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Experiment of Heat Transfer on the Surfaces with Transverse Fins for Flow Direction*

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The heat transfer and the pressure drop are experimentally studied on the extended heating surfaces with transverse fins for flow direction placed in the air channel of rectangular cross section. From the results of experiment on the effects of fin whose pitch and height are p and l respectively, channel height and Reynolds number, empirical equations for the heat transfer coefficient and the friction factor are presented.

It has been shown that the ratio of fin pitch to fin height p/l is a most important factor for the heat transfer and pressure drop performance. In the case of the value of $p/l=1.2\sim1.6$, the finned surface shows the most effective performance. The flow patterns between fins of the surface are observed and the effect of fin materials is also studied experimentally.

1. Introduction

In conventional heat exchangers and fuel elements of a gas-cooled reactor, the heat transfer performance has been improved by attaching fins to the heating surfaces at the gas side. Since the extended surfaces show an increase of resistance simultaneously with an increase of heat transfer by the fins, both the performances of heat transfer and flow resistance have to be considered in application of the extended surfaces. The extended surfaces used in general are ones with longitudinal fins for flow direction and many investigations on these extended surfaces have been performed. Recently various types of extended surface in connection with reactors have also been studied⁽¹⁾⁽²⁾. But in the case of transverse fins for flow direction, there are only a few studies such as the ones carried out by Knudsen and Katz⁽³⁾ using water and by Fortescue and Hall⁽⁴⁾ on the fuel elements of the Calder Hall reactor. It has been shown that these extended surfaces have at a certain dimension of fin a better heat transfer performance than the extended surfaces with longitudinal fins for flow direction, but much of the performance are still unknown.

In this paper, the effects of the dimensions of fin and channel on the heat transfer and the

pressure drop, the flow patterns between the fins and the effects of fin material are studied on the extended heating surfaces with transverse fins for flow direction.

2. Experimental equipment

The general arrangement of the experimental equipment is shown in Fig. 1. The extended heating surfaces with transverse fins for flow direction are placed at the bottom of the air channel of rectangular cross section. The air supplied from the atmosphere flows into the air channel by means of the turbo-fan.

Fig. 2 shows the extended heating surfaces. The extended heating surfaces are made of gun metal (thermal conductivity $\lambda_m=45$ kcal/m hr °C), and the dimensions of them are shown in Table 1. Four extended surfaces No. 1 to 4 of different pitches and seven surfaces No. 4A to 4G which are obtained by cutting the fins of No. 4 are used in this experiment. A plain plate without any fin is also tested for comparison.

The extended surface is heated on the bottom side by a 600 W electric heater. The bottom of the heater is fully insulated with asbestos, but a water box is provided for measuring the escaped heat through the asbestos.

The channel height d is adjustable at various positions by moving the cover forming the upper channel wall. In this experiment, the channel height was taken as $d=32, 24, 16, 8$ mm for each extended surface. The air rate of channel was

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changed by the regulation of the speed of turbo-fan and the exit damper. The air velocity is taken as a mean velocity at the channel above fins. This value is calculated from the measurement of Pitot tube which is located at point D, downstream of the channel, taking into consideration the variation of pressure and temperature between the heating surface and the point D.

All temperatures i. e. the extended surface temperatures, air inlet and outlet temperatures t_{ai} , t_{ao} , and cooling water inlet and outlet temperatures t_{wi} , t_{wo} are measured by thermo-couples (0.45 mm ϕ) and recorded with an electronic self-balancing recorder. The measuring points of temperatures $t_{f1} \sim t_{f4}$ were selected in a fin on which the flow state was expected to be steady. The primary surface temperatures $t_{b1} \sim t_{b4}$ between a given fin and the fin before it are also measured for getting the temperature of the heating base surface as a mean value of these four temperatures. The pressure drop in the extended surface is obtained by

measuring the pressure difference between inlet and outlet of the extended surface i. e. the points B and C with water manometers.

Fig. 2 Extended surfaces

Fig. 1 Schematic diagram of experimental equipment

Table 1 Dimensions of extended surfaces

Surface	Fin pitch p mm	Fin height l mm	p/l	Fin thickness $2y$ mm	Length of surface L mm	Breadth of surface B mm	Number of fins N	Area of heating base surface A m ²	Area of primary surface per unit A A_0 m ² /m ²	Area of fin surface per unit A A_f m ² /m ²
No. 1	50	49	1.020	3.90	390	144	8	0.056 16	0.920	2.01
No. 2	40	49.5	0.808	4.07	391	144	10	0.056 30	0.896	2.53
No. 3	30	48.5	0.619	3.96	388	144	13	0.055 87	0.867	3.25
No. 4	25	49.5	0.505	3.96	391	145	16	0.056 70	0.838	4.05
No. 4A	25	40.5	0.617	3.96	391	145	16	0.056 70	0.838	3.32
No. 4B	25	31	0.806	3.96	391	145	16	0.056 70	0.838	2.54
No. 4C	25	24.5	1.020	3.96	391	145	16	0.056 70	0.838	2.01
No. 4D	25	19	1.32	3.96	391	145	16	0.056 70	0.838	1.56
No. 4E	25	12	2.08	3.96	391	145	16	0.056 70	0.838	0.982
No. 4F	25	5	5.00	3.96	391	145	16	0.056 70	0.838	0.409
No. 4G	25	3	8.33	3.96	391	145	16	0.056 70	0.838	0.246

3. Flow patterns between the fins

The extended surface with transverse fins for flow direction has shown remarkable characteristics that the flowing stream in the channel is strongly disturbed by transverse fins and a number of different flow patterns take place between the fins. These flow patterns seem to be changed according to the fin dimensions and to have great effects on the heat transfer and the pressure drop. Therefore, using another two-dimensional water channel model, which was the same size as the heat transfer experimental equipment, the flow patterns between the fins were observed by means of aluminium powder sprinkled on the water.

Fig. 3 shows photographs and diagrammatical sketches of the flow patterns between the fins. It is interesting to note that eddy patterns between the fins are determined by the ratio of fin pitch to fin height p/l and almost independent of Reynolds number and channel height d . In the photographs, the values of p and d are held constant at 0.025 m and 0.024 m respectively and fin height l is varied.

For $p/l=0.5$, two circular eddies rotate in opposite directions. For $p/l=1$, one stable circular eddy occurs, and for $p/l=1.32$, this eddy becomes elongated and the friction factor, as mentioned later, is reduced to the minimum value. For $p/l=2.08$, a small eddy occurs under the elongated eddy, and for $p/l=5.0$, one elongated eddy forms just behind the fin. Under these flow states at $p/l=5.0$, the friction factor takes a maximum value.

These flow patterns between the fins are similar to the results of experiment on the transverse helical fin tubes investigated by Knudsen et al.⁽³⁾. An observation of these photographs reveals that the flow state in the channel will become almost steady through 3 or 4 fins from the entrance.

4. Heat transfer

The rate of heat transfer to air per unit area of the extended heating base surface can be expressed by the following equation (Fig. 4).

$$q = \alpha_0 A_0 \Delta t_0 + \alpha_f A_f \Delta t_m \quad \dots \dots \dots (1)$$

where

α_0, α_f : heat transfer coefficients on primary and

fin surface

A_0, A_f : areas of primary and fin surface per unit area of the extended heating base surface

$\Delta t_0, \Delta t_m$: temperature differences between primary surface and surrounding air, mean between fin and surrounding air respectively

As usual in the extended surfaces, supposing that

$$\alpha_0 = \alpha_f = \alpha \quad \dots\dots\dots (2)$$

and using fin efficiency

$$\phi_t = \Delta t_m / \Delta t_0 \quad \dots\dots\dots (3)$$

Eq. (1) may be written as follows.

$$q = \alpha (A_0 + \phi_t A_f) \Delta t_0 \quad \dots\dots\dots (4)$$

Then, if α^* indicates the heat transfer coefficient per unit area of the heating base surface,

$$q = \alpha^* \Delta t_0 \quad \dots\dots\dots (5)$$

where

$$\alpha^* = \alpha (A_0 + \phi_t A_f) \quad \dots\dots\dots (6)$$

Fin efficiency of straight fin of constant thickness used in this experiment is given by K. A. Gardner⁽⁵⁾

$$\phi_t = \frac{\tanh u}{u}, \quad u = l \sqrt{\frac{2\alpha}{\lambda_m y}} \quad \dots\dots\dots (7)$$

where, l : fin height m, λ_m : thermal conductivity of fin material kcal/m hr °C, y : half thickness of fin m.

This fin efficiency, as mentioned later, seems to be mainly applied to a longitudinal fin for flow direction from the assumptions in the process of deriving it. However, as it is too complicated to obtain the actual fin efficiency, Eq. (7) is approximately used to represent the results in this paper.

4.1 Experimental results In the experiment, q and Δt_0 are obtained by the following equations,

$$q = (Q_e - Q_w) / A, \quad \Delta t_0 = t_b - t_a$$

Fig. 4 Explanation of notations

hence α^* is immediately calculated from Eq. (5), where, Q_e : electrical input of heater kcal/hr, Q_w : heat absorbed in cooling water kcal/hr, A : area of the heating base surface m^2 , t_b is the temperature of the heating base surface as a mean value of the primary surface temperatures $t_{b1} \sim t_{b4}$, and t_a is the bulk mean temperature of air flowing the upper channel at the measuring part and calculated on the assumption that the air temperature makes linear change from inlet t_{a1} to outlet t_{a0} .

On the other hand, α has the correlation of Eq. (6) with ϕ_t , which is also a function of α as shown in Eq. (7). Therefore, in the first place the graphs of ϕ_t and α were drawn by Eq. (7) for each extended surface, and secondly the graphs of α^* and α were drawn by Eq. (6) using these graphs. Once the value of α^* is known experimentally, the values of α and ϕ_t can be simultaneously obtained from above graphs.

In the case of representing the heat transfer coefficient in dimensionless form, the equivalent diameter must be determined to describe the channel dimensions. It is a problem how to take the equivalent diameter for the extended surfaces, but the following equivalent diameter was used in this paper,

$$d_e = 2bd/(b+d)$$

where b is the channel breadth. The value of channel breadth in this experiment is $b=146$ mm, and the extended surfaces will be regarded nearly as two dimensional fins in the range of small values of d .

Since the experiments were carried out under forced air flow, the results were tried about the applicability of the following empirical equation

$$Nu^*, Nu = f(Re, p/l, d/l) \quad (8)$$

Fig. 5 Heat transfer of extended surface

where

$Nu^* = \alpha^* d_e / \lambda$: Nusselt number of the heating base surface

$Nu = \alpha d_e / \lambda$: Nusselt number of the effective extended surface

$Re = vd_e / \nu$: Reynolds number

and λ : thermal conductivity of air at temperature t_a kcal/m hr $^\circ C$, ν : dynamic viscosity of air at temperature t_a m^2/sec , p : fin pitch m, l : fin height m, d : channel height m. The dimensionless terms p/l and d/l are employed to represent the fin and channel dimensions. The term p/l decides the flow patterns between the fins and the term d/l shows the roughness of the channel.

Fig. 5 shows one of the experimental results. Nu^* and Nu have generally a linear correlation with Re . Though this is not satisfied in the range of lower air velocities, it may be caused by the effect of natural convection.

Fig. 6 shows a correlation of Nu^* , Nu and d/l at $Re = 5 \times 10^4$, Fig. 7 shows a correlation of Nu^* , Nu and p/l at $Re = 5 \times 10^4$ and $d/l = \text{constant}$, respectively.

Fig. 6 Effect of d/l on heat transfer

Fig. 7 Effect of p/l on heat transfer

From these results, the form of Eq. (8) was determined as the following equation.

$$Nu^* = 0.020 R_e^{0.90} (p/l)^{-0.16} (d/l)^{0.13} \quad \dots\dots\dots (9)$$

$$Nu = 0.0017 R_e^{1.04} (p/l)^{0.40} \quad \dots\dots\dots (10)$$

The experimental ranges are $R_e = (1.4 \sim 20) \times 10^4$, $p/l = 0.5 \sim 8.3$ and $d/l = 0.15 \sim 10.7$, and these equations are applied to the cases in which the natural convection has little effect.

As the experiments were carried out using the atmospheric pressure air, Prandtl number Pr was constant ($Pr = 0.71$), but Nusselt number is proportional to $Pr^{0.4}$ according to Kundsen et al.⁽⁸⁾ It has been also well known that Nusselt number is proportional to $Pr^{1/3}$ for the turbulent forced convection of a fluid whose Prandtl number is nearly unity. Though the effect of Prandtl number at separated flow as in this experiment is still uncertain, the following equations are given from Eqs. (9) and (10) on the assumption that Nusselt number is proportional to $Pr^{1/3}$ as in general turbulent flows.

$$Nu^* = 0.023 R_e^{0.90} Pr^{1/3} (p/l)^{-0.16} (d/l)^{0.13} \quad \dots\dots\dots (11)$$

$$Nu = 0.0019 R_e^{1.04} Pr^{1/3} (p/l)^{0.40} \quad \dots\dots\dots (12)$$

It may be considered that flow patterns between the fins are decided by p/l and the revolution of the eddies is prescribed by R_e . From these results, it is interesting to know that Nu , the value based on the effective extended surface, is determined by R_e and p/l , and is independent of d/l .

As the temperature of the heating base surface t_b is in the range of $30 \sim 110^\circ\text{C}$ in this experiment, the radiant heat from the extended surface can be neglected. This was experimentally confirmed by the tests carried out under different heat inputs of the extended surface and different inlet temperatures of air.

4.2 Effects of fin material Experimental

Fig. 8 Effect of fin material

results mentioned above are obtained for the extended surface with gun metal fins made in one piece with the base surface. But the variation of thermal conductivity of fin material brings different fin efficiencies i.e. effective extended surfaces and exerts effects on heat transfer performances.

Therefore, the fins of surface No. 2 were cut off and the grooves of 2 mm depth were provided there. The experiments were carried out on four kinds of extended surfaces which were made by soldering the embedded fins of copper, brass, carbon steel and stainless steel. There may be negligible thermal contact resistance between the fins and the heating base under such mode of attaching the fins⁽⁶⁾, and this was experimentally proved by comparison of gun metal fins in one piece with gun metal fins soldered.

To express the effect of fin material in heat transfer, there are two methods: one is to use the fin efficiency, and the other to introduce a new dimensionless term λ/λ_m considering the heating base surface area⁽⁷⁾. In this term, λ is the thermal conductivity of fluid and λ_m is the thermal conductivity of fin material. The latter method has been employed by Fortescue and Hall⁽⁴⁾ and others⁽⁸⁾⁽⁹⁾.

Fig. 8 shows the results of this experiment on Nu^* and Nu for each fin material. From these results it has been found that the effect of thermal conductivity of fin material can be expressed by Nu based on fin efficiency in the range of this experiment.

Fortescue and Hall suggested that Stanton number $St^* = \alpha^*/\rho g v c_p$ is correlated by $R_e(\lambda/\lambda_m)$, in which ρ , c_p are density and specific heat of fluid and g is acceleration of gravity. Fig. 9 shows the results with the same representation as Fortescue et al. of the case where the base surface area is taken as the heat transfer area. The effect of fin material is also well expressed by this method. The curve in Fig. 9 shows the result obtained by Fortescue et al. Eq. (11), which was obtained before on the assumption of $Nu \propto Pr^{1/3}$, is written as follows with $St^* = Nu^*/R_e R_r$.

$$St^* = 0.023 R_e^{-0.10} Pr^{-2/3} (p/l)^{-0.16} (d/l)^{0.13}$$

But Eq. (11) was obtained as the result of

Fig. 9 Effect of fin material

the experiments under the conditions of $\lambda=0.022$ kcal/m hr °C (air) and $\lambda_m=45$ kcal/m hr °C (gun metal); the above equation can be expressed as follows using $Re(\lambda/\lambda_m)$.

$$S_t^* = 0.011 \left(Re \frac{\lambda}{\lambda_m} \right)^{-0.10} P_r^{-2/3} (p/l)^{-0.16} (d/l)^{0.13} \quad (13)$$

Eq. (12) is also transformed into following form.

$$S_t = 0.0019 Re^{0.04} P_r^{-2/3} (p/l)^{0.40} \quad (14)$$

5. Pressure drop

As the flow resistance in the extended surface, the pressure drop ΔP was evaluated by excluding the inlet and outlet losses caused by sudden contraction and enlargement of the channel cross section. ΔP is calculated from the following equation by using measured pressure drop ΔP_2 between pressure taps B-C, pressure drop ΔP_1 measured in the preliminary tests on the channel with the thin plate fixed upon the fins (which contains the losses at inlet and outlet of the channel and the frictional pressure drop along the channel of rectangular cross section) and calculated frictional pressure drop $\Delta P_0^{(10)}$ along the channel.

$$\Delta P = \Delta P_2 - \Delta P_1 + \Delta P_0$$

Using the Fanning friction factor f , this pressure drop ΔP is given by

$$\Delta P = 4f \frac{L}{d_e} \frac{\rho v^2}{2} \quad (15)$$

where $d_e = 2bd/(b+d)$, L : length of the extended surface m, ρ : density of air kg sec²/m⁴. Some correlations of $f-Re$ on isothermal and nonisothermal conditions were compared in these experiments, but there was no difference between them.

Fig. 10 shows the correlation of f and Re for a turbulent flow obtained on surface No. 1. In general f is a function of Re and dimensionless ratios p/l , d/l which describe the shape of fin and channel, but Re will have little effect on f for a rough surface with transverse fins for flow direction.

Fig. 11 shows the correlation of f and d/l at $Re=5 \times 10^4$. From these results, an empirical equation of f for a turbulent flow is obtained as follows,

$$f = C(d/l)^{0.08} Re^{-0.05} \quad (16)$$

Fig. 10 Friction factor of extended surface

where the coefficient C is defined by the fin dimension p/l as shown in Fig. 12. It is known that f has its minimum value at $p/l=1.2 \sim 1.6$ and has its maximum value at $p/l=4 \sim 6$. Braun and Knudsen⁽¹¹⁾ have reported that for transverse fin tube in water flow, f has its maximum value when the ratio of fin spacing to fin height is $4 \sim 6$, and also for axial labyrinth packing, K. Komotori⁽¹²⁾ (air) and Y. Shimoyama and Y. Yamada⁽¹³⁾ (water) have shown that the value of f reaches a minimum when the ratio of fin spacing to fin height is $1 \sim 1.25$, a maximum when $3.5 \sim 5$. These results are in substantial agreement with those of this experiment. The reason why f has its minimum or maximum value depending on the value of p/l is not clear, but the following supposition may be made from consideration of flow patterns between the fins (Fig. 3).

For a small value of p/l , a great flow resistance is caused by fin tips projecting into the main flow, but when the value of p/l is more than unity, f is decreased because the eddies go out from the fin tips and the passage of main flow becomes comparatively smooth. But when the value of p/l becomes larger than 2, the turbulence of the main flow increases due to a splitting of the eddies between the fins, and especially at nearly $p/l=5$, larger unstable eddies compared to fin height cause an increase in flow resistance and the value of f

Fig. 11 Effect of d/l on friction factor

Fig. 12 Value of C for friction factor

will have its maximum value.

6. Correlation of heat transfer and pressure drop

Since the extended surfaces show in general the increase of flow resistance simultaneously with the increase of heat transfer, the performance as heat exchangers has to be decided in consideration of both the heat transfer and the pressure drop.

6.1 Comparison with plain plate At first consider the heat transfer and the pressure drop of plain plate. The heat transfer coefficient on the outer or inner wall of concentric tubes may be shown by the following equation⁽¹⁴⁾.

$$N_u = 0.023 R_e^{0.80} P_r^{1/3} (\mu_b/\mu_w)^{0.14}$$

where, $d_e = d_2 - d_1$, d_1 : outside diameter of inner tube, d_2 : inside diameter of outer tube and μ_b , μ_w : fluid viscosities at bulk mean temperature and tube wall temperature, respectively. As the heat rate is comparatively small in this experiment, $(\mu_b/\mu_w) \approx 1$ is assumed. Therefore, the upper equation gives

$$N_{u0} = 0.023 R_e^{0.80} P_r^{1/3} \dots (17)$$

The friction factor in the smooth channel of rectangular cross section for turbulent flow is presented by the following equation⁽¹⁰⁾.

$$f_0 = 0.119 R_e^{-0.28} \dots (18)$$

The experimental results on the plain plate without a fin have given a good agreement with Eqs. (17) and (18).

In the next place, compare the heat transfer and the pressure drop of the extended surface with those of the plain plate without a fin. Using Eqs. (17) and (18) for a plain plate, from Eqs. (11) and (16)

Fig. 13 Comparison of extended surface with plain plate

$$\frac{N_u^*}{N_{u0}} = R_e^{0.10} (p/l)^{-0.16} (d/l)^{0.13} \dots (19)$$

$$\frac{f}{f_0} = \frac{C}{0.119} (d/l)^{0.08} R_e^{0.23} \dots (20)$$

Fig. 13 shows correlations of $N_u^*/N_{u0}-R_e$ and f/f_0-R_e for each extended surface at $d/l=1$.

6.2 Performance of extended surfaces In order to take into account both the heat transfer and the pressure drop, the Colburn's heat transfer factor

$$j^* = N_u^*/R_e P_r^{1/3}$$

and the dimensionless ratio j^*/f are used in this paper. For flows in a smooth tube and along a plain plate, $j^* \approx f/2$ is applied (Colburn's analogy). However, as the flow resistance in the extended surface contains a great degree of the form resistance due to fins besides the friction, there is no correlation between j^* and f , so that it may be considered that the larger the value of j^*/f is, the better performance the heating surface will have.

In the case where the fin material is gun metal, j^*/f is obtained from Eqs. (11) and (16).

$$\frac{j^*}{f} = \frac{0.023}{C} R_e^{-0.05} (p/l)^{-0.16} (d/l)^{0.05} \dots (21)$$

Fig. 14 (a) shows the correlation of $j^*/f-R_e$ at $d/l=1$. Surface No. 4, for example, in spite of the highest heat transfer coefficient, has the lowest value of j^*/f owing to the greater flow resistance. Thus, the pressure drop has a greater influence on the performance of the extended surface than the heat transfer. The chain line in Fig. 14 (a) shows the value of j_0/f_0 calculated from Eqs. (17) and (18) concerning the plain plate placed in the channel of rectangular cross section. Fig. 14 (b) presents the correlation of $j^*/f-p/l$ at $R_e=5 \times 10^4$, and in the case of the value of $p/l=1.2 \sim 1.6$, the finned surface has shown the most effective performance.

Fig. 14 Comparison of heat transfer and pressure drop

7. Temperature distribution of fin

In the range of this experiment, the heat transfer performance for various fin dimensions and fin materials has been well shown using Gardner's fin efficiency⁽⁵⁾. But there are many assumptions in deriving Gardner's fin efficiency. Of these assumptions, the following is open to serious doubt.

(i) The heat transfer coefficient is the same all over the fin surface.

(ii) The temperature of the surrounding fluid is uniform.

Especially with transverse fins for flow direction, these two assumptions are hard to satisfy. Therefore, by the measured temperature distribution of fin, a few studies were carried out.

Fin efficiency ϕ_p as shown in the following equation was calculated from graphic integration of the temperature distribution of the fin measured, and this fin efficiency was compared with the Gardner's fin efficiency from Eqs. (3) and (7),

$$\phi_p = \frac{1}{l} \int_0^l (t_{fx} - t_a) dx / \Delta t_0$$

where t_{fx} is fin temperature measured, l is fin height and $\Delta t_0 = t_b - t_a$.

Fig. 15 shows the correlation of ϕ_p/ϕ_t and Re . Figure (a) presents the effect of fin dimensions. Though ϕ_p/ϕ_t is mainly dependent on Re and fin height l , ϕ_p is rather in good agreement with ϕ_t as shown in this figure. Figure (b) shows the case

Fig. 15 Fin efficiency ϕ_p/ϕ_t

of different thermal conductivities of fin material. There is a considerable difference between ϕ_p and ϕ_t for a fin material of low thermal conductivity such as stainless steel. Figure (c) is plotted with the dimensionless term $Re(\lambda/\lambda_m)$ suggested by Fortescue et al. It seems that ϕ_p/ϕ_t is also represented with $Re(\lambda/\lambda_m)$ as in Si^* .

Fig. 16 shows, as an example, the temperature distribution of fin, the temperature distribution of the surrounding air and the local heat transfer coefficient. The temperature distribution of fin measured t_{fx} is a little higher than the theoretical temperature distribution given by Gardner. The temperature t_{ax} shows that of the air measured by inserting a thermocouple. If the temperature difference Δt_x between the temperature of fin and the temperature of the surrounding air is known, the local heat transfer coefficient (the mean value at both sides of fin) at every position of fin can be obtained as follows.

The heat transferred through the cross section x is (Fig. 16)

$$Q = -\lambda_m \frac{dt_{fx}}{dx} 2y$$

Consequently the heat transferred to the air from both sides of the fin height dx is

$$dQ = \lambda_m \frac{d^2 t_{fx}}{dx^2} dx \cdot 2y = \alpha_x \Delta t_x \cdot 2dx$$

$$\therefore \alpha_x = \lambda_m y \frac{d^2 t_{fx}}{dx^2} / \Delta t_x \quad \dots \dots \dots (22)$$

In Fig. 16, α_x is the local heat transfer coefficient

Fig. 16 Distribution of fin temperature, air temperature and local heat transfer coefficient

cient in the case of $\Delta t_x = t_{fx} - t_{ax}$, α_x' is the local heat transfer coefficient in the case of $\Delta t_x = t_{fx} - t_a$ supposing that the temperature of the surrounding air is constant, and α is the mean heat transfer coefficient on the heating surface, as mentioned before. It has been known that the assumptions given by Gardner for fin efficiency are not corresponding with the actual value. Therefore in the smaller range of p/l than the one of this experiment ($p/l \geq 0.5$), this deviation becomes larger and further investigations on fin efficiency will be required.

8. Conclusions

On the extended heating surfaces with transverse fins for flow direction placed in the air channel of rectangular cross section, the effects of fin pitch p and height l , channel height d and Reynolds number R_e are experimentally studied on the flow patterns between the fins, the heat transfer and the pressure drop. The following may be summarized for the experimental ranges: $R_e = (1.4 \sim 20) \times 10^4$, $p/l = 0.5 \sim 8.3$ and $d/l = 0.15 \sim 10.7$.

(1) Regular flow patterns occur between the fins. These flow patterns are determined by p/l , being almost independent of d and R_e .

(2) The heat transfer performance based on the heating base surface is shown by Eq. (13).

(3) The heat transfer performance based on the effective extended surface taking into consideration the fin efficiency is presented by Eqs. (12) and (14). This is determined by R_e and p/l , being independent of d/l .

(4) The friction factor, as shown in Eq. (16), is mainly related with p/l and has its minimum value at $p/l = 1.2 \sim 1.6$, maximum at $p/l = 4 \sim 6$, respectively.

(5) The heat transfer and the pressure drop

are compared with those of the plain plate without a fin. As a parameter of the performance of the extended heating surface, the ratio j^*/j of Colburn's heat transfer factor to friction factor is calculated. By means of the ratio j^*/j , it is shown that the finned surface takes the most effective performance in the case of the value of $p/l = 1.2 \sim 1.6$.

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