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Study on Air Cooled Heat Exchanger with Corrugated Fin Surfaces*
(2nd Report. Characteristics of Corrugated Straight Fin Surfaces)

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Air cooled heat exchangers with corrugated fin surfaces are now in mass production for automotive and air conditioning applications. In these heat exchangers, the fin length is short relative to the equivalent diameter, and particularly, in heat exchangers with corrugated straight fins, heat transfer and flow friction characteristics are governed mainly by the entrance flow region. Most design data so far reported are, however, concerned with a fully developed flow rather than an entrance flow.

In this paper, basic heat transfer and flow friction characteristics are presented on corrugated straight fin surfaces with the ratio of fin length to equivalent diameter, less than 26.5, by employing the steady state and transient techniques. These data were rearranged by the aid of analysis on the entrance region between parallel flat plates, and the result expressed in a correlation in the form, $Nu = \phi[R_e, Pr, (d_e/L), R]$, $f = \psi[R_e, (d_e/L)]$, applicable to the optimum design of air cooled heat exchangers with corrugated straight fin surfaces.

Nomenclature

1. Introduction

The performance of Air Cooled Heat Exchangers with corrugated fin surfaces depends on its fin characteristics. For this reason, corrugated louverd fins of good heat transfer coefficients have been the main stay of Air Cooled Heat Exchangers for automobiles. But, as they also have a large pressure drop, the driving power and noise of the cooling fan are liable to be large. On the other hand, the corrugated straight fins have a smaller heat transfer coefficient than the corrugated louverd fins, but since they have a small pressure drop, the countermeasure against noise is easy. That is the reason why they have come to be appreciated recently. However, as reports on fin characteristics of small L/d_e corrugated straight fins of Air Cooled Heat Exchangers for automobiles, houses, etc. are very few, it is very difficult to design compact Air Cooled Heat Exchangers with corrugated straight fins combined with low-noise fans.

This report is concerned with the fin characteristics of small L/d_e corrugated straight fins of the Air Cooled Heat Exchangers for automobiles, houses, etc. Fin characteristics of high accuracy were measured on a large number of fin cores with different fin elements in practical use range of the Air Cooled Heat Exchanger for automobiles, then they were united into one experimental equation by analysis of flow in the entrance region.

A_t : air side water tube surface area
 A_f : fin surface area
 a : cross-sectional area for flow passage of a fin spacing
 d_e : equivalent diameter for flow passage of a fin spacing
 f : Fanning's friction factor
 h_a : apparent average heat transfer coefficient of the fin
 h_f : actual average heat transfer coefficient of the fin
 L_y, L : flow length of the fin
 L_f : fin length from root to center
 L_e : inlet length of a fin spacing
 L_t : braze length of fin and water tube wall
 P_f : fin pitch
 S : metal thickness
 t_a : air temperature
 T_b : metal temperature of water tube wall
 U : air flow velocity in boundary layer
 U_c : air flow velocity at center of flow passage
 U_a : air flow velocity in front of flow passage
 U_m : mean air flow velocity for flow passage
 U_w : mean water flow velocity in water tube
 x : distance from inlet of flow passage
 χ : nondimensional distance
 $= x/(P_f/2)$
 Ω : overall heat transfer efficiency of air cooled heat exchanger with corrugated fin surfaces
 ξ_c : fin efficiency in case of two-dimensional heat conduction through thin metal fins
 $= \tanh \frac{L_f \sqrt{(L_y/L_t) (2h_f/\lambda_f S_f)}}{L_f \sqrt{(L_y/L_t) (2h_f/\lambda_f S_f)}}$
 η : heat transfer efficiency of water tube wall
 $= \tanh \frac{P_f \sqrt{(h_w/S_b \lambda_b) (1+h_f/h_w)}}{P_f \sqrt{(h_w/S_b \lambda_b) (1+h_f/h_w)}}$

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λ : thermal conductivity
 ν : coefficient of viscosity
 δ : thickness of boundary layer

Subscripts

o : outlet
 b : water tube wall
 i : inlet
 f : fin
 m : average

2. Characteristics of Corrugated Straight Fins

2.1. Test fin core

As reports on the fin characteristics of corrugated straight fin core of relatively small L/d_e and of large (fin width/fin pitch) are very few, (1),(2) 11 kinds of fin cores, fin length $L = 24-83$ mm, fin pitch $2P_f = 3.1-5.7$ mm, $L/d_e = 7.7-26.5$, were manufactured, in a practical size range of Air Cooled Heat Exchanger for automobiles. Table 1 shows the detailed dimensions of test fin cores. Tubeless cores, which consist of corrugated straight fins joined by adhesive cement with and between plates of the same thickness in piles are SF-1 ~ SF-3. On the other hand, SFT-1-8 are test fin cores with flat water tubes composed of corrugated straight fins soldered alternately with several flat water tubes. In manufacturing the test fin cores, as the uniformity of the shape of corrugated fin's flow passage affects fin characteristics, much care was taken to make the shape uniform but an error of around 3% was inevitable. The shape of the corrugated fin's flow passage is as shown in Fig. 1 an intermediate shape between a triangular and a rectangular passage, and it depends a little on the fin pitch.

2.2. Measuring method of the average heat transfer coefficients

The average heat transfer coefficient h_f was measured on tubeless test fin cores by single blow transient test technique and on test fin cores with flat water tubes by steady state test method.

Single blow transient test technique is known as the method to know the average heat transfer coefficient of fins from the change of the core exit air temperature by giving stepwise change to the inflow air temperature into the test fin core, and it was evolved from maximum slope method of Locke. (3) As Howard (13) points out, however, the maximum slope method has a defect that errors suddenly become large below NTU 3.5. These test fin cores are all below 3.5 in NTU, and as they have so small L/d_e that the change of the exit air temperature will not be S curve, maximum slope method may not be useful. Also, as the heat capacity of the test fin core is small, the change of core exit air temperature is so rapid that the conventional method is not applicable. Therefore, improvement was made appropriately on the

test fin cores in this study, and then an improved curve matching method was developed.

Fig. 2 is its flow chart, and it is a method to calculate by an electronic computer the temperature change inside the

Table 1: Dimensions of the test fin core

No.	L mm	$2P_f$ mm	d_e mm	L/d_e	$2L_f$ mm	S_f mm	$2L_f/P_f$	
SF-1	24	3.8	3.10	7.74	10	0.08Cu	5.26	80 × 100
SF-2	28	3.9	3.11	9.03	10	0.08Cu	5.14	80 × 100
SF-3	32	3.8	3.10	10.32	10	0.08Cu	5.26	80 × 100
SFT-1	26	4.0	3.13	8.31	10	0.05Cu	5.00	200 × 200
SFT-2	32	4.0	3.13	10.22	10	0.05Cu	5.00	"
SFT-3	49	4.0	3.13	15.65	10	0.05Cu	5.00	200 × 200
SFT-4	66	4.0	3.13	21.09	10	0.05Cu	5.00	200 × 200
SFT-5	83	4.0	3.13	26.52	10	0.05Cu	5.00	200 × 200
SFT-6	32	3.1	2.43	13.17	10	0.05Cu	6.45	200 × 200
SFT-7	32	5.0	3.76	8.51	10	0.05Cu	4.00	"
SFT-8	32	5.7	4.15	7.17	10	0.05Cu	3.50	"

SFT-2, 6, 7, 8

SFT-5

Fig. 1: Air flow passage of the corrugated fin core

test fin core in accordance with the step-change at just the same airflow temperature as the experiment, and to correct the presumed average heat transfer coefficient and the average air velocity in the test fin core until the core exit air temperature change corresponds to the actually measured air temperature change. Here, as the correction mode was determined by a few trials to quicken convergence, the results were quickly gained by the mode correction of the average heat transfer coefficient and the average air velocity.

Fig. 3 is a rough sketch of a single blow transient test measuring apparatus. The test fin core is mounted on the part of the duct where the air velocity is uniform on the suction side of the blower, and then it is heated approximately 10°C higher than the room temperature. The electric heater, a row of sheath heaters of 0.027ϕ lined in 2 mm pitch like hurdles, is placed approx. 30 mm upstream of the test fin core, and it can be drawn out instantaneously from the air passage by the electro-magnetic valve and the air cylinder. Also, the measurement of the core exit air temperature change has a problem especially in its response as heat capacity of the core is small. Therefore, here the average air temperature change in the fin core exit is recorded by joining a 0.027ϕ chromel-alumel thermocouple the heat capacity of which at its junction may be as small as possible and by arranging 21 pieces in mesh in series connection in the test fin core downstream. The time constant of the thermocouples was 0.09 sec at the air velocity of 1 m/s and 0.03 sec at 20 m/s.

The following Fig. 4 shows the core characteristics of a measuring apparatus for the fin cores with flat water tubes by steady state test method. The test fin core with flat water tubes is mounted on the part of the duct where the air velocity is uniform on the suction side of the blower, then the difference of the average air temperature in front of and at the back of the core was measured by arranging 21 pieces of 0.2ϕ copper-constantan thermocouples in mesh in series connection at a place approx. 50 mm in front of and at the back of the core, and the air side heat transfer rate was measured. Hot water is run through the flat water tubes, and under the constant air side condition, overall heat transfer coefficient in various quantities of hot water flow;

$$K[=Q/(A_t + A_f)(t_{wm} - t_{tm})]$$

is measured. From K_o of $1/U_w^{0.8} \rightarrow 0$ in the relation figure of $1/U_w^{0.8}$ and K , the apparent heat transfer coefficient h_a including the overall heat transfer efficiency Ω of the fin is obtained (Wilson plat method).⁽⁴⁾ Saturated steam is flowed into the tube, and from the average tube wall temperature T_{bm} (in this study measurements were made by fixing the 0.04ϕ copper-constantan thermocouple at several places of the tube wall) and by

$$h_a = \frac{Q}{(A_t + A_f)(T_{bm} - t_{am})} \dots\dots\dots (1)$$

Fig. 2: Flow chart of the single blow transient test technique

Fig. 3: Rough sketch of the single blow transient test apparatus

Fig. 4: Rough sketch of the steady state measuring apparatus

h_a is found (steam-to-air method). All h_a values were measured by both methods on the test core with flat water tubes. In case of hot water, however, for the quantity of heat measured on the air side and the hot water side, the data within 5 % alone were employed.

The average apparent heat transfer coefficient on the air side measured by the steady state test method becomes as follows

$$h_a = \frac{A_t + Q A_f}{A_t + A_f} h_f \quad (2)$$

Here, A_t and A_f are the heat transfer areas on the air side of the water tube and the fin.

In discussing the heat transfer coefficients of the fin, it is necessary to depend on the actual average heat transfer coefficient h_f . For that reason, it is necessary to get h_f which satisfies the overall heat transfer efficiency introduced in the 1st report⁽¹⁵⁾

$$Q = \zeta_c \frac{1 + \frac{h_f}{h_w} \left(1 + \frac{A_f}{A_t}\right)}{1 + \frac{h_f}{h_w} \left(1 + \frac{A_f}{A_t} \zeta_c\right)} \quad (3)$$

and eq. (2), by using repetitive method.

2.3. Actual average heat transfer coefficient and flow friction factor

The actual average heat transfer coefficient by single blow transient test technique was defined by the logarithmic average temperature difference, and the one by steady state test method was defined by the mixed average temperature difference. But as the one by steady state test method has been measured at $(T_{bo} - t_{ao}) / (T_{bi} - t_{ai}) > 0.7$, there is almost no difference from the one defined by the logarithmic average temperature difference.

If the measurement results are expressed by the nondimensional number of usual correlation, that is, the equivalent diameter of the air flow passage $d_e = 4aL / (A_t + A_f)$, and the relation between Re number by the average air flow velocity U_m and $J = (h_f / g \rho U_m C_p) \times P_r^{2/3}$, the core without water tube is expressed as in Fig. 5 and the core with flat water tube as in Fig. 6

If the average air flow velocity is the same, the smaller the fin pitch, or the shorter the fin length, the larger becomes the average heat transfer coefficient. This suggests that because of the test fin core having small L/d_e , a heat transfer has been caused by the entrance flow.

On the other hand, the pressure drop is shown in Figs. 7 and 8 which indicate the relation between the friction factor by Fanning $f = (d_e / 2 \rho U_m^2) (\Delta P / L)$, employed for expressing the core characteristics of the conventional air cooled heat exchanger, and Re number, after the measurement of pressure drop ΔP at the time when both the test fin core without water tube and the test fin core with flat water tube are not

heated employing the single blow transient test apparatus and steady state test apparatus. As in the case of the average heat transfer coefficient, the curves become different depending on the core thickness and the fin pitch. Then, it was tried to indicate the characteristics of these fins by one experimental equation.

Fig. 5: Heat transfer coefficients of the test fin core without water tube

Fig. 6: Heat transfer coefficients of the test fin core with flat water tube

Fig. 7: Friction factors of the test fin core without water tube

3. Analysis of Entrance Flow

The shape of the air flow passage of the corrugated straight fin, employed for automobiles and air cooled heat exchangers of relatively small L/d_e is neither an triangle nor a rectangle, but an intermediate shape of the above two, due to the aspect ratio being more than 3.5, and to the greater part of the air flow passage being close to the flow between the parallel flat plates.

Flow pattern of the parallel flat plates three times (114×50) as large as the test core's fin (Table 1), was observed by Schlieren method. Fig. 9 indicates the above which consists of the case that the entrance flow passes and the case that there are the entrance flow and a fully developed flow following the entrance flow. Measuring the distribution of the air flow velocities in the core exit by a hot-wire at the time the test fin core was not heated, it was made sure that the air flow in the test fin core was similar to the flow between the parallel flat plates so that analysis of the entrance flow between the parallel flat plates was made.

3.1. Heat transfer coefficient in entrance region

There are theoretical analyses by Sparrow,⁽⁵⁾ Stephan,⁽⁶⁾ Hwang⁽⁷⁾ and Mercer et al.⁽⁸⁾ on the heat transfer in the entrance region of the parallel flat plate, and the results of Mercer et al. are so trustworthy that they are transformed and applied to this study.

The numerical solutions are listed by the calculation model of Fig. 10 in the case of constant wall temperature in the relation of $(x/P_f/2)(\nu/2P_f U_a)(1/P_r)$ and the average nusselt number $P_f h_f/\lambda$; transforming this as

$$\left. \begin{aligned} d_e &= 4 \frac{2L_f P_f}{2(2L_f + P_f)}, & R &= \frac{2L_f}{2L_f + P_f} \\ R_e &= \frac{u_m d_e}{\nu}, & N_u &= \frac{h_f d_e}{\lambda} \end{aligned} \right\} \dots (4)$$

it will be the relation of $P_r R_e (d_e/L)$ ($1/4R^2$) and $N_u/2R$ applicable to the air flow passage (Fig. 11).

3.2. Friction factor in entrance region

Reports on the analysis of the entrance flow of the rectangle tube or the parallel flat plates are made by Han,⁽⁹⁾ Schlichting,⁽¹¹⁾ Sparrow,⁽¹⁰⁾ Collins-Schowalter⁽¹⁴⁾ and Bodia-Osterle.⁽¹²⁾ Now, if the length of the entrance region L_e is set in such range that the center air velocity of the air flow passage reaches 98-99 % of the Poisuille flow, there are considerable differences such as Schlichting, $L_e = 0.16 U_m (P_f/2)^2/\nu$, Collins-Schowalter, $L_e = 0.136 U_m (P_f/2)^2/\nu$, Sparrow, $L_e = 0.104 U_m (P_f/2)^2/\nu$.

As there was nothing in these reports which is applicable for this study, the

Fig. 8: Friction factors of the test fin core with flat water tube

(a) entrance flow + developed flow

(b) entrance flow

Fig. 9: Flow pattern in the parallel flat plates (by Schlieren method)

Fig. 10: Calculation model of entrance region in the parallel flat plates

Fig. 11: Calculation model of air flow passage

length of the entrance region of Collins-Schowalter's according to the solution of Bodia-Osterle was utilized, and the friction factor was gained in the entrance region of the parallel flat plate by the profile method of a three order velocity distribution, and was transformed to the air flow passage of the test core with large aspect ratio.

In Fig. 10, assuming that the center air velocity is U_c and the pressure P , in the center of the entrance region between the flat plates,

$$-\frac{dP}{dx} = \rho u_c \frac{du_c}{dx}$$

is formed, and obtained as

$$\Delta P = \frac{1}{2} \rho u_a^2 \left[\left(\frac{u_c}{u_a} \right)^2 - 1 \right] \quad (5)$$

from the above equation. Therefore, if the change in the flow direction of the center air flow velocity U_c/U_a in the flow passage is known, ΔP will be gained, and from

$$f = \frac{\Delta P}{\frac{4L}{d} \frac{1}{2} \rho u_m^2} \quad (6)$$

the friction factor f can be known.

From boundary condition of $U = v = 0$, $\rho U_c \partial U_c / \partial x + \mu (\partial^2 U / \partial y^2)_{y=0} = 0$ at $y = 0$, and $U = U_c$, $\partial U / \partial y = 0$ at $y = \delta$, the following velocity distribution's equation is set.

$$\frac{u}{u_c} = \frac{3}{2} \xi - \frac{1}{2} \xi^3 + \Lambda \left(\frac{1}{4} \xi - \frac{1}{2} \xi^2 + \frac{1}{4} \xi^3 \right) \quad (7)$$

Here, $\xi = y/\delta$, $\Lambda = (\delta^2/\nu) (dU_c/dx)$.

The momentum equation in the case that the air velocity in the main changes to the x direction is well known as below,

$$\frac{d}{dx} \int_0^\delta (u_c - u) u dy + \frac{du_c}{dx} \int_0^\delta (u_c - u) dy = \nu \left(\frac{\partial u}{\partial y} \right)_{y=0} \quad (8)$$

and setting the displacement thickness as δ^* , from the continuous condition,

$$\frac{P_f}{2} u_a = \left(\frac{P_f}{2} - \delta^* \right) u_c \quad (9)$$

$$\delta^* = \int_0^1 \delta \left(1 - \frac{u}{u_c} \right) d\xi \quad (10)$$

are formed.

The first approximate solution In eq. (7), the center flow velocity U_c and the boundary layer thickness δ are the functions of x , but if eq. (7) as the first approximation is not influenced by the change of the central velocity, that is if $U/U_c = 3\xi/2 - \xi^3/2$ is substituted in eq. (10), $\delta^* = 3\delta/8$ will be obtained and from eq. (9), will be obtained,

$$\frac{u_c}{u_a} = \frac{1}{1 - (3/8)\chi} \quad (11)$$

Here, $\chi = \delta/(P_f/2)$. If the above equation is closely approximated by the quadratic equation of χ , it will be in 0.1% accuracy

$$\frac{u_c}{u_a} = 1 + 0.3569\chi + 0.2431\chi^2 \quad (12)$$

and substitution of it in eq. (8) yields

$$\chi(1 + 1.0709\chi + 1.2150\chi^2) d\chi = 21.5393 \frac{1}{R_{ef}} dX \quad (13)$$

Here, $R_{ef} = P_f U_a / \nu$, $X = x/(P_f/2)$.

On the other hand, from eqs. (12), (13) the following will be obtained.

$$\Lambda = \frac{\delta^2 du_c}{\nu dx} = \chi(4.0840 - 1.0419\chi - 0.2787\chi^2) \quad (14)$$

The second approximate solution Setting eq. (14) as the first approximation, the case that the velocity distribution in boundary layer is influenced by the center velocity will be gained. From eqs. (7), (9), (10) and (14), the following will be obtained:

$$\frac{u_c}{u_a} = 1 + 0.3671\chi + 0.0980\chi^2 \quad (15)$$

Substituting eqs. (7), (14) and (15) in eq. (8), will yield

$$\chi(0.0947 + 0.1127\chi + 0.0426\chi^2) d\chi = \frac{1}{R_{ef}} dX \quad (16)$$

and from eqs. (15) and (16), will be obtained,

$$\Lambda = \chi(3.8380 - 2.2472\chi + 0.6611\chi^2) \quad (17)$$

The third approximate solution From eqs. (7), (9) and (17) as the third approximation of the center velocity, the following is gained.

$$\frac{u_c}{u_a} = 1 + 0.3640\chi + 0.1243\chi^2 \quad (18)$$

In the fully developed flow after the entrance region it is already well known that $U_c/U_a = 1.5$, and setting $\chi = 1$, $U_c/U_a = 1.6$ in the first approximate solution (11), $U_c/U_a = 1.47$ in the second approximate solution (15) and $U_c/U_a = 1.49$ in the third approximate solution (18), so that they seem to be almost converged.

If eqs. (7), (17) and (18) are substituted in eq. (8), the following will be obtained.

$$\chi(0.09646 + 0.08750\chi - 0.01003\chi^2) d\chi = \frac{1}{R_{ef}} dX \quad (19)$$

and if the solution is gained as $X = 0$ and $\chi = 0$, the following will be obtained.

$$0.25\chi^2(0.04823 + 0.02917\chi - 0.00251\chi^2) = \frac{1}{R_{ef} P_f} x \quad (20)$$

Therefore, from eqs. (18) and (20),

$$\frac{u_c}{u_a} = 1 + 3.445 \left(\frac{1}{R_{ef} P_f} x \right)^{0.5} \quad (21)$$

can be gained. Eq. (20), if $P_f = \infty$ is assumed, will agree to the solution of the profile method by the third order velocity distribution of the flat plate parallel to the flow,

$$\delta = 4.6\sqrt{\nu x/u_a}$$

Substituting eq. (21) in eq. (5), the pressure drop in the entrance region of the parallel flat plate will be

$$\Delta P = \frac{1}{2} \rho u_a^2 \left[\left\{ 1 + 3.445 \sqrt{\frac{1}{R_e} \frac{x}{L}} \right\}^2 - 1 \right] \quad \dots (22)$$

Substituting the above equation in eq. (6), and using eq. (4),

$$f \frac{L}{d_e} = \frac{1}{4} \left\{ \left(1 + \frac{3.445}{\sqrt{R_e} \frac{d_e}{L}} \right)^2 - 1 \right\} \quad \dots (23)$$

will be gained. If the entrance length $L_e = 0.136 [U_a (P_f/2)^2 / \nu]$ is adopted, the applicable limit of eq. (23) will be:

$$R_e \frac{d_e}{L} \geq 7.5 \quad (R_e \leq 2300) \quad \dots (24)$$

In Fig. 12 the friction factor in the entrance region of the rectangle tube by eq. (23) and the theoretical value in the entrance region of a round tube of Langhaar are compared. In the figure, the fully developed flows of the rectangle tube including the entrance region are the theoretical values of Sparrow and others.

4. Nondimensional Arrangements and Experimental Equation

The nondimensional numbers in the previous chapter were applied to the rearrangement of the actual measured values of Figs. 5, 6, 7 and 8 in order to combine to a single curve each average heat transfer coefficient and friction factor of the corrugated straight fin core.

4.1. Experimental equation of average heat transfer coefficients

The actually measured average heat transfer coefficient of the test fin core is shown in Fig. 13, which was introduced in the relation between $P_r R_e (d_e/L) (1/4R^2)$ and $N_u/2R$ mentioned in the preceding chapter, and which has converged to a curve with approx. 10 % deviation. In this figure, ones with $2L_f/P_f$ more than 3.5 and $L/d_e = 16$ among the reports on the characteristics of the corrugated straight fin of the isosceles triangle flow passage by London⁽¹⁾ and others shown in Table 2, and ones with $2L_f/P_f \geq 3.5$ in Table 3 of $L/d_e < 30$ of Kays and London's⁽²⁾ corrugated straight fin are plotted because they are comparable with the test fin core of this study. Both well agree on the curve.

The smaller the value on the abscissa of the modified Graetz number, the smaller becomes the heat transfer coefficient of

the test fin core than the Mercer and the others' theoretical values between the parallel flat plates, seemingly under influence of the part of the fully developed flow.

From Fig. 13, within the practical range of the corrugated straight fin core in this study, that is, under the condition of $L/d_e \leq 26.5$, and $2L_f/P_f \geq 3.5$, with the shape of the air flow passage being a triangle, a rectangle and an intermediate one of both, the average heat transfer coefficient of $R_e \leq 2300$ can be denoted as below:

$$N_u = 2R \left[1.10 + 0.555 \left(P_r R_e \frac{d_e}{L} \frac{1}{4R^2} \right)^{0.55} \right] \quad \dots (25)$$

$$N_u = \frac{h_f d_e}{\lambda}, \quad R_e = \frac{u_m d_e}{\nu}$$

4.2. Experimental equation of friction factors

As in the case of the heat transfer coefficient, when nondimensional numbers in the entrance region of the air flow

Fig. 12: Friction factors of entrance flow

Fig. 13: Analogous nondimensional arrangement and experimental equation of the average heat transfer coefficients

Table 2: Dimensions of the corrugated straight fin cores of the isosceles triangle flow passage (by London and others)

(London (13))

No.	L mm	P _f mm	d _e mm	L/d _e	2L _f mm	S _f mm	2L _f /P _f
30.33T	63.5	0.84	1.22	52.0	4.32	0.102A1	5.08
10.27T	63.5	2.47	3.84	16.6	13.82	0.254A1	5.52
11.94T	127	2.13	2.87	44.4	6.33	0.152A1	2.94
46.45T	66.8	0.55	0.805	83.0	2.54	0.051A1	9.24

Table 3: Dimensions of the corrugated straight fin cores (by Kays and London)

(Kays-London (1)(9))

No.	L mm	2P _f mm	d _e mm	L/d _e	2L _f mm	S _f mm	2L _f /P _f
5.3	63.3	9.59	6.15	10.3	11.94	0.152	2.48
11.1	63.5	4.57	3.08	20.6	6.35	0.152	2.72
14.7	63.9	3.44	2.59	24.7	8.38	0.152	4.88

Fig. 14: Analogous nondimensional arrangement and experimental equation of the friction factors (in case of the test fin core without water tube)

Fig. 15: Analogous nondimensional arrangements and experimental equation of the friction factors (in case of test fin core with flat water tube)

passage are applied to the rearrangement of the friction factors of the test fin core ($Re \leq 2300$), each of them converges to a single curve, in the case of the test fin core without water tube as seen in Fig. 14, and in the case of the test fin core with water tube as seen in Fig. 15.

In Fig. 14, the friction factors of the cores (Table 2, 3) of $2L_f/P_f \geq 3.5$ by London and others (1), (2) are plotted, and they can be closely approximated with about 10 % deviation as follows;

$$f \frac{L}{d_e} = 0.05 + 15.5 \left(R_e \frac{d_e}{L} \right)^{-0.93} \quad \dots (26)$$

The friction factor of the core (Table 2) by London and others the most part of which is a fully developed flow ($83.0 \geq L/d_e \geq 44.4$) was plotted for reference, which resulted in a good agreement with the above equation. This suggests that although data are more or less imperfect, eq. (26) can be applied when a considerable part of the fully developed flows of $Re d_e/L \leq 10$ exists.

On the other hand, as the friction factor of the test fin core with flat water tube includes the pressure drop of the flat water tube, it is a little larger than eq. (26), and it can be closely approximated with about 10 % deviation as below;

$$f \frac{L}{d_e} = 22.95 \left(R_e \frac{d_e}{L} \right)^{-0.91} \quad \dots (27)$$

provided that $Re d_e/L < 300$. As for $Re d_e/L \geq 300$, due to a very small proportion the pressure drop of the water tube occupies compared with the pressure drop between the fins, it comes closer to the value of eq. (26) without water tube.

5. Conclusions

The heat exchanger with corrugated straight fins for automobiles and housing, of which L/d_e is relatively small and $2L_f/P_f$ is large, is practically used. As there are a few reports on the fin cores of this kind, however, the study on the fin core of $L = 24\text{--}83\text{ mm}$, $2P_f = 3.1\text{--}5.7\text{ mm}$, $L/d_e = 7.7\text{--}26.5$ was made experimentally, and by the application of the result of the analysis of the entrance flow, the data were gathered on a single curve regardless of the fin pitch and the fin length, then the experimental equations were gained concerning the average heat transfer coefficient and the friction factor. From this it can be believed that it is possible to design a good heat exchanger with corrugated straight fins.

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