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Free convective/radiative heat transfer from pin-fin arrays with a vertical base plate (general representation of heat transfer performance)

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Abstract—An experiment was carried out on free convective and radiative heat transfer from dense pin-fin arrays with a vertical isothermal base plate. Data analyses are made in consideration of the temperature distribution in the pin-fins, and of the radiative heat transfer. The free convective heat transfer characteristics of the pin-fin arrays are correlated fairly well with the Nusselt number and Rayleigh number, based on the horizontal spacing of the vertical pin arrays as a characteristic length. The generalized characteristics are similar to those of rectangular fin arrays. An experimental formula for the average heat transfer coefficient is derived.

1. INTRODUCTION

IT IS NATURAL to expect that a heat dissipator or heat sink with pin-fins has excellent performance because the heat transfer coefficient of a slender pin is much greater than that of a flat plate. However, only a few research works have reported on the free convective heat transfer of pin-fin dissipators; for example, a performance comparison between plate-fin-type and pin-fin-type dissipators by Aihara and Hatada [1], the authors' experiment [2] using a pin-fin of 1.23 mm diameter with a 100×100 mm^2 base plate, and Sparrow and Vemuri's experimental studies [3, 4] using arrays of comparatively large pin-fins (diam. 3.18–6.35 mm) arranged with a density of 0.31–1.33 pins cm^{-2}.

Generally speaking, the free convection of pins located in the upper part of multiple pin arrays is affected by the buoyant plumes from pins in the lower part; free convection from adjacent pins also interacts strongly [5]. Accordingly, it is very difficult to estimate the heat transfer characteristics of multiple pin-fin arrays from the heat transfer data on a single wire or a single pin array.

From the above-mentioned viewpoint and for the purpose of the development of high performance heat sinks for cooling electronic equipment, an experimental study has been carried out using 59 types of pin-fin dissipators with slender pins of a density one order higher than that used by Sparrow and Vemuri [3, 4]. The effects of temperature, pin diameter, pin length, length of pin array and pin spacings on the natural cooling performance are examined. The experimental data are analyzed considering the temperature distribution in pin-fins and radiative heat transfer. An experimental formula including all relevant parameters is derived.

2. EXPERIMENTAL APPARATUS AND MEASUREMENT METHOD

Figure 1 shows the construction details of the test heat dissipator and Table 1 lists the main dimensions

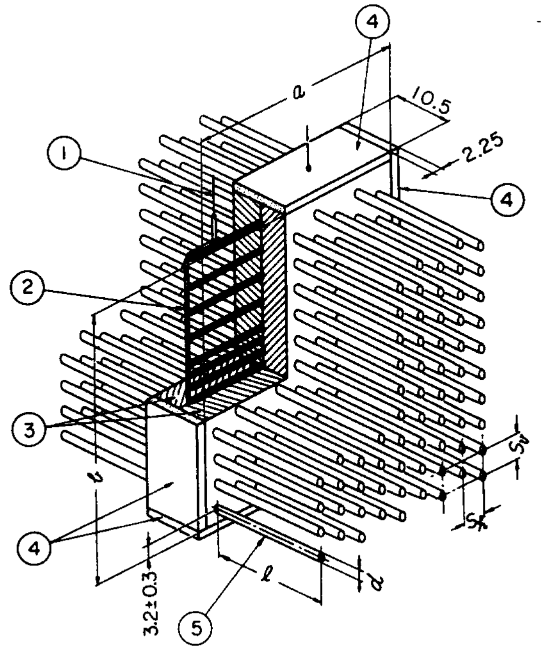

FIG. 1. Cut-away view of a test pin-fin dissipator.

NOMENCLATURE

A heat transfer area
a width of base plate, see Fig. 1.
b vertical height of base plate, see Fig. 1
d pin diameter
g gravitational acceleration
H effective length of a pin array, defined by equation (14)
h average convective heat transfer coefficient
h_{pt} total heat transfer coefficient of pin array, defined by equation (19)
l pin length, see Fig. 1
N population density of pins, (total number of pins)/$ab = (S_h S_v)^{-1}$
$Nu_{p,s}$ average Nusselt number of pin array, defined by equation (20)
$\dot{Q}$ heat transfer rate
$\dot{Q}_t$ total rate of heat transfer on one side of the model dissipator
Pr Prandtl number
Ra_S^* Rayleigh number, based on S_h, defined by equation (13)
S_h horizontal spacing between vertical pin arrays, see Fig. 1
S_v vertical pin spacing, see Fig. 1
T absolute temperature.

Greek symbols
β coefficient of volumetric thermal expansion
ε hemispherical total emissivity
ε_a apparent emissivity of test dissipator, equation (15)
λ thermal conductivity of air
λ_p thermal conductivity of pin material
ν kinematic viscosity of air
σ Stefan–Boltzmann constant
$\bar{\phi}$ dimensionless average temperature of pins, $(\bar{T}_p - T_a)/(T_b - T_a)$.

Subscripts
a ambient condition
b base plate or fluid physical properties at T_b
f fluid physical properties evaluated at the film temperature T_f
p pin
∞ single vertical flat plate.

Superscript
$^-$ average value.

Table 1. Main dimensions of test pin-fin dissipators (unit of dimensions, mm; N, pin number cm^{-2})

	d	a	b	l	S_v	S_h	H	N
Single array	2.00	6	108	70	3.5	∞	100.0	1.08–4.48
					5.0		97.0	
	4.00	6	108	70	7.0	∞	102.0	
					10.0		104.0	
					16.0		100.0	
Multiple arrays	1.23	100	50	32	2.11	5.73	45.5	2.42–9.90
				45	4.26–4.29	2.14	44.1	
				60		3.37	43.8	
						4.62	43.9	
						6.99	44.0	
						9.43	43.9	
			100	32	2.09	9.11	95.3	2.25–10.58
				45	4.25–4.27	2.12	94.7	
				60		2.79	92.6	
						4.59	92.8	
						7.25	92.8	
						9.23	93.0	
						10.43	93.0	
			200	32	4.24–4.25	4.69	194.6	1.86–4.89
				45		6.35	194.1	
				60		8.05	194.1	
						10.36	194.1	
						13.70	194.1	

measured. In general, rear heat loss from the base plate is a cause of major errors in experiments of this kind. Accordingly, the present test dissipators had a symmetrical geometry about a common base (3); this special construction minimized the heat leakage from the base plate. A nichrome heater (2), insulated by porcelain tubes, was inserted in the center of the copper base plate (3). The copper pins (5) were soft-soldered to the base plate in a staggered arrangement. All exposed surfaces of the test dissipators were coated with black epoxy-resin paint, of which the hemispherical total emissivity ε was 0.9 [6]. Heat flux meters (4), made of silicone-rubber, were attached to all edge surfaces of the base plate for measuring heat leakages from these surfaces [6, 7]. In order to obtain an isothermal condition in the base plate, the pitches of the nichrome heaters (2) were varied so as to be small at the lower part of the base plate and large at the upper part. The current leads (1) of the heater were drawn out through the heat flux meter (4) on the upper edge of the base plate.

This test dissipator was suspended by threads at the center of a wooden convection chamber of $0.9 \times 0.9 \times 1.6$ m^3, to allow the base plate to be vertically oriented; the convection chamber was placed within a sheath chamber of $2.7 \times 2.7 \times 1.8$ m^3. Air temperature, T_a, in the convection chamber was measured by a copper constantan thermocouple of 0.1 mm diameter located at the same height as the base plate. The temperature at six representative points on the base plate and 22 representative points on the multiple pin array (15 points for a single pin array) was measured by 0.1 mm diameter copper constantan thermocouples.

The net heat transfer rate, $\dot{Q}_t$, from one side of the test heat dissipator was determined by the following equation:

$$\dot{Q}_t = (\dot{Q}_{in} - \dot{Q}_m)/2 \qquad (1)$$

where $\dot{Q}_{in}$ is the electric input power and $\dot{Q}_m$ is the heat loss through the heat flux meters (4) on the edge surfaces of the base plate. Since the heat losses through the current leads (1) and thermocouple leads were less than 1%, these losses were ignored.

The experiment was carried out in the steady state, over the range of the difference of the base plate and ambient temperatures, $(T_b - T_a)$, from 20 to 85 K. The vertical temperature distribution of the base plate was within $\pm 4\%$ of $(T_b - T_a)$ for the case of $b = 200$ mm, and within $\pm 1\%$ for $b = 50$ mm. In order to observe the convection flow around the test dissipator, flow visualization and velocity measurements were also carried out by fine zinc stearate particles suspended in air [8].

3. RESULTS AND DISCUSSION

3.1. *Velocity distribution around the test heat dissipator*

Figures 2(a) and 3(a) show a typical flow pattern and the velocity distribution in a vertical cross-section 30 mm from the base plate, respectively. Figures 2(b) and 3(b) show a typical flow pattern and the velocity distribution in a vertical cross-section passing through the center of the dissipator. The flow towards the heat dissipator was stable, but the leaving flow showed a large fluctuation because of the disturbances induced while passing through the pin arrays. Thus, the velocity distributions of the leaving flow in the figures are based on plots of their maximum values. Since the pin array is more open to the surroundings than a plate-fin array, the pressure difference between its interior and the ambient is small; hence, fresh air inflow from the open side gaps of the pin-fin dissipator was lower than might otherwise be expected. Accordingly, it may be supposed that the influence of the base plate width, a, is not so large.

3.2. *Average heat transfer coefficient of single pin array*

A preliminary heat transfer experiment was carried out using single pin arrays, to determine the optimum vertical pin spacing, S_v, of the test multiple pin arrays.

In the present paper, the average convective heat transfer coefficient, h_p, of the pin arrays is defined as

$$h_p = \dot{Q}_p / A_p(\bar{T}_p - T_a) \qquad (2)$$

where A_p is the surface area of the pin arrays, $abN\pi d(l + d/4)$; $\bar{T}_p$ is the average pin temperature; and $\dot{Q}_p$ is the convective heat transfer rate from the pin arrays as given by

$$\dot{Q}_p = \dot{Q}_t - \dot{Q}_b - \dot{Q}_r. \qquad (3)$$

Here, the rate of convective heat transfer from the base plate, $\dot{Q}_b$, is obtained from the following equation:

$$\dot{Q}_b = A_b h_b (T_b - T_a) \qquad (4)$$

where A_b is the effective heat transfer area of the base plate $ab - N\pi d^2/4$, and h_b is the average heat transfer coefficient. However, no value of h_b for any of the pin-fin dissipators has yet been obtained; hence, the value of the average heat transfer coefficient for a vertical isothermal flat plate, given by equation (11), is utilized as an approximate substitute for h_b in the present section.

The radiative heat transfer rate from the single pin array and its base plate, $\dot{Q}_r$, can be represented approximately by the following equation, within a few percent error for $\varepsilon = 0.9$:

$$\dot{Q}_r \cong \varepsilon\sigma(\bar{T}_p^4 - T_a^4)[ab\{1 + N\pi dl + d/4\} - \varepsilon\pi dl F_{pp}(abN - 1)] \qquad (5)$$

where

$$F_{pp} = (2/\pi)[\sin^{-1}(d/S_v) + \sqrt{((S_v/d)^2 - 1)} - (S_v/d)]. \qquad (6)$$

Typical average heat transfer coefficients, h_p, obtained from the above equations, are plotted in Fig. 4 in dimensionless form against the Grashof number,

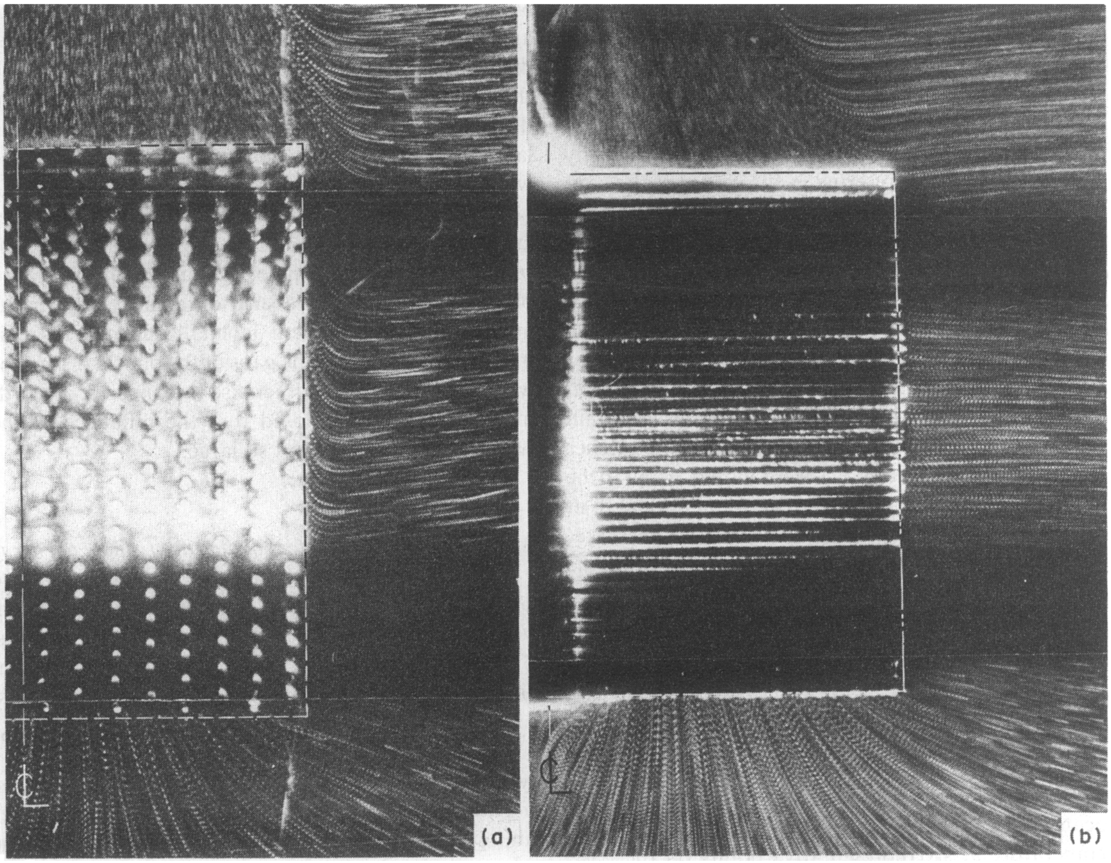

FIG. 2. Streamlines of a typical convection flow around a test pin-fin dissipator (a = 100, b = 100, l = 60, S_v = 4.26, S_h = 6.20 mm, $T_b - T_a$ = 20.3 K).

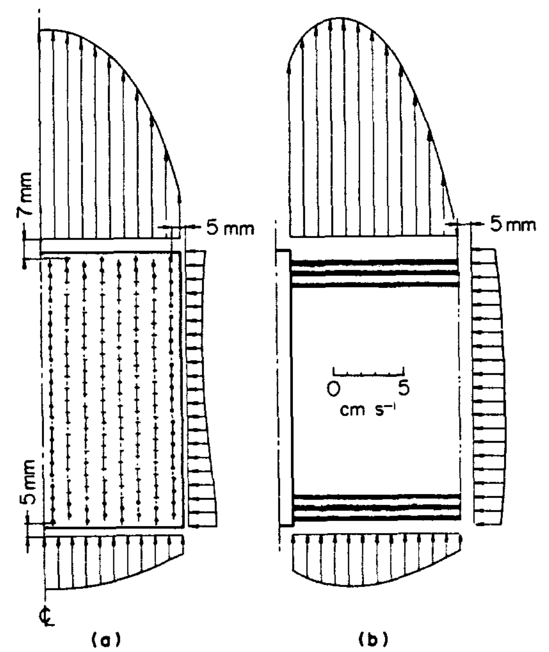

FIG. 3. Typical velocity distributions around a test pin-fin dissipator (a = 100, b = 100, l = 60, S_v = 4.26, S_h = 6.20 mm, $T_b - T_a$ = 20.3 K).

Gr_H, defined as

$$Gr_H = g\beta(\bar{T}_p - T_a)H^3/\nu_f^2 \tag{7}$$

where H is the effective length of a pin array, defined by equation (14). Marsters [9] performed an experiment on heat transfer from a single array of uniformly heated cylinders. Some of his data, which were obtained under nearly the same experimental conditions as the present one, are also plotted in Fig. 4, after reducing with the temperature of a cylinder at the mid-height. It should be noted that both the present and Marsters' data correlate well with the dimensionless parameters based on the effective array length, H, in spite of the differences in the pin diameter, pin length, and thermal condition, and that the gradient of the data on the log–log plot is 1/4. These data can be well correlated by the following formula, as shown by the solid lines in the figure:

$$h_p H/\lambda_f = [0.311 + 0.454 \ln (S_v/d)] Gr_H^{1/4}. \tag{8}$$

By comparison with the heat transfer coefficient for vertical rough surfaces [10], this empirical formula may be considered applicable to S_v/d = 1–4 and $Gr_H = 10^6$–10^8.

From equations (2), (8) and (14), the vertical pin

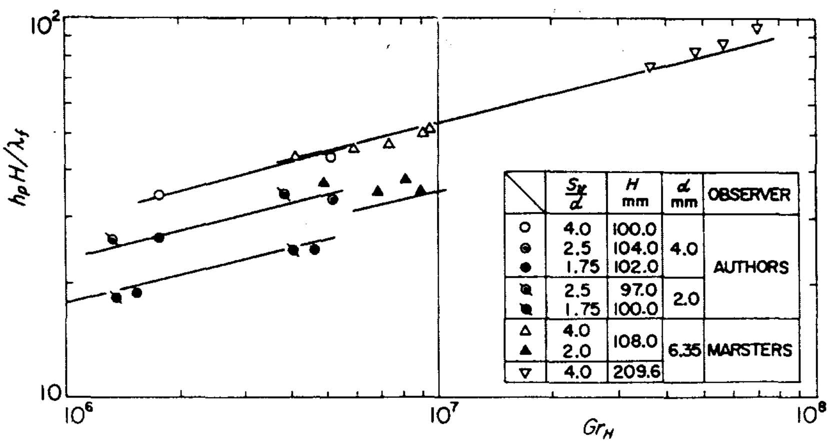

FIG. 4. Average heat transfer coefficient h_p for single pin arrays.

spacing which gives the maximum heat transfer rate per space occupied by a single pin array, $\dot{Q}_p/dHl$, is obtained as $S_v = 1.72$ mm for $d = 1.23$ mm and $H = 100$ mm. However, the expected pin spacing which gives the maximum heat transfer rate per amount of pin material is $S_v/d \geqslant 10$ [9, 11]. From these examinations and feasibility of manufacturing, the pin spacing selected for the present multiple pin arrays is $S_v = 4.3$ and 2.1 mm (see Table 1).

3.3. *Heat transfer performances of heat dissipator with multiple pin-fin arrays*

3.3.1. *Total heat transfer coefficient.* Figure 5 shows a relation typical of each rate of heat transfer revealed by the present experiment. The radiative heat transfer

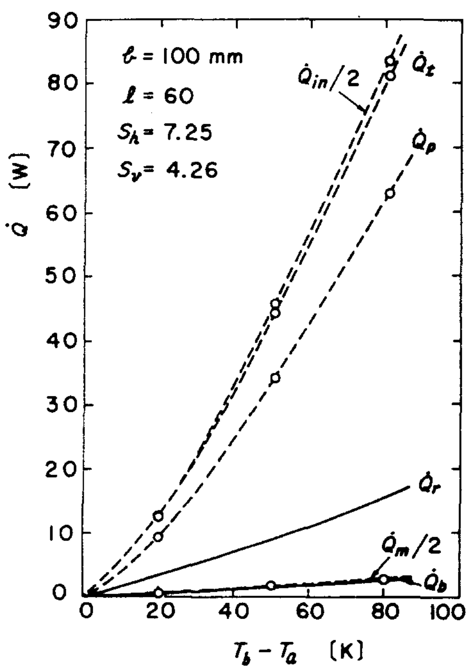

FIG. 5. Typical relation of each heat transfer component of a test heat dissipator with multiple pin-fin arrays.

component of the total heat transfer was 19–48% throughout the whole experiment; these values were rather large compared with those of a plate-fin dissipator [7].

Incidentally, Sparrow and Vemuri [3] tried to correlate their experimental data on the total heat transfer coefficient $h_t' = \dot{Q}_t/A_b(T_b - T_a)$ with the following Nusselt and Rayleigh numbers:

$$(Nu)_{S\text{-}V} = h_t'b/\lambda_f = \dot{Q}_t/a(T_b - T_a)\lambda_f \qquad (9)$$

$$Ra_b = Pr_f[g\beta(T_b - T_a)b^3/v_f^2]. \qquad (10)$$

Figure 6 is a trial plot of the present experimental data in terms of Sparrow and Vemuri's dimensionless parameters; in the figure, the similarity solution [12] of the average heat transfer coefficient, h_x, for a single vertical flat plate which is expressed approximately by the following equation, is also drawn by the solid line

$$Nu_x (= h_x b/\lambda_f) = 0.515 Ra_b^{1/4}. \qquad (11)$$

Denoting the total fin effectiveness as E_t, the Nusselt numbers $(Nu)_{S\text{-}V}$ and Nu_x are related by $(Nu)_{S\text{-}V} = E_t\, Nu_x$.

Thus, the $(Nu)_{S\text{-}V}$ data are scattered over a wide range according to the pin spacings and pin lengths; the values of $(Nu)_{S\text{-}V}$ increase with the base height, b. Consequently, for the case of a pin-fin dissipator with a large density, N, and a large convective component, as in the present study, it is difficult to obtain a general representation of its heat transfer characteristics in terms of the parameters defined by equations (9) and (10). Thereupon, the present data analysis is carried out by examining a specific convective component of the pin arrays.

3.3.2. *Determination of the average heat transfer coefficient for multiple pin arrays.* The average convective heat transfer coefficient, h_p, is determined as follows.

(i) For the average heat transfer coefficient of the base plate, h_b, the following heat transfer equation for the base plate of a vertical plate-fin dissipator is adopted approximately, because that for a pin-fin

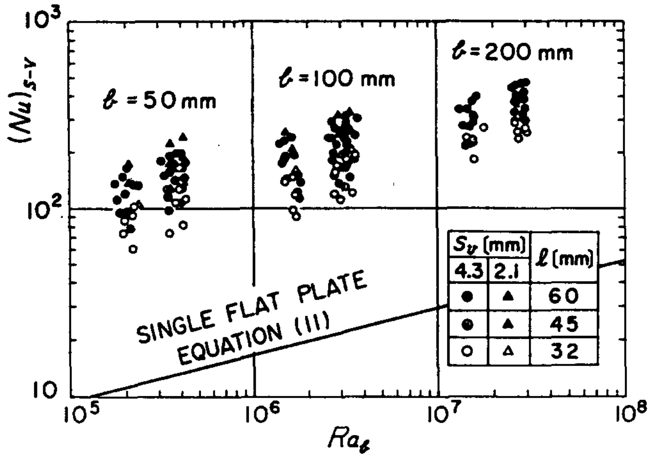

FIG. 6. Plot of the present experimental data on total heat transfer coefficient in terms of Sparrow and Vemuri's parameters [3].

dissipator has not yet been obtained. This may result in a slight overestimation of the heat transfer rate from the base plate, $\dot{Q}_b$, compared to the true value

$$h_b S_h/\lambda_b = (Ra_S^*/70)[1-\exp\{-(57/Ra_S^*)^{2/3}\}] \quad (12)$$

$$Ra_S^* = Pr_b[g\beta(T_b-T_a)S_h^3/\nu_b^2]S_h/H. \quad (13)$$

The applicable range of the above equation is $Ra_S^* < 10^3$. A more precise formula applicable for $Ra_S^* = 20-\infty$ is given in the original work [6]. Thus, H in equation (13) is the effective length of a vertical array of pins; it is given by the following equation when the pin number, N_v, in each pin array is equal:

$$H = (N_v-1)S_v+d. \quad (14)$$

When the pin numbers are not equal in odd and even lines, N_v is taken as the average of both values.

(ii) To the authors' knowledge, no method has produced a successful general representation of the radiative heat transfer from a radiator with multiple pin-fin arrays and a base plate. Hence, in the present report, the rate of radiative heat transfer, $\dot{Q}_r$, is estimated by the authors' numerical method [13] for radiant heat transfer from arbitrary three-dimensional black bodies; thus, the estimation is based on the assumption of a black surface, though the real emissivity, ε, of the surface of the test dissipator was 0.9. This assumption gives an error within a few percent, owing to the Hohlraum effect between the pin arrays and the base plate, as predicted from the analyses of two-dimensional semi-infinite pin arrays [14] and three-dimensional plate-fin arrays [15].

Now we introduce the apparent emissivity, ε_a [15], of the imaginary surface of area $ab+2l(a+b)$, covering the pin-fin dissipator. The radiative heat transfer from the dissipator is then expressed as:

$$\dot{Q}_r = \varepsilon_a\sigma(\bar{T}_p^4-T_a^4)[ab+2l(a+b)]. \quad (15)$$

The average temperature, $\bar{T}_p$, of the pins in this equation is predicted with an effective temperature difference, $(\bar{T}_p-T_a)_e$, derived in procedure (iii). Figure 7 shows a plot of the apparent emissivity, ε_a, obtained by the above-mentioned method, against the projected area ratio of pin to base, d^2/S_vS_h, and the total pin number $ab/S_hS_v(=abN)$. In the present experimental range of $d^2/S_vS_h = 0.03$–1, $ab/S_hS_v = 120$–1060, and $l/d = 26$–50, the effect of l/d on ε_a is very small (within an error of 2%); hence, ε_a can be shown to correlate by the following approximate equation:

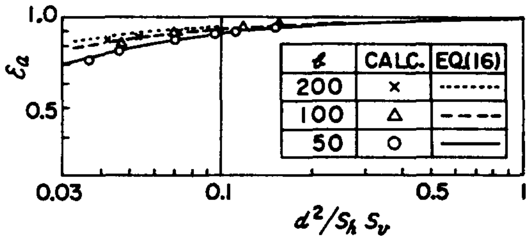

FIG. 7. Apparent emissivity ε_a of heat dissipators with multiple pin-fin arrays.

$$\varepsilon_a = (d^2/S_hS_v)^{\sqrt{(S_hS_v/ab)}}. \quad (16)$$

(iii) Since the test dissipators had a high density of pins, it was difficult to soft-solder thermocouples onto all the pins, with the exception of the pin-tips and the outside pins. Accordingly, from the viewpoint of practical fin design, the effective temperature difference between the pins and the ambient fluid, $(\bar{T}_p-T_a)_e$, which is obtained through the following procedure, is utilized as an approximate substitute for the average temperature difference, $(\bar{T}_p-T_a)$. Thus, the average heat transfer coefficient, h_p, is determined from equation (2).

Denoting the total heat transfer coefficient of the pin arrays as h_{pt}, the effective temperature difference, $(\bar{T}_p-T_a)_e$, can be obtained as a solution of a one-dimensional thermal conduction problem as follows:

$$\bar{\phi} = (\bar{T}_p-T_a)_e/(T_b-T_a) = \tanh(Ml)/Ml \quad (17)$$

where

$$M = 2\sqrt{(h_{pt}/(\lambda_p d))}. \tag{18}$$

However, it would be more practical to suppose the rate of radiative heat transfer from the base plate to be very small compared with the total heat transfer rate from the test dissipator because of the low value of $A_b/A_p = 0.01$–0.1 and the shielding effect of the large number of pins. Therefore, the total heat transfer coefficient, h_{pt}, based on the effective temperature difference, $(\bar{T}_p - T_a)$, can be expressed approximately as follows:

$$h_{pt} = (\dot{Q}_t - \dot{Q}_b)/[A_p(T_b - T_a)\bar{\phi}]. \tag{19}$$

Thus, $(\bar{T}_p - T_a)_e$ or $\bar{\phi}$ is determined by solving equations (17) and (19) simultaneously; $\bar{\phi}$ for the test dissipator was between 0.82 and 1.00.

As mentioned above, the coefficient of convective heat transfer, h_b, for the base plate, the rate of radiative heat transfer, $\dot{Q}_r$, and the effective temperature difference, $(\bar{T}_p - T_a)$, are obtained; then, substituting these values into equations (2)–(4), the average convective heat transfer coefficient for the multiple pin arrays, h_p, can be determined.

3.3.3. *General representation of average heat transfer coefficient, h_p.* The empirical formula (8) of heat transfer from a single pin array and the thermal boundary layer surrounding the array [11] closely resemble those for a vertical flat plate [5]. The fresh air inflow from the open side gap of the pin-fin dissipator was rather small, as shown in Figs. 2 and 3. Accordingly, the heat transfer behavior of the pin-fin dissipator is expected to be similar to that of a plate-fin dissipator [7]. Thereupon, the measured heat transfer coefficient, h_p, for the multiple pin arrays is rendered in terms of the following dimensionless parameters which are similar to those for the plate-fin arrays; the results are shown in Fig. 8

$$Nu_{p,s} = h_p S_h/\lambda_b \tag{20}$$

$$\bar{\phi}\, Ra_S^* = Pr_b[g\beta(\bar{T}_p - T_a)S_h^3/\nu_b^2]S_h/H. \tag{21}$$

Here, the thermophysical properties in these dimensionless parameters are evaluated at the base temperature, T_b, for simplicity, on the basis of the result of the authors' numerical analysis [16] regarding free convection in vertical ducts.

It is seen by examining Fig. 8 that the measured average Nusselt numbers, $Nu_{p,s}$, fall on or near the respective curves for each pin spacing, S_v, though small deviations remain depending on the pin length, l. In the figure, Aihara's empirical formula [7] for rectangular plate-fin arrays is also plotted for reference as dashed curves corresponding to the S_h/l values of the test pin-fin dissipators.

If we assume that the rate of convective heat transfer from the real pin array with effective length, H, is produced with an imaginary rectangular plate-fin of thickness d and length H, circumscribing the corresponding pin array and taking $H \cong N_v S_v$, then the apparent heat transfer coefficient, h_a, for this rectangular plate can be obtained as follows:

$$h_a = (\pi d/2S_v)h_p. \tag{22}$$

By applying this relation, a correlation of $(\pi d/2S_v) \times Nu_{p,s}$ vs $\bar{\phi}\, Ra_S^*$ is obtained as shown in Fig. 9, where the present empirical formula (8) for a single pin array is also plotted in rearranged form as follows:

$$(\pi d/2S_v)Nu_{p,s} = 0.534(d/S_v)[1 + 1.46\ln(S_v/d)](\bar{\phi}\, Ra_S^*)^{1/4}. \tag{23}$$

By introducing the new parameters $(\pi d/2S_v)Nu_{p,s}$

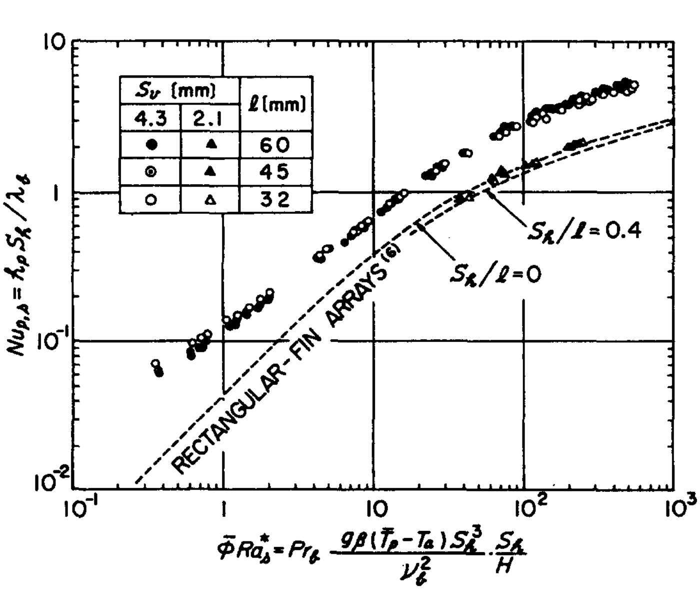

FIG. 8. Average convective heat transfer coefficient for multiple pin arrays rendered in terms of dimensionless parameters, equations (20) and (21).

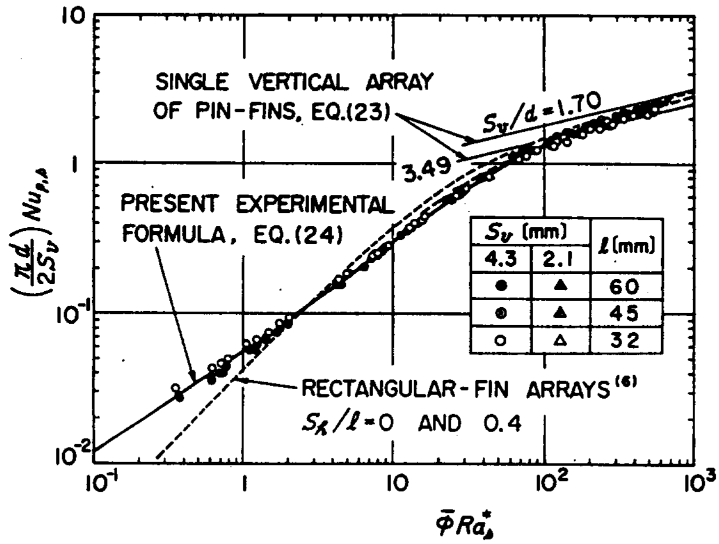

FIG. 9. General representation of average convective heat transfer coefficient h_p for multiple pin arrays.

and $\bar{\phi}\ Ra_S^*$, the heat transfer from the multiple pin arrays begins to show very similar characteristics to that from rectangular plate-fin arrays. With an increase in the modified Rayleigh number, $\bar{\phi}\ Ra_S^*$, the modified Nusselt number of the multiple pin arrays asymptotically approaches equation (23) for a single pin array. Thus, all the measured values in the present experiment can be well correlated by the following empirical formula within a deviation of $\pm 10\%$, regardless of the values of pin spacings, S_h and S_v, effective length of pin array, H, pin diameter, d, pin length, l, or temperature difference, $(T_b - T_a)$:

$$\left(\frac{\pi d}{2S_v}\right) Nu_{p,s} = \frac{1}{20}(\phi\ Ra_S^*)^{3/4}\left[1 - \exp\left(\frac{-120}{\bar{\phi}\ Ra_S^*}\right)\right]^{1/2} + \frac{1}{200}(\bar{\phi}\ Ra_S^*)^{1/4}. \quad (24)$$

The applicable range of this equation is $\bar{\phi}\ Ra_S^* = 0.3$–600, $S_v/d = 1.7$–3.5, and $l/H = 0.16$–1.37.

Although it is difficult to determine the relative merits and demerits of pin-fin dissipators and rectangular plate-fin dissipators exactly without a detailed examination such as in refs. [17, 18], they may be roughly described as follows. Namely, as can be seen from Fig. 9, the plate-fin arrays are more compact than the pin-fin arrays for $\bar{\phi}\ Ra_S^* = 5$–100, owing to the chimney effect; however, for $\bar{\phi}\ Ra_S^* < 3$, the pin-fin arrays with greater exposure to the ambient are superior to the plate-fin arrays owing to the effect of fresh air inflow. These tendencies are generally as expected from comparisons of free convection characteristics between plate-fin arrays and rectangular ducts [7] and between finite and infinite parallel plates [19, 20]. If the pin diameter is equal to the thickness of plate-fin, the heat transfer rate per unit material consumption from the pin arrays is four times that of the plate-fin arrays.

4. CONCLUSIONS

An experimental study was carried out on free convective heat transfer from 59 types of pin-fin dissipators, with a vertical base plate, a density of 1.08–10.58 pins cm^{-2}, and a surface emissivity of 0.9 in air. The results obtained are as follows.

(1) According to a flow visualization experiment and a velocity measurement of the flow field, the fresh air inflow from the open side gaps of the pin-fin heat dissipators was comparatively small; accordingly, the influence of the base plate width is not expected to be so significant.

(2) The average convective heat transfer coefficient, h_p, for a single pin array is well correlated by the Nusselt number, Nu_H, and Grashof number, Gr_H, based on the effective length of the pin array, H. The relation of Nu_H vs Gr_H is similar to that for a single vertical flat plate. Thus, a general representation of the present and Marsters' data on h_p, an empirical formula (8) for the ratio of vertical pin spacing to diameter $S_v/d = 1$–4 and $Gr_H = 10^6$–10^8 was derived.

(3) Numerical analysis was carried out on radiative heat transfer from a heat dissipator with multiple pin-fin arrays. A general expression (16) of the apparent emissivity for the pin-fin dissipator was presented.

(4) The average convective heat transfer coefficient of the multiple pin arrays, which is obtained by subtracting the convective heat transfer rate of the base plate and the radiative heat transfer rate of the heat dissipator from the total heat transfer rate, can be well correlated with the dimensionless parameters, based on the effective array length, H, and the horizontal pin spacing, S_h. The correlation is similar to that for rectangular fin arrays.

(5) By introducing the modified Nusselt and Rayleigh numbers proposed here, an empirical formula (24) was derived for a general representation of

the average heat transfer coefficient for multiple pin arrays, regardless of the values of pin spacings, S_h and S_v, effective array length, H, pin diameter, pin length, pin material, or temperature.

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TRANSFERT THERMIQUE PAR CONVECTION NATURELLE ET RAYONNEMENT POUR UN ARRANGEMENT D'AIGUILLES SUR UNE BASE PLANE VERTICALE (PRESENTATION GENERALE DES PERFORMANCES THERMIQUES)

Résumé—On conduit une expérience sur le transfert thermique par convection libre et rayonnement pour un arrangement dense d'aiguilles sur un plan vertical isotherme. Des analyses des résultats tiennent compte de la distribution de température dans les aiguilles et du rayonnement thermique. Les caractéristiques du transfert par convection naturelle des aiguilles sont unifiées à l'aide des nombres de Nusselt et de Reynolds basés sur l'espacement horizontal des nappes verticales d'aiguilles, considéré comme longueur caractéristique. Les caractéristiques généralisées sont semblables à celles des arrangements d'ailettes rectangulaires. On dérive une formule expérimentale pour le coefficient moyen de transfert thermique.

WÄRMEÜBERTRAGUNG DURCH FREIE KONVEKTION UND STRAHLUNG AN NADELRIPPENANORDNUNGEN MIT VERTIKALER GRUNDPLATTE

Zusammenfassung—Der Wärmeübergang durch freie Konvektion und Strahlung von einer dicht besetzten Nadelrippenanordnung mit vertikaler isothermer Grundplatte wird experimentell untersucht. Die Temperaturverteilung in den Nadelrippen und der Strahlungswärmeaustausch wird analysiert. Der Wärmeübergang durch freie Konvektion an den Nadelrippenanordnungen kann recht gut mit Hilfe der Nusselt-Zahl und der Rayleigh-Zahl korreliert werden, kennzeichnende Abmessung ist dabei der waagerechte Abstand zwischen zwei Nadelrippen. Die verallgemeinerte Beziehung ist ähnlich derjenigen bei Rechteckrippen-Anordnungen. Es wird eine empirische Beziehung für den mittleren Wärmeübergangskoeffizienten angegeben.

СВОБОДНОКОНВЕКТИВНЫЙ/РАДИАЦИОННЫЙ ТЕПЛОПЕРЕНОС ОТ УПАКОВКИ ИГОЛЬЧАТЫХ РЕБЕР С ВЕРТИКАЛЬНОЙ ОПОРНОЙ ПЛАСТИНОЙ

Аннотация—Проведены эксперименты по свободноконвективному и радиационному теплопереносу от плотной упаковки игольчатых ребер с вертикальной изотермической опорной пластиной. Полученные данные анализируются с учетом распределения температур в игольчатых ребрах, а также радиационного теплопереноса. Взаимосвязь между характеристиками свободноконвективного теплопереноса от упаковки игольчатых ребер достаточно хорошо устанавливается числами Нуссельта и Рэлея с использованием в качестве характерного размера горизонтального зазора между штырями. Обобщеные характеристики в данном случае сходны с полученными для прямоугольных оребренных упаковок. Выведена экспериментальная формула для среднего коэффициента теплопереноса.