Finned tube heat exchanger : correlation of dry surface heat transfer data

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FINNED TUBE HEAT EXCHANGER: CORRELATION OF DRY SURFACE HEAT TRANSFER DATA

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RESUME

Les échangeurs tubulaires à ailettes sont largement utilisés dans l'industrie et en particulier dans les secteurs du chauffage, de la réfrigération et de la climatisation. L'auteur utilise les données de transfert de chaleur de faisceaux de tubes circulaires avec plaques-ailettes continues ou circulaires pour mettre au point une méthode empirique de prédiction de la variation du facteur de transfert thermique moyen (J) avec le nombre de Reynolds côté air.

L'auteur assume que du côté air, pour un nombre de Reynolds entre 200 et 2000 environ, la relation entre J et Re peut s'exprimer par la fonction:

$$J = C_1 \, Re^{C_2}$$

L'auteur prend aussi pour acquis que C_1 et C_2 dépendent des dimensions physiques de l'échangeur thermique. Il utilise une méthode progressive d'analyse par régression linéaire multiple et les données de transfert de chaleur déjà publiées pour déterminer les groupes de dimensions de couronne qui ont la plus grande influence afin de définir C_1 et C_2. Une comparaison des valeurs C_1 et C_2 calculées selon cette technique avec des valeurs expérimentales indique que la présente technique est satisfaisante.

Afin de mieux vérifier la méthode, l'auteur obtient des données expérimentales pour un faisceau de 8 rangées du même type. Les résultats démontrent que la présente méthode peut être utilisée pour ce type d'échangeur dans les limites données de nombres de Reynolds côté air et les limites des différentes groupes adimensionnels données dans le texte.

No. 2544

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ABSTRACT

Finned tube heat exchangers are widely used in industry and, in particular, in the area of heating, cooling and air conditioning. Heat transfer data of tube banks with circular tubes and circular or continuous flat plate fins are used to develop an empirical technique to predict the variation of the average heat transfer J-factor with the air side Reynolds number.

It is assumed that over the air side for Reynolds number between about 200 and 2000, the functional relationship between J and Re is:

$$J = C_1 \, Re^{C_2}$$

Also, it is assumed that C_1 and C_2 are dependent on the physical dimensions of the heat exchanger. A step-by-step multiple linear regression analysis method and the previously published heat transfer data are used to determine the most influential groups of coil dimensions to determine the quantities C_1 and C_2. Comparison between the quantities C_1 and C_2 calculated using this technique with those determined experimentally indicate that the present method is satisfactory.

To further verify this method, experimental data were obtained on an eight row coil of the same family. The results showed that the present method can be used for this type of heat exchanger over the given range of air side Reynolds number and the range of the different dimensionless groups given in the text.

INTRODUCTION

Rapidly escalating costs and decreasing supplies of energy result in an increasing emphasis on economic energy performance as one of the building design criteria. Consequently, accurate estimates of the performance of the mechanical components of the heating, ventilating and air-conditioning (HVAC) system are required at various phases of the design process. Finned tube heat exchangers are among the mechanical components that must be given considerable attention, since they represent the thermodynamic link between the water and air sides of the building HVAC system.

The energy analysis computer programs used to predict energy requirements of buildings demand, as input information, the characteristic performance of the heat exchanger.

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The thermal performance of multi-row finned tube heat exchangers is dependent upon parameters related to their physical dimensions in addition to the usual hydrodynamic parameters. Standard methods [1] have been established to predict heat exchanger performance using measurements taken over a range of air and water conditions. For example, the "effectiveness" method adopted by ASHRAE [2] is adequate in determining the heat transfer rate of a heat exchanger. Nevertheless, this method is dependent on the evaluation of the overall heat transfer coefficient U. The latter is calculated using the expression:

$$U = \frac{\dot{m} c_p}{A} \left[\frac{1}{\alpha \dot{m} + \beta \dot{m}^{0.2}} \right]^*$$ (1)

The constants α and β in Eq 1 are dependent upon physical characteristics and must be uniquely determined for each coil. For example, some coil simulation programs [3 and 4] require the variation of the air side heat transfer coefficient with the air flow Reynolds number for heat transfer rate calculation. Others, however, use the coil manufacturers' data which, in some cases, are not necessarily experimental data, but include extrapolated or interpolated points.

In this paper, previously published heat transfer data for dry, finned-tube exchangers are used to develop an empirical technique to calculate the average film heat transfer coefficient on the air side. This analysis is restricted to the family of heat exchangers with staggered circular tubes and either circular or continuous flat plate fins. This type of heat exchanger is the most commonly used in heating, ventilating and air-conditioning applications.

This analysis is aimed at determining the air side heat transfer coefficient h only. The overall U coefficient can be determined from h and the other thermal resistances, including those due to conduction, contact, water side convective heat transfer and conduction through extended surfaces.

ANALYSIS

The thermal characteristic performance of a finned tube heat exchanger is usually expressed in terms of the relation between the air side Reynolds number Re, and the average heat transfer Colburn J-factor [5] as follows.

$$J = \psi \ (\text{Re})$$ (2)

where Re is based on the coil hydraulic diameter D_h [6]. The average heat transfer J-factor is based on the maximum air mass flux G, and is defined as:

$$J = \left(\frac{h}{G \ c_p} \right) \ Pr^{2/3}$$ (3)

where h is the average film heat transfer coefficient on the air side, and is based on the log-mean temperature difference [6].

Experimental heat transfer data for representative coils, when plotted as J-factor vs Re, suggests that the functional relationship given by Eq 2 can be expressed as:

$$J = C_1 \ Re^{C_2}$$ (4)

The quantities C_1 and C_2 are constants for a particular coil over the Re range of 200 to 2000 considered in this case. It is postulated that C_1 and C_2 are dependent on the physical characteristics of the heat exchanger. Determination of the most appropriate coil characteristics requires the solution of the heat equation over tube banks. The latter is an

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*Description of symbols is given in the nomenclature.

extremely complex equation and virtually impossible to solve except for trivial cases [7]. A non-dimensional analysis can be utilized, however, to determine the dimensionless groups that may influence the coil parameters C_1 and C_2. It is assumed that:

$$C_1 = \phi_1 \left(\frac{F_d}{S_t}, \frac{F_t}{D_h}, \frac{F_d}{S_\ell}, \frac{F_t}{F_s}, \frac{S_d}{F_d}, \frac{F_t}{F_h}, A_r \right) \tag{5}$$

and,

$$C_2 = \phi_2 \left(\frac{F_d}{S_t}, \frac{F_t}{D_h}, \frac{F_d}{S_\ell}, \frac{F_t}{F_s}, \frac{S_d}{F_d}, \frac{F_t}{F_h}, A_r \right) \tag{6}$$

In Eq 5 and 6, the longitudinal tube spacing S_ℓ, and the transverse tube spacing S_t are seen to influence the thermal characteristic performance of heat exchangers. This has been shown by the studies by Grame [8] and Jones [9]. On the other hand, fin spacing F_s, fin thickness F_t and fin height F_h also affect the performance of finned tube heat exchangers as reported by Briggs [10]. By investigating the heat transfer process through the primary and secondary surfaces, it can be shown that the ratio F_t/F_s is proportional to the ratio of heat transfer through the fin base to that through the bare tube surface. Also, the geometry of the fin is described by its aspect ratio F_t/F_h, and is proportional to the ratio of the conduction to convection heat transfer.

Heat transfer performance of tube banks is determined by the flow pattern, which is strongly dependent on the arrangement of the tubes [11]. The relative position of the tubes in the tube bank affect the turbulence produced in the flow field, and consequently influence the overall heat transfer performance of the heat exchanger. The dimensionless groups F_d/S_t and F_d/S_ℓ are considered to take account of the tube arrangement. For a continuous flat plate fin, F_d is the equivalent fin diameter based on an equal circular area of a non-circular fin [12]. This proved to be appropriate when determining the equivalent fin efficiency [13, 14 and 15]. Finally, the ratio of the minimum flow area to the frontal area of the coil A_r, is included as an additional parameter to represent flow characteristics.

Eq 5 and 6 are rearranged into the following workable form:

$$C_1 = a_0 \, A_r^{a_1} \left(\frac{F_d}{S_t} \right)^{a_2} \left(\frac{F_t}{D_h} \right)^{a_3} \left(\frac{F_d}{S_\ell} \right)^{a_4} \left(\frac{F_t}{F_s} \right)^{a_5} \left(\frac{S_d}{F_d} \right)^{a_6} \left(\frac{F_t}{F_h} \right)^{a_7} \tag{7}$$

and:

$$C_2 = b_0 \, A_r^{b_1} \left(\frac{F_d}{S_t} \right)^{b_2} \left(\frac{F_t}{D_h} \right)^{b_3} \left(\frac{F_d}{S_\ell} \right)^{b_4} \left(\frac{F_t}{F_s} \right)^{b_5} \left(\frac{S_d}{F_d} \right)^{b_6} \left(\frac{F_t}{F_h} \right)^{b_7} \tag{8}$$

Experimental heat transfer data of twenty different heat exchangers, as described in Appendix A, was used in conjunction with Eq 7 and 8 to obtain an expression for the quantities C_1 and C_2 in terms of the a's and b's. This was done numerically using step-by-step multiple linear regression analysis [16, 17]. With the aid of a digital computer, the regression analysis method was utilized to select the most significant dimensionless groups in Eq 7 and 8. The result is summarized in the following expressions for C_1 and C_2:

$$C_1 = 0.159 \left(\frac{F_t}{F_h} \right)^{0.141} \left(\frac{D_h}{F_t} \right)^{0.065} \tag{9}$$

and

$$C_2 = -0.323 \left(\frac{F_t}{F_h}\right)^{0.049} \left(\frac{F_s}{F_t}\right)^{0.077} \qquad (10)$$

For simplicity, the quantities C_1 and C_2 determined using Eq 9 and 10 will be referred to hereafter as $C_{1,a}$ and $C_{2,a}$ respectively. Agreement between C_1 and $C_{1,a}$ is very satisfactory, as shown in Fig. 1. Also, the comparison between C_2 and $C_{2,a}$ is given in Fig. 2 to show the good agreement between the experimental and analytical values. Further confirmation of this agreement is best illustrated in Fig. 3. In this graph, $J/C_{1,a}$ is plotted vs Re $^{C_{2,a}}$ for the set of coils used in this correlation. For each coil, four different points are plotted, and the agreement is very satisfactory. A summary of the data points of these coils is given in Table A-3 of Appendix A.

Because of the limited amount of experimental heat transfer data on different types of heat exchangers available in the literature, Eq 9 and 10 are restricted to the family of heat exchangers given in Appendix A (that is, tube banks of staggered circular tubes, four rows or more, with circular or continuous flat plate fins). Furthermore, these equations are only valid where the values of the dimensionless groups fall in the range of variables considered in the analysis, and the air Reynolds number is between about 200 and 2000. Table 1 gives the range of applicability of the correlation in terms of the dimensionless groups used in the regression analysis.

EXPERIMENTAL VERIFICATION

Experimental work was conducted using an 8-row staggered circular tube, continuous fin heat exchanger for the purpose of further verification of the correlation. Fig. 4 is a schematic diagram showing the pertinent dimensions of the tested coil. Description of this heat exchanger is given in Appendix A (coil No. 21 in Table A-1). A detailed description of the heat exchanger and of the test facility is given in Ref 18 and 19.

The resulting relationship between J-factor and Re is shown in Fig. 5. The quantities C_1 and C_2 were found to be 0.101 and -0.369 respectively as determined from the experimental results. On the other hand, $C_{1,a}$ and $C_{2,a}$ determined using Eq 9 and 10 were found to be 0.104 and -0.366 respectively. The J-Re relationship as determined analytically using the correlation is also plotted in Fig. 5 for comparison purposes.

DISCUSSION

For the air side Reynolds number between about 200 and 2000, Eq 9 and 10 are shown to be satisfactory in determining the coil parameters C_1 and C_2. Investigation of Eq 9 and 10 shows that the fin aspect ratio F_t/F_h has effects on both C_1 and C_2. An increase in the fin aspect ratio results in an increase in C_1 (and thus in J-factor) and a decrease in C_2 (and hence in the slope of J vs Re curve). The combined effect of increasing the fin aspect ratio is an increase in the average J-factor. This in fact is expected, since the fin efficiency is a monotonically increasing function in F_t/F_h [12], and an increase in the fin efficiency results in an increase in the overall heat transfer coefficient, and hence the average J-factor. The J-factor is also shown to increase with the ratio of hydraulic diameter D_h to fin thickness, F_t, due to the increased heat transfer area for convection to that for conduction. This suggests that heat exchangers of this type are being limited by the convective heat transfer area on the air side.

The experimental work done using an eight-row coil of the same family studied in this analysis showed that the experimentally determined coil parameters C_1 and C_2, of this coil are different by about 3% from those predicted analytically using Eq 9 and 10. Also, Fig. 5 shows that the experimentally determined J-factor is 3 to 5% lower than that predicted analytically over the range of air Reynolds number between about 200 and 2000. The set of coils selected for this analysis was chosen to represent those in common use in the HVAC system. Some experimental data in the literature is for coils tested outside the Reynolds number range considered in this analysis. These coils were not included so that the present results would be applicable only for the indicated range (200 < Re < 2000).

The method presented in this paper should not be considered as a final solution of prediction of heat exchanger performance. It does, however, present a relatively simple technique applied to at least one family. Perhaps further data may be included to generalize this method for

other families of heat exchangers. On the other hand, other less influential dimensionless groups may be identified using more data in the regression analysis.

CONCLUSION

In this paper, the dry surface heat transfer data is correlated for one family of finned tube heat exchangers. This is the multi-row, staggered circular tube arrangement with either circular or continuous flat plate fins. The characteristic performance of such heat exchangers was found to be correlated in the form:

$$J = C_1 \, Re^{C_2}$$

over the air side Reynolds number between about 200 and 2000.

The heat exchanger parameters C_1 and C_2, were found to be expressible in terms of the physical dimensions of the heat exchanger, namely fin thickness, spacing and height and coil hydraulic diameter.

Further verification of the proposed method of predicting the J-Re relationship was obtained by comparing the results of this model with new experimental heat transfer data obtained for an eight row coil of the same family.

Similar analysis could be made for other types of heat exchangers. Such studies are, however, limited by the amount of experimental heat transfer data available for that purpose.

NOMENCLATURE

Symbol	Definition	Units
a_0 through a_7	Constants in Eq 7	
A	Heat transfer area	m^2
A_r	Minimum flow area/coil face flow area	
b_0 through b_7	Constants in Eq 8	
c_p	Air specific heat at constant pressure	$J/(kg \cdot K)$
C_1 and C_2	Coil parameters given in Eq 4	
$C_{1,a}$ and $C_{2,a}$	Coil parameters as determined from Eq 9 and 10 respectively	
D_h	Coil hydraulic diameter	m
F_d, F_s, F_t	Fin diameter, spacing and thickness respectively	m
F_h	Fin height $\{F_h = 0.5(F_d - S_d)\}$	m
G	Maximum air mass flux	$kg/(m^2 \cdot s)$
h	Average film heat transfer coefficient	$W/(m^2 \cdot K)$
J	Overall heat transfer J-factor defined by Eq 3	
$\dot{m}$	Mass rate of flow	kg/s
N_f	Number of fins per in.	
Pr	Air side Prandtl number	
Re	Air side Reynolds number	

Symbol	Definition	Units
S_d, S_ℓ, S_t	Tube diameter, longitudinal spacing and transverse spacing respectively	m
U	Overall heat transfer coefficient	$W/(m^2 \cdot K)$
α, β	Constants in Eq 1	
ϕ_1, ϕ_2, ψ	Functions	

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TABLE 1

Dimensionless Groups Used
in the Regression Analysis

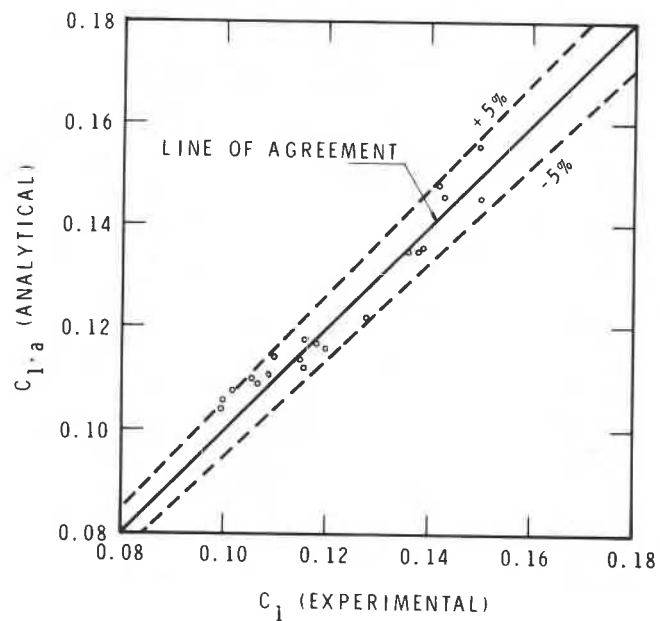|	$\dfrac{D_h}{F_t}$	$\dfrac{F_t}{F_h}$	$\dfrac{F_d}{S_\ell}$	$\dfrac{F_s}{F_t}$	$\dfrac{F_d}{S_t}$	$\dfrac{S_d}{F_d}$	A_r
	3.0	0.01	0.87	2.0	0.76	0.37	0.35
Range:							
	33.0	0.45	1.27	25.0	1.4	0.85	0.62

Fig. 1   *Comparison between C_1 and $C_{1'a}$*

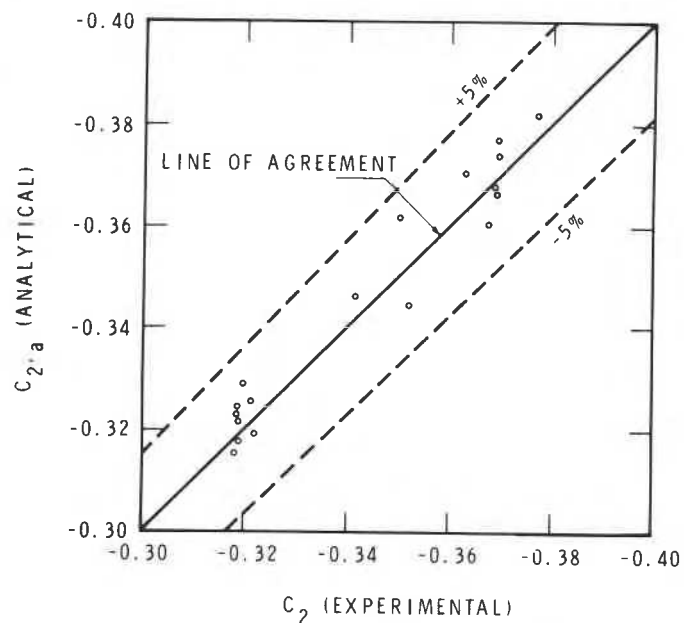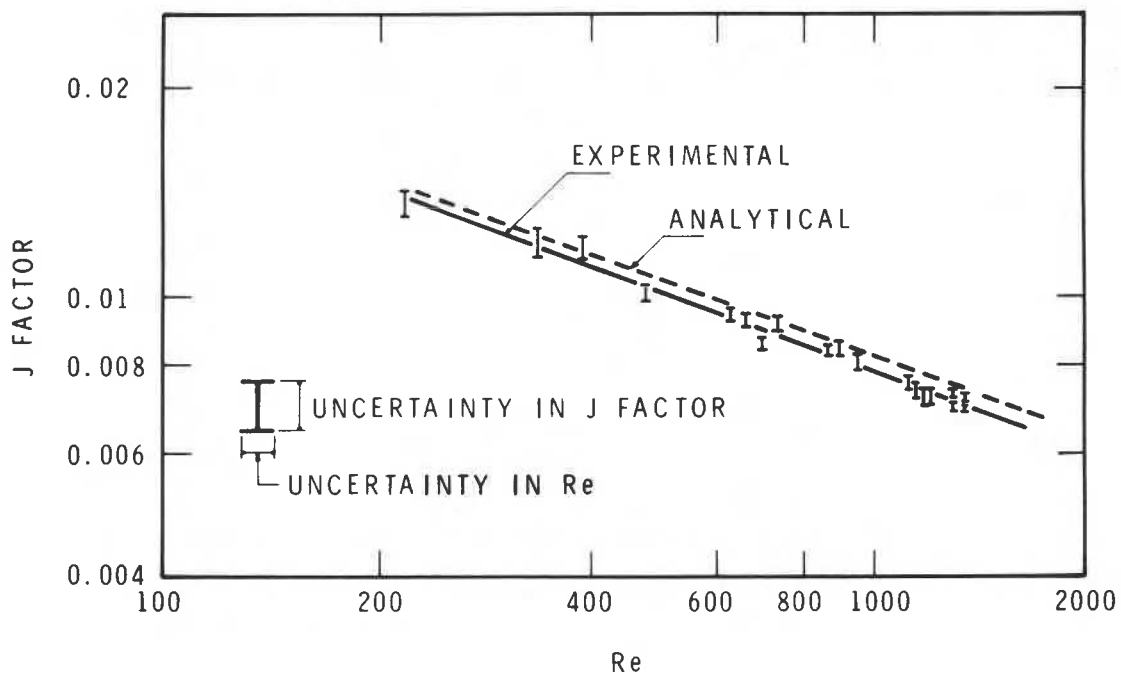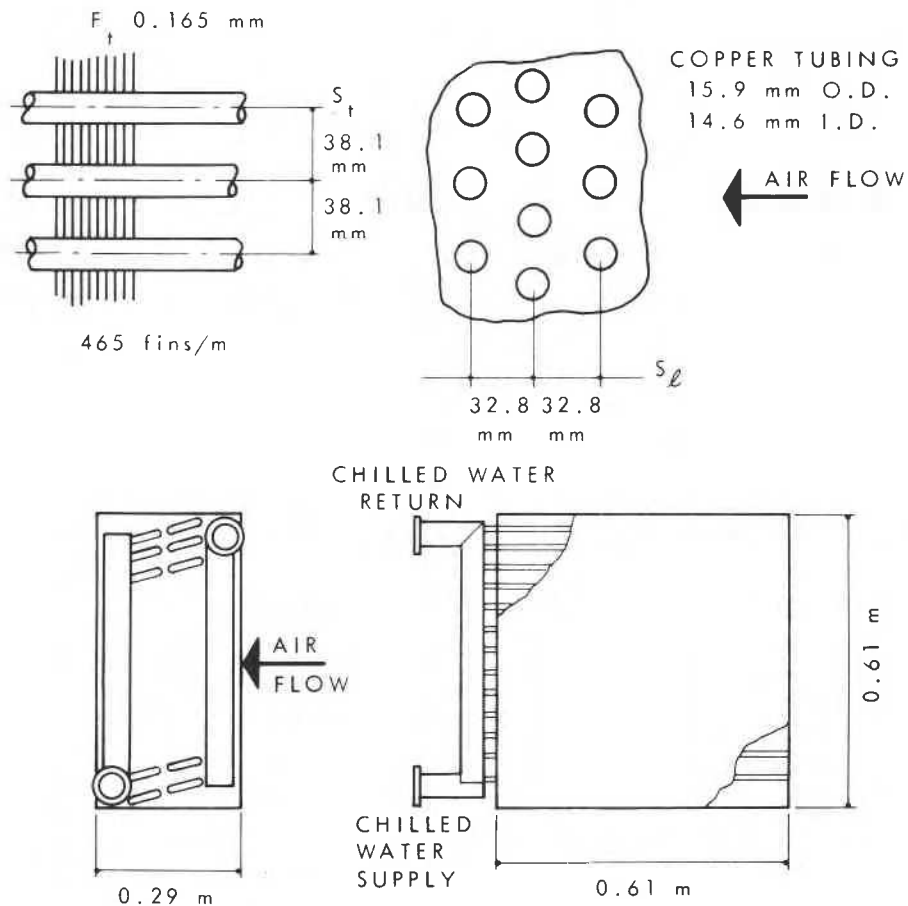

Fig. 2   *Comparison between C_2 and $C_{2'a}$*

Fig. 3   *Correlation of heat transfer data*

Fig. 4   Pertinent dimensions of the tested coil

Fig. 5   Dry surface characteristic curve
of the tested 8-row coil (18)

APPENDIX A

HEAT EXCHANGERS DATA FOR REGRESSION ANALYSIS

In this paper heat transfer data of 20 different heat exchangers were used in the regression analysis. An additional coil (coil No. 21) was used to carry out experimental tests for further confirmation of the analytical results. Table A-1 gives a summary of dimensions of all the coils included in this paper.

Tables A-2 and A-3 give more information about these heat exchangers including some points on their characteristic curves.

TABLE A-1

Coils Dimensions For Regression Analysis

Coil No.	S_d, in.	F_d, in.	N_f, fins/in.	D_h, in.	F_t, in.	A_r —	S_t, in.	S_ℓ, in.
1	0.625	0.737	19.5	0.187	0.016	0.352	0.964	0.838
2	0.546	0.762	11.3	0.209	0.017	0.434	0.964	0.835
3	0.554	0.894	11.2	0.186	0.018	0.487	1.08	0.935
4	0.438	0.895	7.2	0.293	0.022	0.59	1.08	0.935
5	0.549	0.89	8.0	0.22	0.042	0.49	1.08	0.935
6	0.535	0.89	6.25	0.24	0.08	0.50	1.08	0.935
7	1.15	2.3	8.0	0.24	0.017	0.53	2.45	2.13
8	0.93	1.97	9.82	0.203	0.021	0.59	2.25	1.95
9	0.95	2.23	9.65	0.181	0.022	0.613	2.45	2.13
10	1.16	2.75	7.9	0.188	0.018	0.59	2.8	2.44
11	1.1	2.02	9.02	0.196	0.017	0.5	2.19	1.9
12	1.03	2.0	10.1	0.184	0.013	0.53	2.19	1.9
13	0.53	1.32	6.67	0.183	0.006	0.56	1.25	1.08
14	0.53	1.32	7.67	0.157	0.006	0.56	1.25	1.08
15	0.53	1.32	9.17	0.133	0.006	0.55	1.25	1.08
16	0.53	1.32	11.7	0.11	0.006	0.54	1.25	1.08
17	0.53	1.32	14.5	0.08	0.006	0.54	1.25	1.08
18	0.4	1.05	8.0	0.14	0.013	0.53	1.0	0.87
19	0.68	1.83	7.75	0.14	0.016	0.50	1.5	1.75
20	0.39	1.04	8.1	0.15	0.006	0.58	1.0	0.87
21	0.63	1.62	11.8	0.11	0.007	0.54	1.5	1.29

TABLE A-2

Summary Of Heat Exchangers Data

Coil	Ref. No.	No. of Rows	C_1	C_2	Type of Fin	Fin Material
1	10	6	0.15	-0.319	Circular	Cu
2	10	6	0.15	-0.318	"	Cu
3	10	6	0.138	-0.319	"	Cu
4	10	6	0.136	-0.32	"	Aℓ
5	10	6	0.143	-0.319	"	Cu
6	10	6	0.142	-0.319	"	Cu
7	10	6	0.11	-0.319	"	Aℓ
8	10	6	0.116	-0.319	"	Aℓ
9	10	6	0.11	-0.319	"	Aℓ
10	10	6	0.116	-0.319	"	Aℓ
11	10	6	0.118	-0.32	"	Aℓ
12	10	6	0.115	-0.32	"	Aℓ
13	A-1	4	0.106	-0.377	Flat Plate	Cu
14	A-1	4	0.107	-0.37	"	Cu
15	A-1	4	0.102	-0.351	"	Cu
16	A-1	4	0.10	-0.369	"	Cu
17	A-1	4	0.1	-0.368	"	Cu
18	A-2	-	0.128	-0.363	"	Cu
19	A-2	-	0.12	-0.381	"	Cu
20	A-3	6	0.12	-0.37	"	Cu
21	18	8	0.101	-0.369	"	Aℓ

TABLE A-3

Coils Heat Transfer J-Factor

Coil No.	Reynolds Number			
	300	600	900	1200
1	0.0244	0.0209	0.0172	0.0157
2	0.024	0.0195	0.0171	0.0156
3	0.0224	0.0179	0.0158	0.0144
4	0.022	0.0176	0.0155	0.0141
5	0.0231	0.0186	0.0163	0.0149
6	0.023	0.0184	0.0162	0.0148
7	0.0178	0.0143	0.0126	0.0115
8	0.0188	0.015	0.0132	0.0121
9	0.018	0.0144	0.0127	0.0116
10	0.0188	0.0151	0.0132	0.0121
11	0.0189	0.0152	0.0133	0.0122
12	0.0186	0.0149	0.0131	0.0119
13	0.0124	0.0095	0.0082	0.0073
14	0.013	0.010	0.0086	0.0078
15	0.0137	0.0108	0.0093	0.0084
16	0.0122	0.0094	0.0081	0.0073
17	0.0123	0.0095	0.0082	0.0074
18	0.0161	0.0125	0.0108	0.0098
19	0.014	0.011	0.009	0.0081
20	0.0145	0.0112	0.0096	0.0087
21	0.0123	0.0095	0.0082	0.0074

REFERENCES: APPENDIX A TABLES

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