit
$(\ )_{b,i}$	bulk conditions at the fin input
$(\ )_t$	tube conditions
$(\ )$	fin conditions, no subscript on T and θ

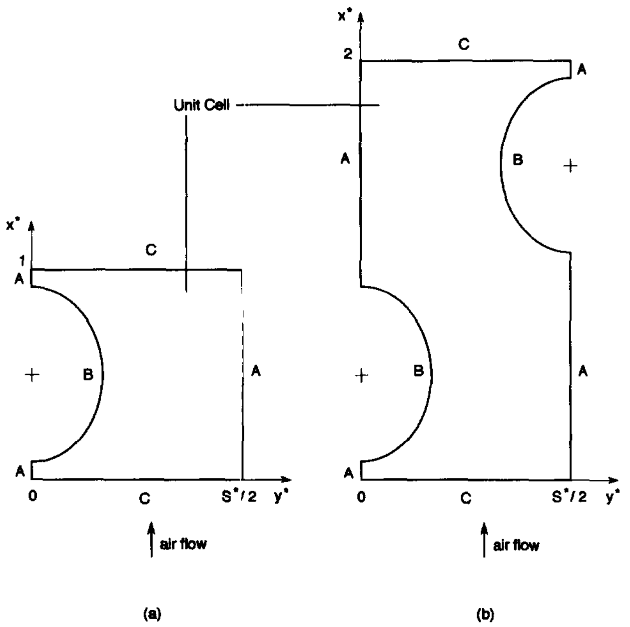

Figure 1 Schematic diagram and computational domain of a unit cell

Equations 1 and 2 are rewritten with appropriate nondimensional groups as follows:

$$\frac{\partial^2\theta}{\partial x^{*2}}+\frac{\partial^2\theta}{\partial y^{*2}}-2\,\mathrm{Nu}\frac{k_{\mathrm{air}}/k}{D_h^* t^*}(\theta-\theta_b)=0 \tag{3}$$

and

$$\theta_b=\frac{1}{\mathrm{Re}\,\mathrm{Pr}\,A_c^*}\int_0^{x^*}\mathrm{d}\xi^*\int_0^{S^*/2}\mathrm{Nu}(\theta-\theta_b)\,\mathrm{d}y^* \tag{4}$$

The nondimensional boundary conditions are as follows:

$$\text{(A) } y^*=0 \text{ and } y^*=\frac{S}{2L}:\ \frac{\partial\theta}{\partial y^*}=0 \tag{5}$$

$$\text{(B) } \theta_t=1 \tag{6}$$

$$\text{(C) } x^*=0 \text{ and } x^*=1:\ \frac{\partial\theta}{\partial x^*}=0 \tag{7}$$

Sparrow and Ramsey (1978) suggested for bundles of tubes in rectangular channels, the following heat and mass transfer analogy:

$$\mathrm{Nu}=\mathrm{Sh}\left(\frac{\mathrm{Pr}}{\mathrm{Sc}}\right)^m \tag{8}$$

where $m=0.4$, for the system air-naphtalene, $\mathrm{Pr}=0.7$ (air), and $\mathrm{Sc}=2.5$ (air-naphtalene). Because Pr and Sc are constant, Equation 8 is also valid for $\overline{\mathrm{Nu}}$ and $\overline{\mathrm{Sh}}$.

The $\mathrm{Nu}(x^*, y^*)$ distribution required to solve the system of Equations 3–7 is obtained by means of experimental data (Ximenes 1981), through the heat and mass transfer analogy, using Equation 8, with $m=0.4$.

By definition, the fin efficiency η is expressed as the ratio between the actual fin–air heat rate and the fin–air heat rate if the fin were isothermal and with the same temperature as the tube surface. Therefore:

$$\eta=\frac{\dot{Q}}{\dot{Q}_i} \tag{9}$$

where

$$\dot{Q}=\dot{m}c_p(T_{b,e}-T_{b,i})=\dot{m}c_p\theta_{b,e}(T_t-T_{b,i}) \tag{10}$$

$$\dot{Q}_i=\dot{m}c_p(T_{b,e}^i-T_{b,i})=\dot{m}c_p\theta_{b,e}^i(T_t-T_{b,i}) \tag{11}$$

Thus, Equation 9 is rewritten as follows:

$$\eta=\frac{\theta_{b,e}}{\theta_{b,e}^i} \tag{12}$$

where $\theta_{b,e}$ comes from the solution of Equations 3–7 and $\theta_{b,e}^i$ is the solution of Equation 4, with $\theta=1$ (isothermal fin).

Next we return to the assumption adopted for writing Equation 2 and 4, which neglect the air–tubes heat exchange in presence of the air–fin heat exchange. The resulting effect is a discrepancy that affects similarly $\theta_{b,e}$ and $\theta_{b,e}^i$. Hence, in the computation of the efficiency η, given by Equation 12, almost no loss of accuracy is expected with the present model.

The energy balance on a $\mathrm{d}x\times S/2$ element of the isothermal fin reads as follows:

$$\mathrm{d}q_i=\dot{m}c_p\,\mathrm{d}T_b^i=\mathrm{d}x\int_0^{S/2}h(T_t-T_b^i)\,\mathrm{d}y \tag{13}$$

Noting T_b^i is a function of x alone, and T_t is a constant (isothermal tube wall), Equation 13 is rewritten as

$$\dot{m}c_p\frac{\mathrm{d}T_b^i}{T_t-T_b^i}=\mathrm{d}x\int_0^{S/2}h\,\mathrm{d}y \tag{14}$$

Integrating over the fin, we write

$$\dot{m}c_p\int_{T_{b,i}}^{T_{b,e}^i}\frac{\mathrm{d}T_b^i}{T_t-T_b^i}=\int_0^L\mathrm{d}x\int_0^{S/2}\mathrm{d}y \tag{15}$$

The fin-averaged heat transfer coefficient is then written as follows:

$$\bar{h}=\frac{\dot{m}c_p}{A_f}\ln\left(\frac{1}{1-\theta_{b,e}^i}\right) \tag{16}$$

Next, using Equation 16, the heat exchanged between the air free stream and the isothermal fin is written as follows:

$$\frac{\dot{Q}_i}{T_t-T_{b,i}}=\dot{m}c_p\theta_{b,e}^i=A_F\bar{h}\frac{\theta_{b,e}^i}{\ln\left(\dfrac{1}{1-\theta_{b,e}^i}\right)} \tag{17}$$

The nondimensional bulk temperature at the fin outlet $\theta_{b,e}^i$ in terms of Pr, $\overline{\mathrm{Nu}}$, Re, A_F^*, and A_c^* is given by:

$$\theta_{b,e}^i=1-e^{-(A_F^*/A_c^*)(\overline{\mathrm{Nu}}/\mathrm{Re}\,\mathrm{Pr})} \tag{18}$$

The fin-averaged Nusselt number $\overline{Nu}$ was reported by Rosman et al. (1984) for the circular arrangement and the fin-average Sherwood number $\overline{Sh}$ by Ximenes (1981) for the elliptic arrangements, in the form of correlations ($200 < Re < 1500$) as follows

Circular tubes:

$$\frac{\overline{Nu}}{Pr^{0.4}} = 4.18 + 1.5 \times 10^{-3} Re^{1.15} \text{ (one row)} \tag{19}$$

$$\frac{\overline{Nu}}{Pr^{0.4}} = 3.58 + 8.46 \times 10^{-4} Re^{1.24} \text{ (two row)} \tag{20}$$

Elliptic tubes:

$e = 0.5$:

$$\overline{Sh} = 7.62 + 5.12 \times 10^{-3} Re \text{ (one-row)} \tag{21}$$

$$\overline{Sh} = -3.778 \times 10^{-6} Re^2 + 0.013 Re + 3.413 \text{ (two-row)} \tag{22}$$

$e = 0.65$:

$$\overline{Sh} = 2.43 + 3.44 \times 10^{-1} Re^{0.49} \text{ (one row)} \tag{23}$$

$$\overline{Sh} = 4.123 \times 10^{-7} Re^2 + 0.008 Re + 3.636 \text{ (two row)} \tag{24}$$

For the elliptical arrangement it is necessary to use Equation 8 to obtain the fin-averaged Nusselt number from the values of the fin-averaged Sherwood number given by Equations 21–24.

Note that in a practical engineering approach, for a larger number of rows, the fin-averaged Nusselt number computed for two rows is a fairly good approximation. This is explained by the fact that, with a large number of rows, the flow will be fully developed; therefore, with no significant changes in the fin-averaged Nusselt number for a particular geometry, either circular or elliptic. This behaviour is observed experimentally comparing three-row circular results reported by Saboya and Sparrow (1976), with two-row circular results from Equation 20.

The actual air–fin heat exchange is obtained from a balance of energy combined with Equation 12, by the following:

$$\dot{Q} = \dot{m} c_p \eta \theta^i_{b,e} (T_t - T_{b,i}) \tag{25}$$

Numerical method and results

Equations 3–7 were solved numerically by a second-order finite-differences scheme. The method is summarized by the following algorithm:

(1) Starting with $\theta = 1$ (isothermal fin), obtain θ_b by numerical integration of Equation 4;
(2) Solve Equations 3 and 5–7 for $\theta(x^*, y^*)$ using θ_b obtained from step 1, with a second-order central differences scheme;
(3) Using $\theta(x^*, y^*)$, recalculate θ_b with Equation 4;
(4) Convergence is achieved with simultaneous verification of the following criteria:

$$\frac{|\theta^{j+1} - \theta^j|_2}{|\theta^j|_2} < 10^{-5} \tag{26}$$

and

$$\frac{|\theta_b^{j+1} - \theta_b^j|_2}{|\theta_b^j|_2} < 10^{-5} \tag{27}$$

(5) If the conditions established in step 4 are not satisfied, update $\theta_b^j = \theta_b^{j+1}$ and $\theta^j = \theta^{j+1}$ and restart the process from step 2;
(6) Once Equations 20 and 21 are verified, θ^j and θ_b^j represent the converged solution to Equations 3–7.

The appropriate mesh, for the domain represented in Figure 1 was obtained by successive refinements, starting with a coarse mesh. The previous algorithm was applied to each mesh identified by an index k, determining θ^k. The k mesh so that results do not change anymore was determined according to the following 1% criterion:

$$\frac{|\theta^{k+1} - \theta^k|_2}{|\theta^k|_2} < 10^{-2} \tag{28}$$

Regular meshes with different total number of nodes were determined to obtain the results shown in this section, according to the geometry of each heat exchanger being analyzed.

The resulting system of n algebraic equations is solved numerically for θ^j at each j-iteration of the algorithm presented in this section. In matricial form, we write:

$$K^j \theta^j = f^j \tag{29}$$

where K^j is a square matrix $n \times n$, of constant coefficients determined by Δx, Δy and θ_b^j through Equations 3–7. Similarly the n-vector f^j is determined.

K^j is a banded matrix, and ill-conditioned due to the existence of the Neumman boundary conditions in the problem; therefore, a direct solver for banded systems, based on LINPACK FORTRAN subroutines (Dongarra et al. 1978) was applied to solve the linear systems defined by Equation 29.

Numerical results were obtained for air ($Pr = 0.7$), for one- and two-row tubes and plate fin heat exchangers. Initially, to test the code herein developed, result for circular tubes are obtained and compared to previous literature results (Rosman et al. 1984). Finally, new results are computed for elliptical tubes with eccentricity, $e = 0.5$ and 0.65.

Rosman et al. (1984) computed the fin efficiency for circular tubes and plate fin heat exchangers making use of local Nusselt numbers obtained experimentally, with the heat and mass transfer analogy exponent $m = 0.4$. It was observed that the use of fin-averaged Nusselt numbers in the fin efficiency computations produced a maximum discrepancy of 8% for high Reynolds numbers, and excellent agreement for low Reynolds numbers, as compared with the results obtained with local Nusselt numbers. The fin efficiency results for circular tubes in this study were obtained with fin-averaged Nusselt numbers and compared to results obtained with local Nusselt numbers by Rosman et al. (1984). The nondimensional parameters for the experimental setup were: $(S/L) = 1.1545$, $(D/L) = 0.4622$, $(\delta/L) = 0.0894$, and $(t/L) = 0.0082$ (Rosman 1979). Figure 2 shows the fin efficiency results for one- and two-row heat exchangers obtained in this work in comparison with the results reported by Rosman et al. (1984). The results are in good agreement, except for the two-row heat exchanger with high Reynolds numbers, as expected by the use of fin-averaged Nusselt numbers in the present computations.

The results for one- and two-row elliptic tube and plate fin heat exchangers were obtained using Nusselt numbers computed from experimental measurements made by Ximenes (1981) and the heat and mass transfer analogy exponent $m = 0.4$. The same parameters S/L, δ/L, and t/L as the circular setup were used

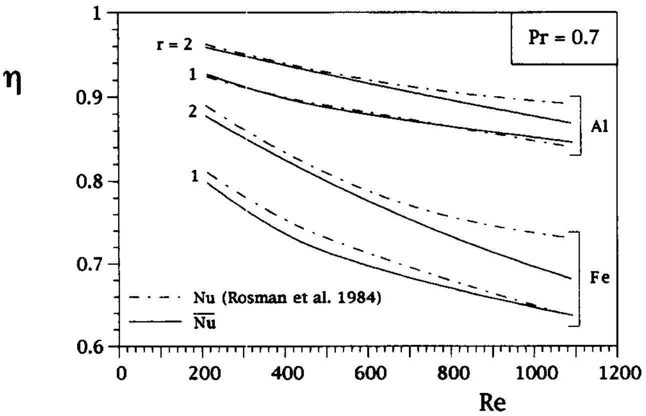

Figure 2 Fin efficiency versus Re for one and two-row circular tubes and plate fin heat exchangers (aluminum and iron)

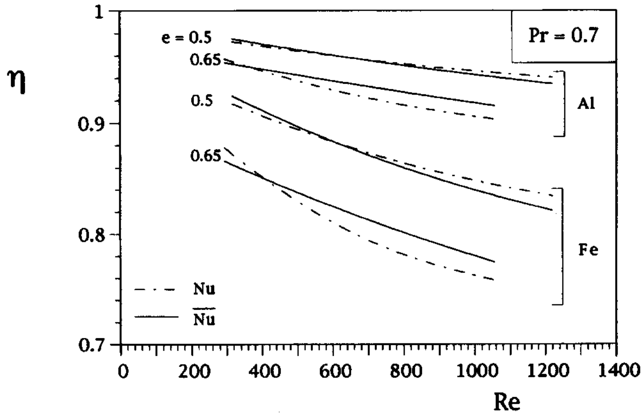

Figure 4 Fin efficiency versus Re for two-row elliptical tubes and plate fin heat exchangers (aluminum and iron)

to build the elliptical one. Two new parameters were necessary in the elliptical arrangement, $b/L = 0.2305$, and $e = 0.5$ and 0.65.

Figures 3 and 4 show the behavior of the fin efficiency with respect to the variation of the Reynolds number, for one- and two-row heat exchangers, respectively. Results obtained for local and fin-averaged Nusselt numbers are compared, both for aluminum and iron. In general, there is good agreement, and the most unfavorable situation is observed for iron heat exchangers with $e = 0.65$, with a maximum discrepancy of 7.5%. Note that, in all cases, the aluminum heat exchangers are more efficient than iron heat exchangers, because the aluminum has a higher thermal conductivity than the iron and, consequently, with the fin operating in almost isothermal conditions.

Figures 5 and 6 illustrate the temperature distribution of iron plate fins of one- and two-row heat exchangers with Re = 299 and 1576, respectively, with $e = 0.5$. The effect of the variation of the Reynolds number is observed comparing Figure 5 with Figure 6. The isotherms show that as Re increases, the plate fin temperature drops to lower levels, thus the fin efficiency decreases.

Figures 7 and 8 show fin efficiency results versus the variation of the Reynolds number for one- and two-row circular and elliptic tubes heat exchangers made from iron and aluminum, respectively. All results were obtained based on the local Nusselt numbers. As observed in the previous results, the one-row arrangement is less efficient than the two-row one. The elliptical arrangement is more efficient than the circular one. Furthermore, the most efficient design is the two-row one with $e = 0.5$.

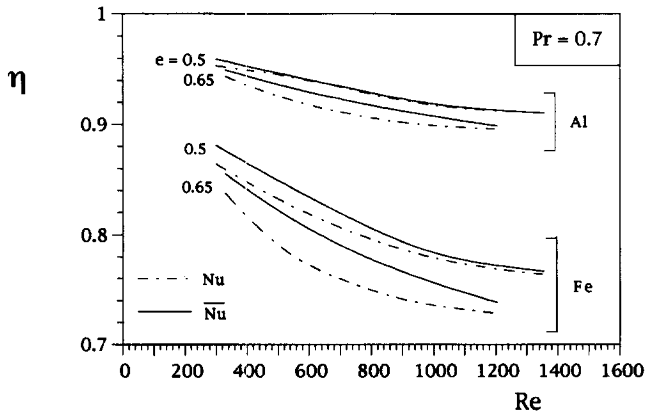

Figure 3 Fin efficiency versus Re for one-row elliptical tubes and plate fin heat exchangers (aluminum and iron)

In a quantitative analysis, it is interesting to note that a 18% maximum relative efficiency gain, in comparison with the traditional circular arrangement, was observed for the elliptical arrangement with $e = 0.5$. This result shows excellent agreement with a 15% gain of previous experimental observations reported by Brauer (1964).

Concluding Remarks

In this paper, we developed a theoretical model to study elliptic tubes and plate fin heat exchangers. New heat transfer solutions are obtained numerically, for the one- and two-row elliptic arrangements, and results for the circular arrangement are compared to previous literature results (Rosman et al. 1984) with good agreement. The elliptical arrangement with eccentricity $e = 0.5$, is the most efficient one, among the cases studied in this paper, either in the one-row or in the two-row setup. The efficiency is higher for the two-row heat exchanger, confirming a trend already observed by Rosman et al. (1984) for circular tubes. Because of its higher thermal conductivity, the aluminum heat exchangers show a higher efficiency than the iron ones.

From the heat transfer point of view, we showed that the elliptic configuration is more efficient than the circular one.

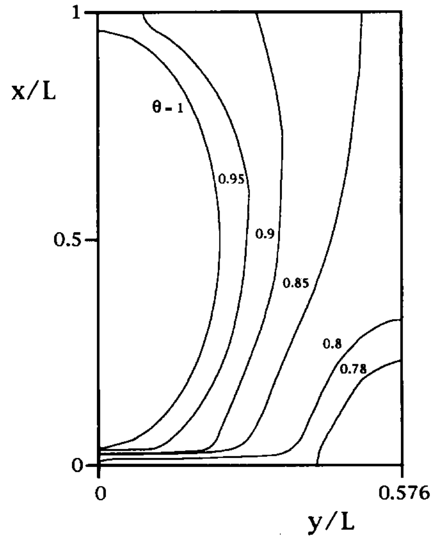

Figure 5 Isotherms for one-row elliptical tubes and plate fin heat exchangers (e=0.5; iron; Re=299)

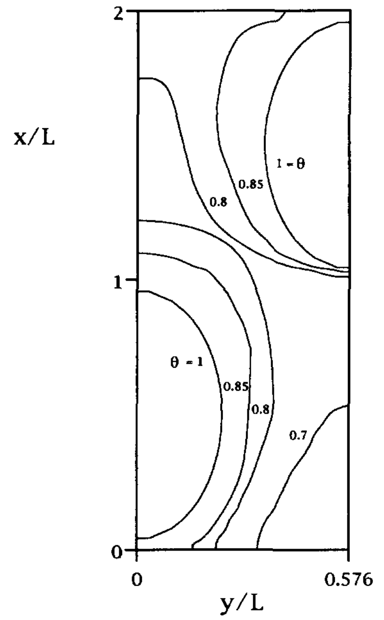

Figure 6 Isotherms for two-row elliptical tubes and plate fin heat exchangers (e=0.5; iron; Re=1576)

Among the cases studied in this paper, the maximum relative efficiency gain was 18% for $e = 0.5$. Brauer (1964) reported relative reductions in pressure drop up to 18%, comparing the elliptic configuration with the circular one. Recently, Bordalo and Saboya (1995) reported experimental results showing relative pressure drop reductions of up to 25%, in a similar experiment (this observation was registered in a comparison between a three-row elliptical arrangement with $e = 0.5$ and a similar circular arrangement).

Based on the combination of these two effects, fin efficiency gain and pressure drop reduction, the main conclusion of this work is that the elliptic tubes and plate fin heat exchangers are expected to have a considerable better overall performance than the circular tubes and plate fin heat exchangers, operating under similar conditions. This conclusion provides the necessary basis for the investment in developing a 3-D model for the circular and elliptic tubes and plate fin configuration, with an arbitrary number of rows and tubes, and the natural follow-up optimization study, under the fixed volume constraint proposed for heat

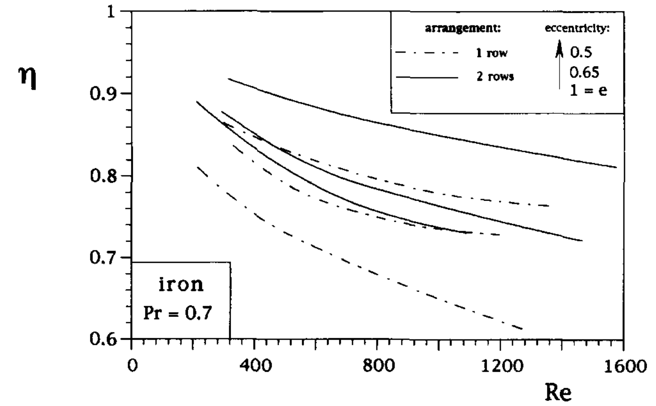

Figure 7 Fin efficiency versus Re for one and two-row elliptical and circular tubes and plate fin heat exchangers (iron)

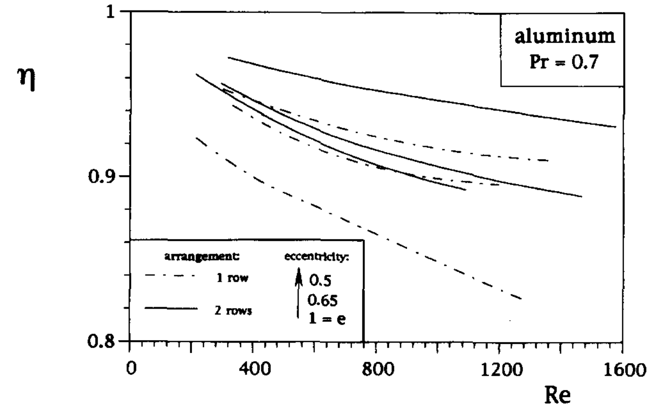

Figure 8 Fin efficiency versus Re for one and two-row elliptical and circular tubes and plate fin heat exchangers (aluminum)

exchangers without plate fins, in previous studies (Bejan et al. 1995; Stanescu et al. 1996).

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