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TRANSFER COEFFICIENTS FOR PLATE FIN AND ELLIPTICAL TUBE HEAT EXCHANGERS

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SUMÁRIO

Com a finalidade de se determinar coeficientes de transferência para trocadores de tubos elíticos aletados, experiências de troca de massa foram realizadas utilizando-se a técnica de sublimação do naftaleno. Por meio da analogia entre transferência de calor e massa, os resultados podem ser convertidos em resultados para troca de calor. Os coeficientes de transferência foram comparados com aqueles para trocadores de tubos circulares e a comparação não revelou grandes diferenças. Esta é uma conclusão positiva, porque o uso de tubos elíticos pode reduzir consideravelmente a queda de pressão, sem afetar as características de transferência. (autor)

SUMMARY

In order to determine transfer coefficients for plate fin and elliptical tube exchangers, mass transfer experiments have been performed using the naphthalene sublimation technique. By means of the heat-mass transfer analogy, the results can be converted to heat transfer results. The transfer coefficients were compared with those for circular tube exchangers and the comparison revealed no major differences. This is a positive outcome, since the use of elliptical tubes may reduce substantially the pressure drop, without affecting the transfer characteristics. (author)

1. Introduction

The present paper is concerned with the transfer characteristics of plate fin and elliptical tube heat exchangers used in air conditioning machines. A survey of published heat transfer information relating to such heat exchange devices revealed that the most extensive set of results is concerned with circular tube exchangers.

For circular tube exchangers with plate fins, the results reported by Shepherd [1], Saboya and Sparrow [2,3], constitute the most complete information available in the literature. Several heat exchanger configurations were studied. In[2,3], the tool for obtaining the transfer coefficients was the heat-mass transfer analogy in conjunction with the naphthalene sublimation technique.

In the case of finned elliptical tube exchangers, a paper by Schulenberg [4] appears to be the only significant work. Although interesting and informative, the paper is much more qualitative than quantitative.

In the present research, experimental results are provided for the average transfer coefficients in rectangular plate fin and elliptical tube heat exchangers. The heat transfer problem was simulated by means of the naphthalene sublimation technique and the heat-mass transfer analogy.

To employ the method, naphthalene plates were cast in a specially designed mold. A pair of such plates, spaced apart by elliptical disks, formed the analogical system. Average transfer coefficients were determined by weighing the plates before and after a data run with a precision balance.

The experimentally determined average transfer coefficients will be presented in dimensionless form and compared with similar results for finned circular tube heat exchangers provided by [2,3]. As it is verified by the results that will be reported later, the replacement of the circular tubes by elliptical tubes does not affect the rate of heat transfer adversaly. The performance advantage of the elliptical tubes results from their lower pressure drop due to the smaller wake region on the fin behind the tube [5]. A schematic diagram of the physical problem under study is presented in Figs. 1 and 2.

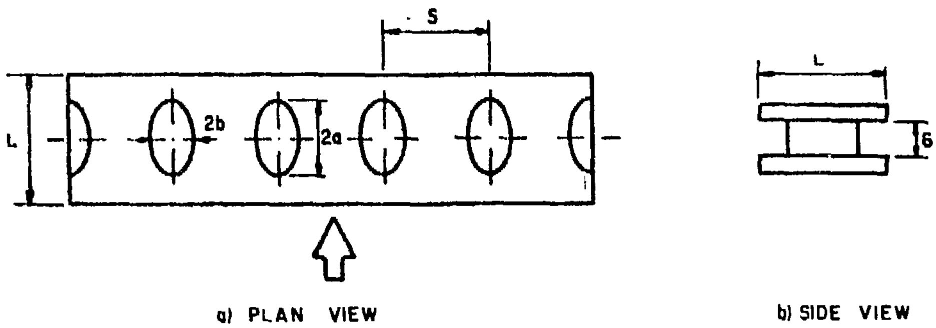

Fig. 1. Schematic of one-row plate fin and elliptical tube heat exchanger

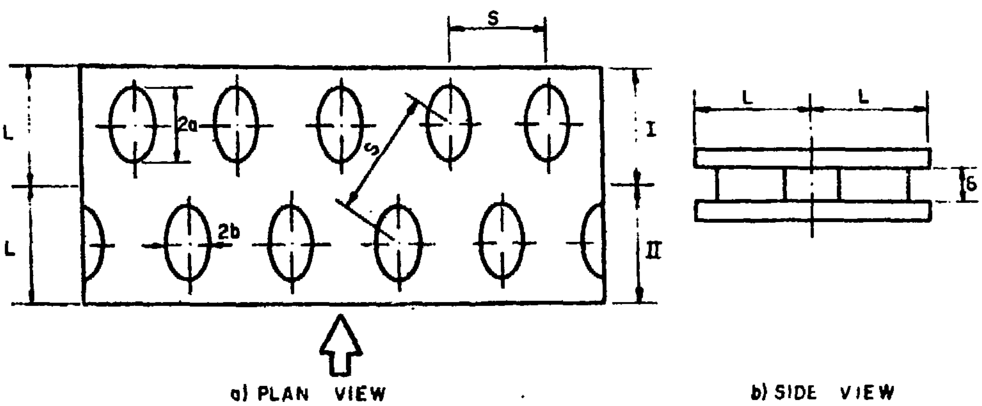

Fig. 2. Schematic of two-row plate fin and elliptical tube heat exchanger

Figs. 1 and 2 contain dimensional nomenclature designated by a, b, L, S and δ. The dimensionless parameters which govern the results are: $\delta/2b$; $S/2b$; $L/2b$; b/a ; Re; Sc or Pr. Re is the flow Reynolds number, Pr is the Prandtl number and Sc the Schmidt number. The dimension ratios for the present test apparatus were: $\delta/2b = 0.193$; $S/2b = 2.50$; $L/2b = 2.17$; $b/a = 0.50$ and 0.65. These values are typical of heat exchangers encountered in air conditioning machines. The actual values of the apparatus dimensions are: $\delta = 1.65$ mm ; $L = 18.50$ mm ; $S = 21.30$ mm ; $2a = 13.12$ mm and 17.06 mm ; $2b = 8.53$ mm. As mentioned before, the present results will be compared with the results of [2,3] for circular tubes. The dimension ratios for the test apparatus of [2,3] are identical to those of the present configurations. It is only necessary to replace 2b by the tube diameter.

The Reynolds number of the flow was varied during the course of the research from 150 to 1300. Once again, this range is relevant to air conditioning applications. The Schmidt number, which is the analogue of the Prandtl number, is 2.5 for the naphthalene air system.

2. Test Apparatus and Experimental Procedures

Fig. 3 is a schematic side view of the test apparatus. As shown, air from the laboratory room is draw into the channel formed by the plates. Upon leaving the test section, the air exits to a plenum chamber from which it passes successively to a calibrated flow meter and a blower, and then to an exhaust system.

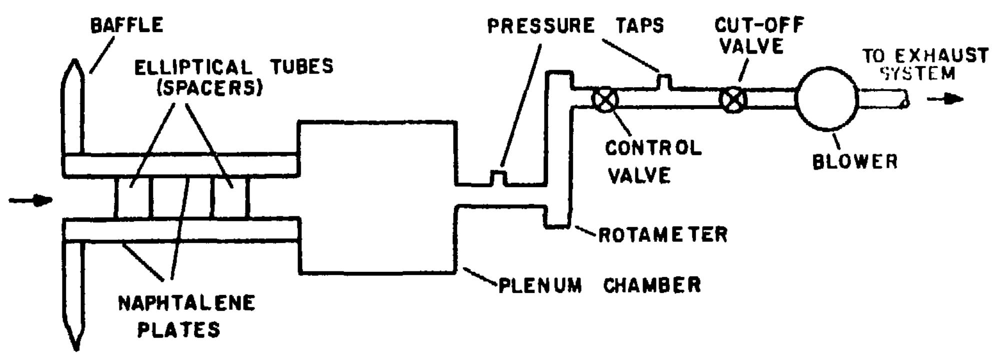

Fig. 3. Schematic side view of the test apparatus

The naphthalene plates used in the experiments were cast in a metallic mold whose components had been polished to a high degree of smoothness. The details of the mold fabrication, of the casting and removal processes are described in [6].

For the overall mass transfer, the measurements were made with a Sartorius precision balance which could be read to within 0.05 milligrams for specimens having a mass up to

200 g. Measurements of the flow rate were made with a calibrated rotameter. Typically, the uncertainty of the flow rate measurements was only 1 percent. The temperature of the air entering the test section was given by a laboratory thermometer that could be read to 0.1 deg C. The duration time of a data run was measured with a timer capable of discriminating to within 0.1 sec. The atmospheric pressure was sensed by a barometer of column of mercury whose smallest scale division was 0.1 mm.

3. Evaluation of Results

Attention will now be given to a brief description of the data reduction procedures. Further details are available in [6].

Let M_T denote the overall mass transfer, A the corresponding fin surface area, τ the duration time of a data run, $(\Delta\rho_n)_m$ the log-mean concentration difference, and D_m the coefficient of mass diffusion. A dimensionless representation can then be made by introducing the average Sherwood number $\overline{Sh}$

$$\overline{Sh} = M_T\ (2\delta)/A\ (\Delta\rho_n)_m\ \tau\ D_m \tag{1}$$

The log-mean concentration difference $(\Delta\rho_n)_m$ depends on the bulk concentration of naphthalene vapor at the exit $(\rho_n)_L$ and the wall concentration $(\rho_n)_w$. The bulk concentration of naphthalene vapor in the air entering the exchanger is zero. At the exit $(\rho_n)_L = M_T/\dot{V}\tau$, where $\dot{V}$ is the volume flow rate; at the wall $(\rho_n)_w$ is a function of the room temperature, as given by the Sogin vapor pressure correlation [7].

The Reynolds number, employed as independent variable, is given by

$$Re = (2\delta)\ G/\mu \tag{2}$$

where G is the mass velocity based on the frontal flow area.

The conversion of the Sherwood number results to Nusselt number can be accomplished by the relation

$$\overline{Nu} = (Pr/Sc)^m \, \overline{Sh} \qquad (3)$$

where m is 0.4 [6].

A detailed error analysis of the experimentally determined Sherwood and Reynolds numbers is described in [6]. From equation (1), the uncertainty in the Sherwood number was found to be within 7.3 percent. From equation (2), the uncertainty in the Reynolds number was in the range 2.0 - 3.0 percent.

4. Results and Discussion

Average mass transfer coefficients, expressed in terms of the average Sherwood number $\overline{Sh}$, were evaluated from equation (1). These results are plotted in Figs. 4, 5, 6, 7 and 8. The ordinate variable is $\overline{Sh}$ and the abscissa variable is Re, which was calculated from equation (2).

Figs. 4, 5, 6 and 7 also contain the results for circular tube heat exchangers provided by references [2] and [3]. Inspection of the figures reveals that there is no major differences between the present results, for b/a = 0.65 and 0.50, and those from [2,3], for b/a = 1.0 (circular tubes). A small difference, in favor of the elliptical tubes, can be observed in Figs. 6 and 7, for b/a = 0.50. It should be noticed that Figs. 4 and 6 are for one-row heat exchangers, while Figs. 5 and 7 are for two-row heat exchangers. In Figs. 4 and 5 the curves for b/a = 0.65 and 1.0 are coincident.

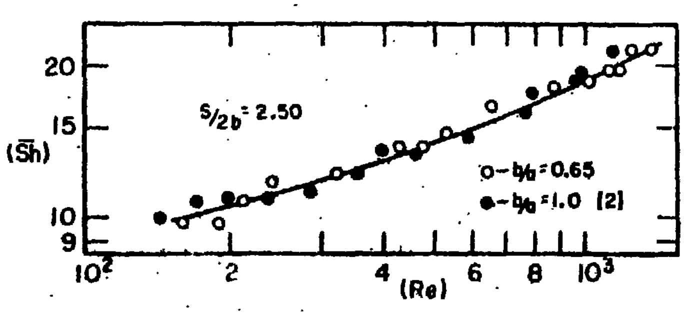

Fig. 4. Sherwood numbers for one-row exchangers

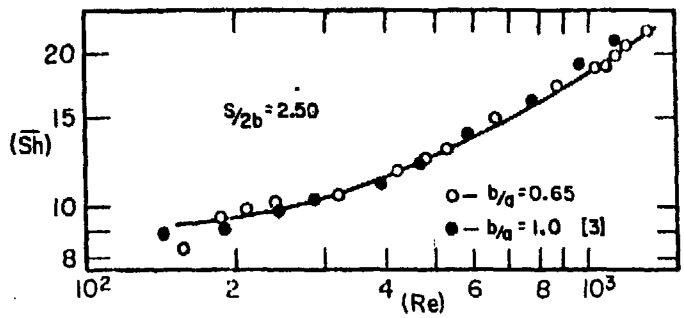

Fig. 5. Sherwood numbers for two-row exchangers

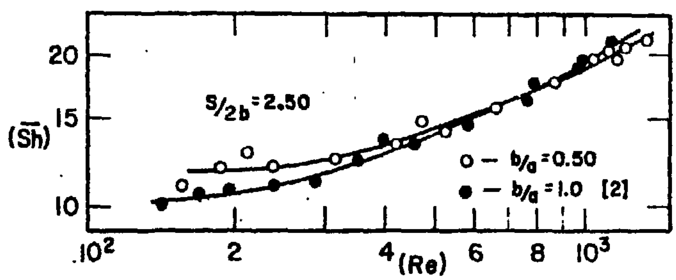

Fig. 6. Sherwood numbers for one-row exchangers

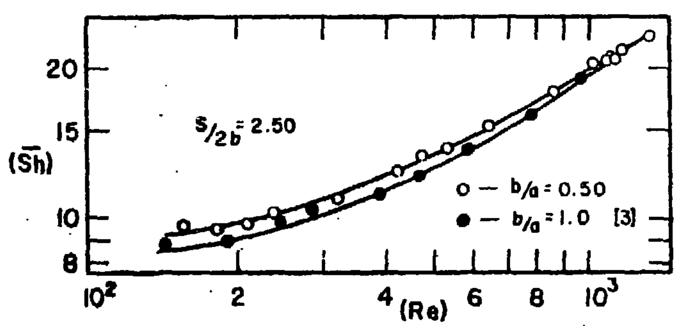

Fig. 7. Sherwood numbers for two-row exchangers

The most important finding of the present work is that the replacement of the circular tubes by elliptical tubes does not affect the rate of heat transfer adversaly. This is a fortunate outcome, since the elliptical tubes may reduce substantially the pressure drop [5] without affecting the transfer characteristics.

Fig. 8 shows the Sherwood number results for two elliptical tube exchangers, both with one row of tubes and b/a = 0.50. One has S/2b = 2.50 and the other has S/2b = 3.53. It is seen that the exchanger with smaller distance between the elliptical tubes has higher transfer coefficients. This is due to the acceleration of the air in the region between the tubes. Then, the use of elliptical tubes makes possible the reduction of the tube distance, increasing the transfer coefficients, without increasing the pressure drop to much.

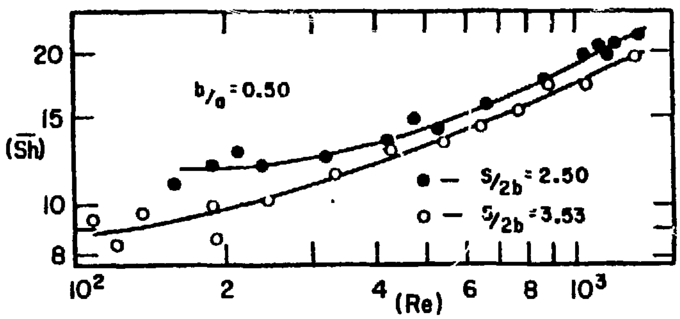

Fig. 8. Effect of the tube distance on the transfer coefficients of one-row elliptical tube exchangers

Table 1 gives the Sherwood-Reynolds correlations for the several cases discussed. These correlations were obtained by least-square fitting of the experimental data. Tipically, the average dispersion was 2.5 percent.

Table 1. Sherwood-Reynolds Correlations

b/a	S/2b	Number of Rows	Correlations
1.0	2.50	1	$\overline{Sh} = 9.20 + 3.64 \times 10^{-3}(Re)^{1.15}$
1.0	2.50	2	$\overline{Sh} = 7.82 + 2.10 \times 10^{-3}(Re)^{1.24}$
0.50	3.53	1	$\overline{Sh} = 6.59 + 7.36 \times 10^{-2}(Re)^{0.72}$
0.50	2.50	1	$\overline{Sh} = 10.14 + 8.58 \times 10^{-3}(Re)^{1.00}$
0.50	2.50	2	$\overline{Sh} = 6.82 + 2.63 \times 10^{-2}(Re)^{0.89}$
0.65	2.50	1	$\overline{Sh} = 3.45 + 5.28 \times 10^{-1}(Re)^{0.49}$
0.65	2.50	2	$\overline{Sh} = 6.52 + 2.98 \times 10^{-2}(Re)^{0.86}$

In the case of the two-row plate fin and tube heat exchangers, overall rates of mass transfer have also been determined for the portions of the fin associated with the first and second rows of tubes. The quantities $\dot{M}_I$ and $\dot{M}_{II}$ respectively denote the overall rates of mass transfer in regions I and II of Fig. 2. $\dot{M}_T$ is defined as $\dot{M}_I + \dot{M}_{II}$. The relative transfer capabilities of the two portions of the fin are given in Table 2. From Table 2, it is seen that region I is more efficient than region II when the Reynolds number is low. As the Reynolds number increases, more and more of a parity is established between the two regions. This is due to a vortex systema [3], adjacent to the second row of tubes, which grows in intensity as the Reynolds number increases.

Table 2. Relative Transfer Capabilities of Regions I and II

Re	b/a = 1.0		b/a = 0.65		b/a = 0.50	
	$\dot{M}_I/\dot{M}_T$	$\dot{M}_{II}/\dot{M}_T$	$\dot{M}_I/\dot{M}_T$	$\dot{M}_{II}/\dot{M}_T$	$\dot{M}_I/\dot{M}_T$	$\dot{M}_{II}/\dot{M}_T$
150	0.64	0.36	0.60	0.40	0.67	0.33
200	0.62	0.38	0.60	0.40	0.65	0.35
250	0.61	0.39	0.60	0.40	0.62	0.38
350	0.59	0.41	0.60	0.40	0.59	0.41
450	0.58	0.42	0.60	0.40	0.57	0.43
600	0.57	0.43	0.58	0.42	0.54	0.46
700	0.56	0.44	0.57	0.43	0.53	0.47
800	0.55	0.45	0.57	0.43	0.52	0.48
1000	0.54	0.46	0.55	0.45	0.50	0.50
1200	0.52	0.48	0.53	0.47	0.49	0.51

5. Concluding Remarks

It has been demonstrated that the replacement of circular tubes by elliptical tubes, in finned heat exchangers, does not decrease the rate of heat transfer. The performance advantage of the elliptical tubes results from their lower pressure drop [5]. It has also been demonstrated that a reduction of the distance between the tubes increases the transfer coefficients. Such a reduction may increase the pressure drop too much, if circular tubes are used. Then, the elliptical tubes may offer significant performance advantage in multi-row plate fin and tube heat exchangers. It is only necessary to decrease S/2b.

Before closing, it should be said that the production of elliptical tubes is today completely automatic. Schulenberg [4] reported that his company produced in 1966 more than 70 miles of this type of tube per week. This is a conclusive proof that there is no commercial disadvantage when cost comparison is made with conventional heat exchanger tubes.

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