MEASUREMENTS FOR FLOW RESISTANCE COEFFICIENT VALUES OF PERFORATED PLATES IN NATURAL AIR CONVECTION FLOWS

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ABSTRACT

A new method of obtaining flow resistance coefficient for perforated plates in low Reynolds number flows, such as natural air convection, was proposed. And some experimental values are obtained from the viewpoint of thermal design of electronic equipment casings where some perforations often used in the outlet and inlet vent to prevent dust from entering the casing.

The proposed method is based on the two expressions written for the enthalpy balance and for the balance between aerodynamic resistance and buoyancy force in a ventilation model. By these two expressions, flow resistance coefficient values can be evaluated even when only dissipated power and temperature rise is known. As a result, the relationship of the flow resistance coefficient values for perforated plates, Reynolds number and porosity coefficient is presented in a form fitting for practical engineering use. 18 kinds of perforated plates have been used in this study and the range of Reynolds number defined by the hole diameter and the air velocity in the ventilation duct was from 0.4 to 95. The experimental results include the Reynolds number and porosity coefficient influence on the resistance coefficient for perforated plates, as well as the perforated plates thickness to the diameter of hole effect on the flow resistance coefficient..

INTRODUCTION

With recent rapid increase in close packaging density and higher power for electronic equipment, the cooling design involved in casings which enclose them has been become important. Although it is easy to keep heat-dissipating components cooled by using a fan to blow air over them, nevertheless, if correctly ventilated, it would obviously be cheaper, more reliable and less noisy to choose natural convective cooling. There is, however, a need for flow resistance data regarding the flow path in the casing to effectively design ventilation methods, where the most significant factors in the flow path resistance are considered to be vent perforations, such as perforated plates [1].

The objectives of this paper, therefore, is to present some experimental flow resistance values for perforated plates in a natural convective flow, providing basic thermal data necessary for designing electronic equipment casing ventilation. However, in the conventional measuring method, which measures only in the higher Reynolds number region such as fully developed turbulent flow, there is a disadvantage in that the utmost measurement precision cannot be attained due to the pressure loss and velocity values being unidentifiable small. Data at a low Reynolds number, therefore, such as natural air convection are scarce. All the resistance data for perforated plates were obtained for forced convective flows (higher Reynolds number) [2-4]. Smith and Van Winkle [5] studied discharge coefficients through perforated plates at a wide Reynolds number range of 2000 to 20000 and Kolodzie and Van Winkle [6] also made a study at a 400 to 3000 Reynolds number range. Their work was limited to a lower porosity coefficient range of 0.023 to 0.158. Therefore, in this paper, a new method was designed to evaluate a flow resistance coefficient by considering the balance between motive power force for natural convection and flow resistance to movement and flow resistance values for perforated plates obtained by using the new method are presented. Ishizuka et al have already applied the new method to obtain several flow resistance coefficient values for wire nets in natural air convection flow [7, 8]

EVALUATION FOR FLOW RESISTANCE COEFFICIENT FOR PERFORATED PLATES

A ventilation model, shown in Figure 1, is considered. Assuming a uniform temperature distribution and a one-dimensional steady state flow, the two expressions below can be written, one for the overall energy balance and the other for the balance between flow resistance and buoyancy force.

$$Q = \rho c_p A u \Delta T \quad (1)$$

$$(\rho_a - \rho) g h = K_T \rho u^2 / 2 \quad (2)$$

where, Q is dissipated power, ρ is air density, c_p is specific heat of the air at constant pressure, u is air flow velocity, A is the duct cross section area, T is temperature rise, g is acceleration due to gravity, h is the distance between the perforated plate and the heater, and Kt is the flow resistance coefficient for the total system. The subscript, a, means the value at atmospheric pressure. Since the pressure change in the system is small, the expression is obtained assuming a perfect gas.

$$\frac{(\rho_a - \rho)}{\rho} = \frac{\Delta T}{T_a} \qquad (3)$$

and the equation for K_t is obtained as follows,

$$K_T = 2gh\Delta T^3 (\rho c_p A / Q)^2 \qquad (4)$$

This relationship has one important feature. The coefficient K_T can be determined from the measurements of only dissipated power and temperature rise. In addition, these two quantities are relatively easy to measure. By taking advantage of this, it is possible to evaluate the flow resistance coefficient for a perforated plate without measuring the pressure drop. Next, K_T is divided into two parts, K_0 and *K*, where K_0 is the coefficient for the system without a perforated plate. The *K* value can easily be obtained by using this relation and by making the same measurements as for K_t but without a perforated plate. The net coefficient, *K*, for the perforated plate can be obtained by simply subtracting K_0 from K_t, as follows,

$$K = K_T - K_0 \qquad (5)$$

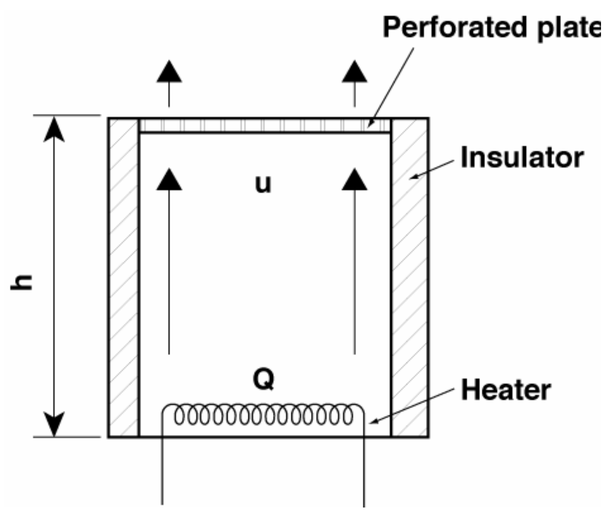


Fig.1 Ventilation model

EXPERIMENTS

Experimental Apparatus

Figure 2 illustrates the measurement system for realizing the ventilation model shown in Figure 1. The system consists of inner and outer ducts, 1.6 mm diameter sheath electric resistance heater (used as the input heater) on the inner duct bottom and a resistance plate on the top. Then, in order to make the inner duct as adiabatic as possible, a ribbon auxiliary heater, which has 70 ohm electric resistance and 30 mm in width, was wound around the outer wall of the duct. The outer duct was placed for use in wrapping the inner duct and to heat it efficiently. The two ducts were made of vinyl chloride and supported by a table constructed from angle iron. *To* measure temperature in the duct, nine thermocouples, made from 0.1 mm copper-constant, were placed in the duct. Thermocouples were located on the wall, in the wall vicinity and in the center of the top, middle and bottom sections of the duct, respectively, as shown in Fig.2. The distance from the bottom to the top, middle and bottom section are 0.92h, 0.5h and 0.2h, respectively, where h is the duct height. Another sheath thermocouple, 1.6 mm in diameter and 200 mm in length was prepared to measure temperature at other positions. Three of the nine thermocouples were cemented to the wall with epoxy glue. Temperature T, used in Eq. (4), was obtained from the average temperature values, except for those at the wall. The input heater shape was determined by trial and error, so that the temperature distribution should be uniform in the duct. Consequently, shape was as illustrated in Fig.3.

The ratio of height to diameter in this particular set-up was set at 5.7. The reason was as follows:

Firstly, K_0 measurement was made without a wire net. A linear relationship was obtained between K_0 and *H/D*. *D* from those measurements as shown in Fig.4. The 5.7 *H/D* ratio is at about the middle straight line which was obtained. Also, it was taken into account that, for a higher duct, it is more difficult to realize an isothermal condition inside the duct.

For this particular *H/D* ratio, K_0 becomes a function of the Reynolds number $Re_{,D}$, based on the duct diameter *D* and velocity u obtained from Eq. (1). From the results in Fig.5, the following expression has been obtained. and duct height *h* was done as *h*=5.*7D*. *D*=77.2 mm. The input heater has 7.9 ohm electric resistance as shown in Fig.3

EXPERIMENTAL PROCEDURE

After a resistance plate was placed on the top of the duct, the switches of the input heater and the ribbon heater were turned on. The upper and lower limits of the input heater power value in watts were determined by adjusting the heater voltage so that air temperature rise ΔT should be between 5 K and 80 K. This range is determined from a practical point of view. Monitoring center temperature Tc wall temperature Tw by a digital thermometer with ±0.1 K accuracy, ribbon heater voltage adjustments were repeated until the T_c and T_w readings met the following criterion. It is taken as a criterion by considering the measuring time and practical utility. Reynolds number on the duct flow, of course, were varied by input power voltage adjustment for the heater and since the amount of heat removed from the upper surface other than the free area of a resistance plate by natural convection , was estimated within 1 % of input power, it was neglected.

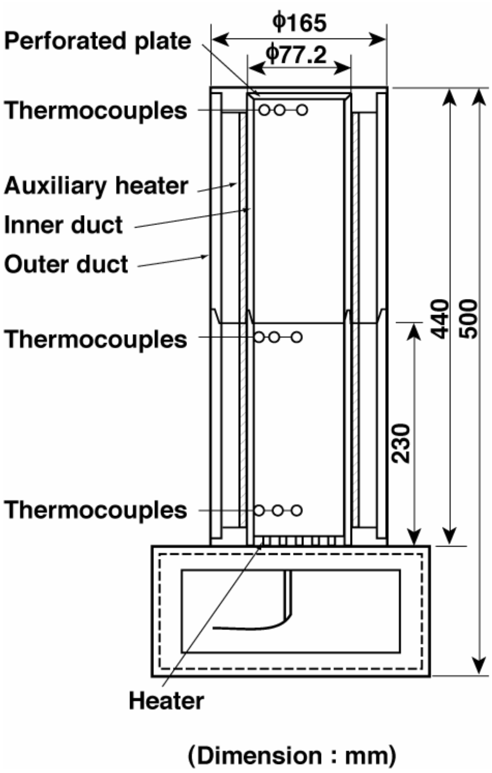


Fig.2 Measuring system

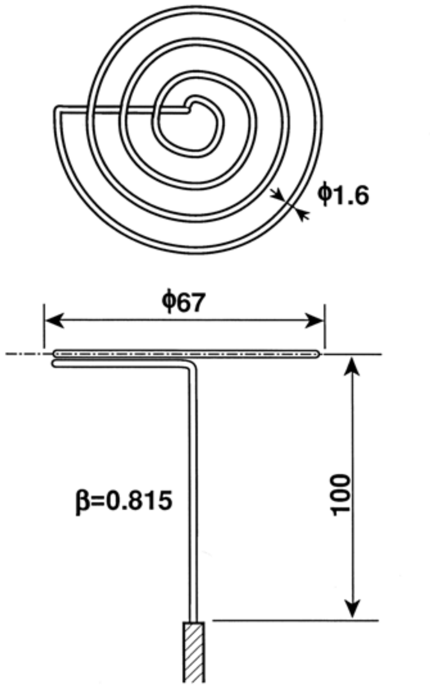


Fig.3 Heater

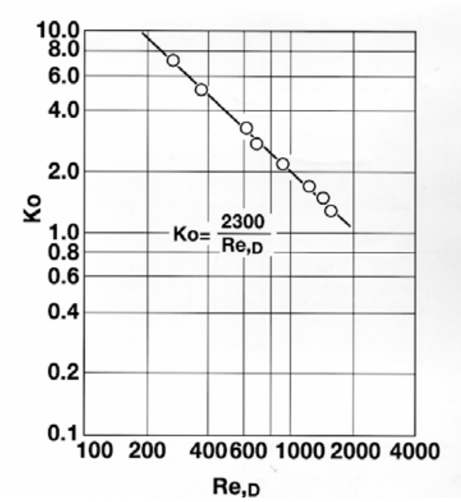


Fig.4 Relationship between flow resistance coefficient K_0 for measuring system without a perforated plate and tReyno0lds number based on the duct diameter

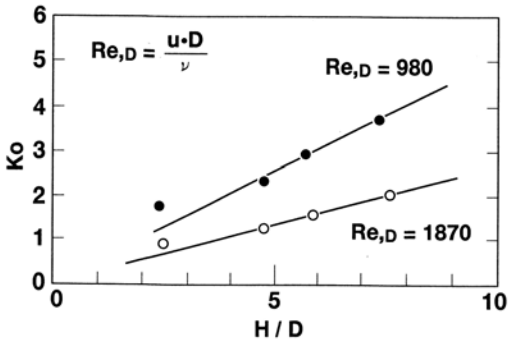
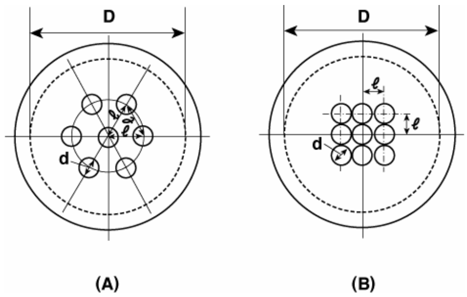


Fig.5 Relationship between flow resistance coefficient K_0 for measuring system without a perforated plate and the duct height *H*

$$\frac{T_w - T_c}{T_c} \leq 0.02 \qquad (6)$$

EXPERIMENTAL RESISTANCE PLATES

Figure 6 shows the layouts for a perforated plate. The hole pattern for perforated plates were hexagonal and square, as shown in Figs.(4a) and (4b), respectively. Parameters for the 24 perforated plates are listed in Table 1. Perforated plates were made of drilled aluminum.



(A) Hexagonal hole arrangement
(B) Square hole arrangement

Fig.6 Hole layout of perforated plates

EXPERIMENTAL RESULTS AND DISCUSSION

TEMPERATURE RISE DISTRIBUTION IN THE DUCT

Figure 7 shows temperature rise distribution in the duct with the ribbon heater switched on and off. When the ribbon heater is switched off, T_w is much lower than T_c and temperature distribution is not uniform. When the heater is switched on, temperature distribution is very uniform. Temperature measurement locations are illustrated in small figure in Fig.5, where y denotes the distance from the bottom and r/r_0 is the duct radius ratio. When the duct wall s cooler than the duct center, there may be an up flow of air heated by the input heater in the center and a down flow of air cooled by the wall in the wall vicinity. This may be a cause for no uniform temperature rise. However, from the results, it can easily be seen that the auxiliary ribbon heater has a strong effect on the uniform temperature rise distribution.

Table 1 Perforated Plate Dimensions

Plate No.	d (mm)	ℓ/d Ratio	t/d Ratio	β Ratio	Pertoration pattern
P1	4.0	1.25	0.5	0.58	Hexagonal
P2		1.50		0.40	
P3		1.75		0.30	
P4		2.0		0.23	
P5	5.0	1.20	0.2	0.63	
P6		1.40		0.46	
P7		1.60		0.35	
P8		1.80		0.28	
P9	4.0	1.50	0.25	0.34	Square
P10			0.50		
P11	2.0	2.0	0.50	0.21	
P12	8.0	1.50	0.125	0.29	
P13			0.50		
P14	4.0	3.0	0.25	0.10	
P15			0.50		
P16			0.75		
P17			1.0		
P18			1.5		
P19			2.0		
P20	6.0	2.0	0.16	0.18	
P21			0.33		
P22			0.67		
P23			1.00		
P24			1.33		

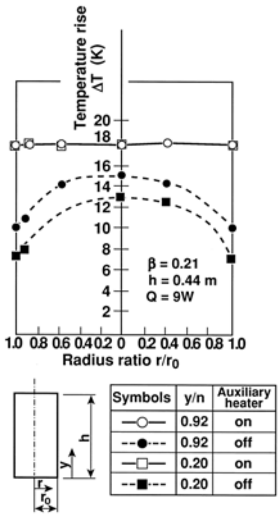


Fig.7 Temperature rise distribution in the duct when auxiliary heater switch is on or off

FLOW RESISTANCE COEFFICIENT OF MEASURING SYSTEM

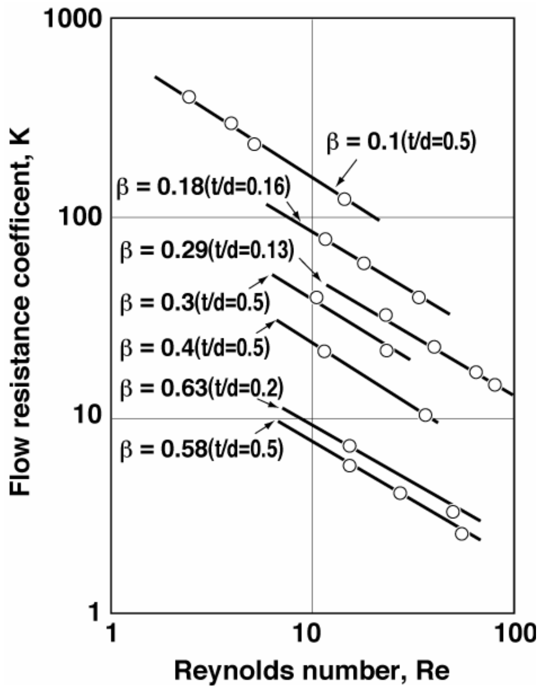


Fig 8 Relationship between flow resistance coefficient *K* and porosity coefficient β

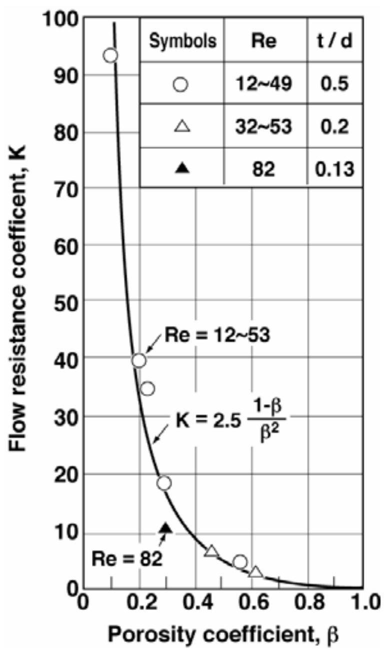


Fig.9 Relationship between the flow resistance coefficient K and Reynolds number Re

RELATIONSHIP BETWEEN FLOW RESISTANCE COEFFICIENT K AND REYNOLDS NUMBER Re

Figure 8 shows the relationship between the flow resistance coefficient K and Reynolds number Re for perforated plates. Reynolds number Re is based on the hole diameter for a perforated plate. From Fig.8, the following expression is obtained,

$$K = B_1 / \mathrm{Re}^{0.63} \quad (7)$$

where B_1 is constant for Re. However, if the flow resistance coefficient is considered for a single plate, the Re range is not so wide, since air temperature rise ΔT should be limited, as previously mentioned.

RELATIONSHIP BETWEEN FLOW RESISTANCE COEFFICIENT K AND POROSITY COEFFICIENT β

Figure 9 shows the relationships between flow resistance coefficient K and porosity coefficient β for perforated plates. Porosity coefficient β represents the ratio of free area for a perforated plate, considering the duct cross sectional area. For a perforated plate, the free area is calculated from the number of hole, the hole size and shape variety. In Fig.8, the results for the perforated plates are presented mainly at the ratio value of *t/d*=0.5 covering a 12 to 82 Reynolds number range. The *Re* range may be sufficient for practical use. Arranging the experimental results, using the following formula proposed by Collar (1939).

$$K = C\frac{(1-\beta)}{\beta^2} \quad (8)$$

where *C is* constant. The *C* value was chosen as *C*=2.5 to match most of the results for perforated plates. That is

$$K = 2.5\frac{(1-\beta)}{\beta^2} \quad (9)$$

Equation (11) predicts the results very well, except for the K value at *Re*=82 and *t/d*=0.125. It is suggested, from Fig.9, that as β decreases to less than 0.1, Eq. (11) may not predict the results well and that flow resistance coefficient K may be affected stronger by the ratio t/d than by Reynolds number *Re*. The former, however, is trivial, since the data for β<0.1 is of no practical use. Although the K value changes sharply for a 0.2 to 0.4 of β values. Eq. (10) predicts the change well.

COMPARISON BETWEEN PRESENT DATA AND DATA AT HIGHER REYNOLDS NUMBER

The comparison between present data in laminar flow and data at higher Reynolds numbers, such as turbulent flow, is interesting. Figure 10 shows flow resistance coefficient K as a function of porosity coefficient þ for a wide Reynolds number influence for a wide range on the flow resistance coefficient of perforated plates and to check the reliability of present data from a comparison of the tendency for the porosity coefficient influence. The chosen data at higher Reynolds numbers are those obtained by Taylor (1949) (500<Re<1500) and Macphail (1939) (Re=156). Present data are covering a 34-54 Reynolds number range. It can be seen from Fig.10, that Macphail's datum is very near on the present data line and that there is not a not a marked difference between the present data and Taylor's data, in spite of the large Reynolds number difference. It may be clear from this that the Reynolds number influence on flow resistance coefficient for perforated plates is rather weak and that the present data are considered to be reasonable. Thus it is expected that Eq. (10) should be applicable to the higher Reynolds number ranges.

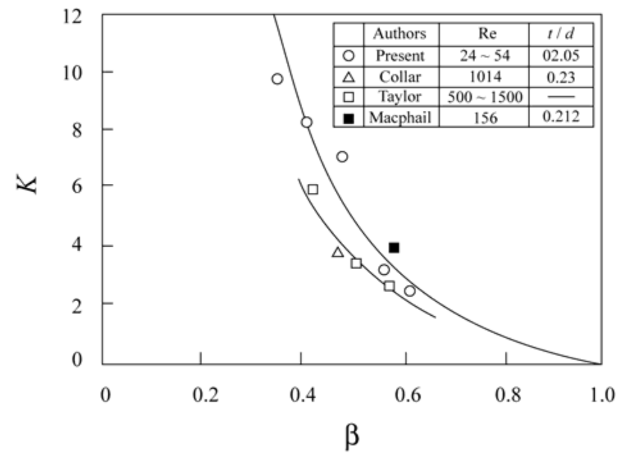


Fig.10 Comparison between present data and data at higher Reynolds numbers

RELATIONSHIP AMONG FLOW RESISTANCE COEFFICIENT K, REYNOLDS NUMBER Re AND POROSITY COEFFICIENT β

From a practical point of views, it is desirable to correlate the K values with Reynolds number Re and porosity coefficient β, altogether. Figure 11 shows the relationship among flow resistance coefficient *K*, Reynolds number *Re* based on hole diameter d and porosity coefficient β Flow resistance coefficient K values are arranged as a function of Re・ $\beta^2/(1-\beta)$, as shown in Fig.11. The term, *Re*・ $\beta^2/(1-\beta)$ was applied by considering the Eqs.(8) and (10). All the data are correlated well with the following expression from the best fit to the measured data.

$$K = 40\left\{\mathrm{Re}\,\beta^2/(1-\beta)\right\}^{-0.63} \tag{10}$$

$2.5 < Re < 82,\ 0.1 < \beta < 0.63,\ t/d=0.5$

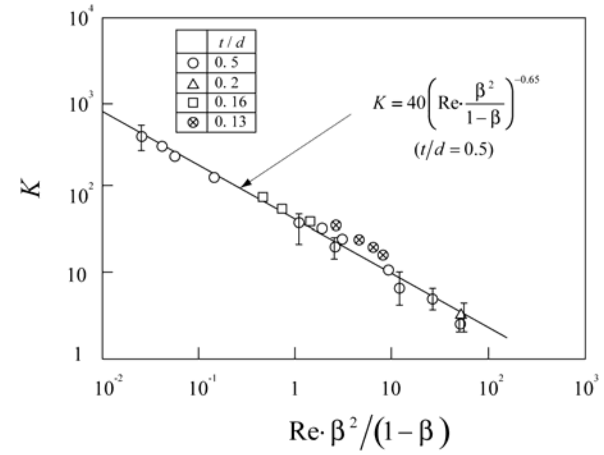


Fig.11 Relationship among the K values, Reynolds number Re and porosity coefficient β

Especially when the plate thickness to hole diameter ratio t/d is 0.5, the K values were correlated very well with equation (11). In Fig.11, finite-length lines, with open circular symbols showing measured values, define uncertainty width for the respective measurement value. For a value without the line, it is shown that the values with uncertainty width are within the respective white symbols. Here, uncertainty width is obtained from equation (5), considering that temperature rise uncertainty $d\Delta T$ is estimated as ±0.5 K and the heat leak at $dQ/Q = 0.02$ by using the following expression. The expression is conducted from the Eq. (3) as shown by Kline [9].

$$dK/K = ((9(d\Delta T)^2 + 4(dQ/Q)^2))^{1/2} \tag{12}$$

Using Eq. (12), it was found that the maximum uncertainty width was 25 % at β =0.63 due to the small temperature rise.

RELATIONSHIP BETWEEN FLOW RESISTANCE COEFFICIENT K AND PERFORATED PLATE THICKNESS TO HOLE DIAMETER t/d

It is clear that flow resistance coefficient values for perforated plates are also a function of the plate thickness. Therefore, the efforts were made to find the correlation between flow resistance coefficient and plate thickness. Figure 12 shows the relationship between flow resistance coefficient ratio $K/K_{0.5}$ and the plate thickness to hole diameter ratio *t/d* where $K_{0.5}$ denote the flow resistance coefficient value at *t/d* = 0.5. The reason for this is that the experimental results at *t/d*= 0.5 were presented mainly in Fig.9. From the results, the linear correlation can be expressed with good approximately by the following equation.

$$K/K_{0.5} = 0.33t/d + 0.835 \tag{11}$$

As the relationship data between flow resistance coefficient and the thickness to hole diameter ratio are scarce even at higher Reynolds number regions, Eq. (13) is considered to be important.

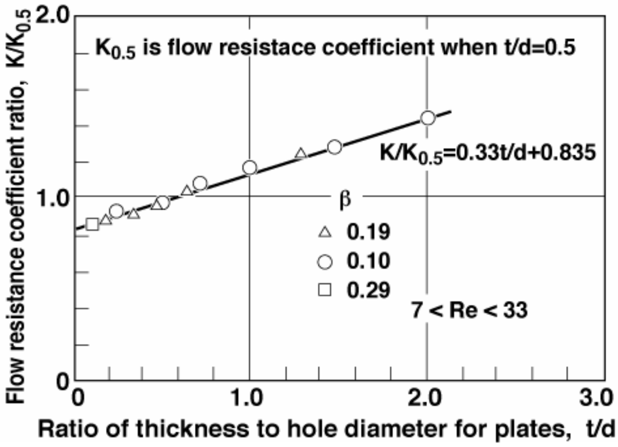


Fig.12 Relationship between flow resistance coefficient ratio $K/K_{0.5}$ and the plate thickness to hole diameter ratio t/d

CONCLUSION

A measuring system used to obtain flow resistance coefficient values for perforated plates in natural air convection has been designed and flow resistance coefficient values for perforated plates are presented. The results are very well correlated as functions of Re ^{2}/ (1-) and t/d. The results are very useful for designers who are concerned with electronic equipment thermal dissipation.

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