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ABSTRACT

The forced convection experiment of fan plate-finned tube heat exchangers which are mainly used in air conditioners and the forced convection model have been studied for which the heat transfer effect of this type exchangers has not been proposed by a model. All measurements were made when there is no refrigerant flow in the pipe at Reynolds's number of 580. According to the experiment results, basing on the threshold value that is defined as the strength of virtual average flow of which the free flow velocity capacity was used as a standard, the forced convection effect of the flow between the finned tube set and the fan can be divided into two regions. To fully utilize the heat transfer effect of the fan plate-finned tube heat exchangers, the model of the forced convection is proposed by velocity scale and length scale here.

NOMENCLATURE

V: Velocity
R_e: Reynolds number
x: Distance between plate-finned tube set and propeller
V_{ave}: Average velocity
V_e: Velocity scale
D: Propeller diameter
r: Radius of the divided calculation
U: Inlet velocity of plate-finned tube set
D_e: Length scale
N_u: Nusselt number

INTRODUCTION

Calculation of the heat transfer caused by forced convection is useful to design a compact heat exchanger. The efficient generation and utilization of heat convection have influences host of design parameters, such as the location between the fan and the plate-finned tube set, characteristics of the fan, dimensions and layout of the plate-finned tube set, and the location of other peripheral mechanical parts. Hence, forced convection modeling under actual production conditions of the inlet wind velocity of the plate-finned tube set meeting its design requirement is essential to designing of a compact heat exchanger.

Usually heat characteristics of forced convection in a uniform flow field (i.e., uniform inlet wind velocity for the plate-finned tube set) of a planar plate-finned tube set were tested in a wind tunnel [(1), (2), (3)]. They developed a correlation of heat transfer and flow loss of heat exchangers using length scale and velocity scale for forced convection under uniform flow conditions [(4)]. When designing an actual compact heat exchanger, however, inlet uniform wind velocity to the plate-finned tube set such as in above-mentioned wind tunnel cannot be expected and it is difficult to precisely predict the heat transfer and flow loss performance of the heat exchanger as a consequence. The prediction may require measuring the complex flow field of the plate-finned tube set in the practical heat exchanger in detail taking into account the effects of forced convection in terms of the intensity of the wind velocity, and then categorizing the forced convection area by virtual average flow velocity on the basis of free flow volume [(5)]. This kind of studies may contribute to pneumatic performance enhancement of blowers and efficiency improvement of passively controlled flow [(6)] for various kinds of exhausters to transport materials.

Authors made an investigation into the intensity distributions of forced convection flow around a plate-finned tube set (at each cross section of, x/D=-1, x/D=-0.75, x/D=-0.5) to establish a correlation of heat transfer and flow loss performances of a heat exchanger under actual production conditions, and then have come up with a forced convection model of a heat exchanger by defining a length scale and a velocity scale taking flow field affecting on fan characteristics in consideration.

EXPERIMENT APPARATUS AND TECHNIQUES

Coordinate System and experimental Conditions

Figure 1 shows the heat exchanger under experimenting, and coordinates system, which is a right-hand system of coordinates originating from the center of the propeller, with the x-axis direction being in parallel with the propeller shaft, the y-axis direction perpendicular to the ground, and the z-axis direction perpendicular to the x-y plane.

The fan propeller is 465 mm in diameter, and the distance from its top to the rear of the motor is 223.2 mm. The propeller has a bell mouth. Flow intensity measurement was carried out at the three cross sections in the flow direction located at x=-0.5D, -0.75D and -D from the coordinate origin, with measurement intervals of 0.05D in the y- and x- axis directions. Experiment results are presented with the bend being turned 90 degrees clockwise. Table 1 summarizes the test conditions. Propeller fan velocity intensity distributions were measured in the same system of coordinates with the finned tube set and other surrounding mechanical elements (bell mouth retained in position) removed. The measurement was carried at the two cross sections in the flow direction located at x=-0.5D and-D from the coordinate origin.

With the plate-finned tube set at a inlet wind velocity of 1.5 m/s [(5)], the typical length [(5)] De_c=5.8 mm based on the minimum free volume of the heat exchanger and the typical length D=465mm based on the fan propeller diameter yielded the Reynolds number 589, or 4.67 x 10^4. The plate-fined tube set comprised a row of samples, with a fin pitch of 1.8 mm and a fin width of 19.05 mm. There was no refrigerant flow in the tube during experimenting.

Measuring Method

Flow intensity measurement was carried out using a non-directional spherical wind velocity measuring probe (Kanomax, 1504) and a multipoint anemometer (Kanomax, 6604). Figure 2 shows a picture of the probe. The measuring probe was calibrated at a temperature of 21.8℃ and a humidity of 70.0%RH under an atmospheric pressure of 1017.7 hPa as shown in Figure 3. Measurement signals were transferred to a personal computer at a sampling frequency of 1 kHz and for a sampling duration of 120 seconds. The measurement data was then subjected to averaging.

The spherical wind velocity measuring probe was mounted on a 3D traverser with a traveling accuracy of 0.8 mm, which was capable of traveling 500 mm in the flow direction, 800 mm in the spanwise direction, and 700 mm in the vertical direction.

RESULTS AND DISCUSSION

Streamwise velocity contour of fan

Figure 4 shows a contour of the velocity strength of the fan in the suction area at x/D=-0.5 and x/D=-1. The vertical and horizontal axes represent the distances in the vertical and spanwise directions made dimensionless by the propeller diameter D, while the velocity intensity is made dimensionless by the design requirement inlet wind velocity U (U=1.5 m/s). Due to restrictions of the limited space available, only the test results in the first quadrant of the y-z plane are presented here. Figure 4 (a) shows that V/U ranged from 1 to 1.5 (1.5 to 2.25 m/s) at x/D=-0.5 in the regions of 0.5 > y/D and z/D> 0.15. V/U is found to span from 1 to 1.5 (1.5 to 2.25 m/s) in the positive y- and z-axis directions in the regions of 0.65 > y/D and z/D > 0.5. These findings indicate fast flow in the region of the fan propeller diameter, keeping up with a flow intensity at inlet wind velocity of 1.5 m/s up to 1.3 D. Sudden drops in flow velocity were observed in the regions of y/D < 0.15 and z/D < 0.15 in the region equivalent to the motor diameter (3/20D). The sudden change in flow intensity in the region of 0.25 > z/D >0.15 was due to the presence of a fan clamp. In Figure 4 (b), the flow intensity reached 0.9 to 1.05 (1.35 to 1.58 m/s) at x/D = -1 in the regions y/D and z/D < 0.45, indicating that rapid convergence at the center of the cross section under test in the region of V>1.5 m/s when compared with the distributions at x/D=-0.5. These findings suggest that the core region (with essentially uniform distributions of flow) on the intake side of the propeller fan is so short that the kinetic energy is concentrated in the region of -0.48 > x/D > -1 and (-0.48 > x/D being the limitation on the motor size).

From the measurement results given in Figure 4, the plate-finned tube set can be locate to meet the design inlet wind velocity against the distributions of the fan velocity intensity. The horizontal axis in Figure 5 represents the position in the flow direction x/D made dimensionless by D, while the vertical axis designates the velocity ratio V/U, with parameters being the values of y/D and z/D. Figure 5 indicates that the flow intensity gets higher than 1.5 m/s in the regions of y/D and z/D <1.2 as the plate-finned tube set is located at x/D<-0.625. In the region of -1< x/D

< -0.625, the intensity of the forced convection field generated by the fan stands at about 1 m/s in the regions of y/D and z/D <1.2.

Spanwise variation of velocity

Figure 6 shows a plate-finned tube set located (rear) at x/D = -0.64 (See Figure 5). Distributions of the flow intensity at the two typical positions in the flow direction x/D = -0.5 and x/D = -0.75 (51 mm apart from the plate-finned tube set plane) are presented. The vertical and horizontal axes in Figure 6 represent the plate-finned tube set in the z-axis direction made dimensionless using the velocity ratio V/U and D, respectively.

First, distributions at the flow position x/D = -0.5 (Figure 6 (a)) are examined. At each measurement position of y/D, spanwise distributions of the V/U essentially range from 1 to 2.5. Any distributions out of this range have been associated with the fan clamp jig, effects from the motor, and the upper and lower boundaries of the plate-finned tube set. In comparison with the results in Figure 5, the flow through the fins in the plate-finned tube set has gone through a significant dynamic pressure drop, possibly serving as a means of heat transport. Distributions of the flow in the plane perpendicular to the x-axis of the plate-finned tube set, with the exception of the aforementioned results out of range, have less variation at each measurement position in the y-axis direction than do distributions in the plane in parallel with the x-axis, probably because of changes in the distance between the edge of the propeller side and the measurement position.

Next, the measurement position x/D=-0.75 in the vicinity of the wind flow inlet of the plate-finned tube set is examined (Figure 6 (b)). Spanwise distributions of the velocity ratio V/U fell between 1 and 1.2 when compared with distributions in the velocity at x/D=-0.5 at the exit of the plate-finned tube set, decreasing on both side and minimizing at the bend of the plate-finned tube set. Falls in the velocity intensity on the left side (z/D>-0.5) are due probably to the structural effects of the neighborhood of the plate-finned tube set as shown in Figure 1. The gradient of changes in the velocity intensity of the plate-finned tube set is expressed as $\left|\frac{D}{U}\cdot\frac{\partial V}{\partial z}\right|$ > 3.5, where the velocity has an intensity of 1.5 m/s or lower - lower than the design inlet wind velocity for the heat exchanger.

Forced convection model

Definition of quite flow section

First, the flow rate in the flow directions x/D = -0.5 and x/D = -1 is determined.

$$Q = \iint_{S_i} 2\pi r V_a dr \tag{1}$$

where

r: Radius of the divided calculation

V_a: Average flow velocity in the area of the divided calculation

Next, the average flow velocity is obtained.

$$V_{ave} = Q / S \tag{2}$$

where

S: Area delineating the flow velocity distributions ($S = r^2 \cdot \pi$)

The table below summarizes the calculation results.

x/D	-0.5	-1
V_{ave} /U	1.2	0.7
r /D	0.7	0.95

where

$$\frac{x}{D} = f\left(\frac{V_{ave}}{U}\right) \tag{3}$$

$$\frac{r}{D} = g\left(f\left(\frac{V_{ave}}{U}\right)\right) \tag{4}$$

With the equivalent passage cross section definition: $V_{ave}/U=1$, the value of r/D is the radius of the equivalent passage cross section made dimensionless, so that the plate-finned tube set location x/D can be determined from Eqs. (3) and (4) and from Figure 7. Further, the plate-finned tube set cross section can be determined by solving the equation:

$$S_{cs} = 4r^2 \tag{5}$$

Length scale and velocity scale

Length scale definition:

Length scale (D_e) = 4 (Equivalent passage cross section)・(Fin width)/(Heat transfer area) (6)

Velocity scale definition:

Velocity scale (V_e) = A ｛Σ[(Fin pitch)・(Row Pitch)/Equivalent passage cross section]｝ (7)

where,

$$A = (\frac{\rho_{ai}}{\rho_a}) \cdot U$$

The relationship between the virtual average flow velocities V_e and V_{ave} relative to a free flow volume is expressed in an equation as:

$$\frac{V_{ave}}{U} = \frac{V_{ave}}{V_e} \cdot \frac{\sum [\, P_F \cdot S_1 \,]}{\pi \cdot r^2} \cdot (\frac{\rho_{ai}}{\rho_a}) \qquad (8)$$

Division flow field according to the V_e

When $\frac{V_{ave}}{U} = 1$, $V_e = V_{ave} \cdot \frac{\sum [\, P_F \cdot S_1 \,]}{\pi \cdot r^2} \cdot (\frac{\rho_{ai}}{\rho_a})$ results from $\frac{x}{D}$. If the value of V_e is defined as a threshold, the value of x/D allows the area of forced convection to be divided into two regions in the system of coordinates shown in Figure 1; the region (a) in which a forced convection intensity higher than V_e is obtained with a reduced equivalent passage cross section and one (b) in which a forced convection intensity lower than V_e is available with an increased equivalent passage cross section. The Reynolds number and the Nusselt number of a plate-finned tube set compact heat exchanger of this type can be calculated from Eqs. (9) and (10).

$$R_e = \frac{V_e \cdot D_e}{\nu_a} \qquad (9)$$

$$N_u = \frac{h \cdot D_e}{\lambda_a} \qquad (10)$$

Thus, if the fan propeller diameter D and the design inlet wind velocity U are known, the plate-finned tube set location x /D and the equivalent passage cross section (πr^2) can be determined from Eqs. (3) and (4) and from Figure 7. Then, the length scale and the velocity scale of the forced convection flow field can be calculated from Eqs. (9) and (10). Namely, if the plate-finned tube is thus located, the design inlet wind velocity U will yield the effect of forced convection in that area of the plate-finned tube set that is wetted by r. x/D being a function of V_{ave}/U, the virtual average flow velocity V_e relative to a free flow volume allows the area of forced convection to be divided. Thus, with its fan and plate-finned tube set meeting the design inlet wind velocity U, this model establishes the mutual positional relationship between production machines, equivalent passage cross section, and the cross section of the plate-finned tube set.

CONCLUSIONS

To examine changes in the spanwise flow intensity in the forced convection flow field in a compact heat exchanger that is built of a fan and a plate-finned tube set, and also characteristics of the fan, under its production conditions, distributions of the flow around the plate-finned tube set and in the after-wash area of the fan were measured. The forced convection model of a heat exchanger findings of this study are summarized as follows:

1) The positional relationship between a fan and a plate-finned tube set is shown to produce the forced convection intensity stronger than the designed inlet wind's of U.

2) Equivalent passage cross-section definition established a length scale and a velocity scale for the forced convection flow field, and is available to define the required dimensions of the plate-finned tube set.

3) The value of V_e allows the area of forced convection to be divided.

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Table 1 Experimental conditions

	Front panel	Side Panel
x	-D,-3D/4,-D/2	-D,-3D/4,-D/2
y	-12D/20~0~12D/20	-12D/20~12D/20
z	-3D/4 ~0~13D/20	7D/10~6D/5

Fig.2 Measuring probe

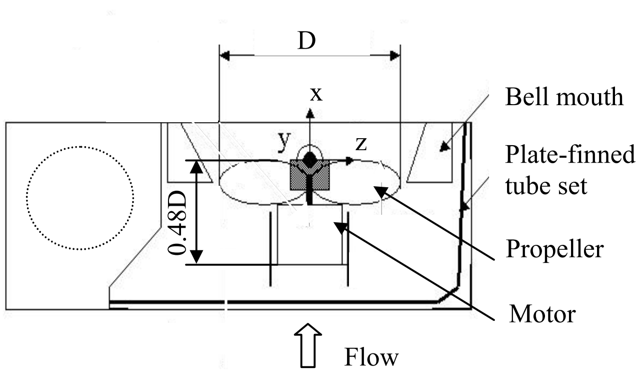


Fig.1 Flow field and coordinate system (Top view)

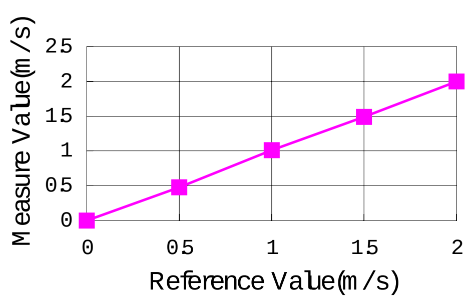
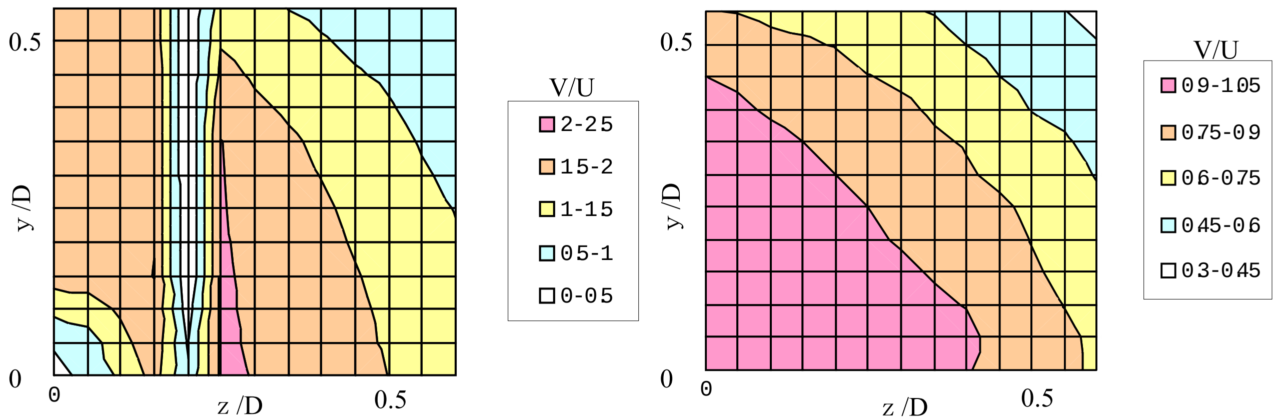


Fig.3 The probe response



(a) x/D=0.5 (b) x/D=1

Fig. 4 Velocity contour of fan for representative streamwise locations

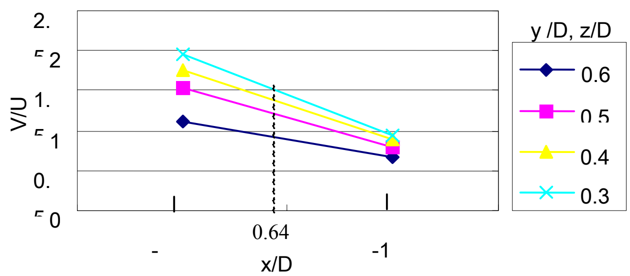
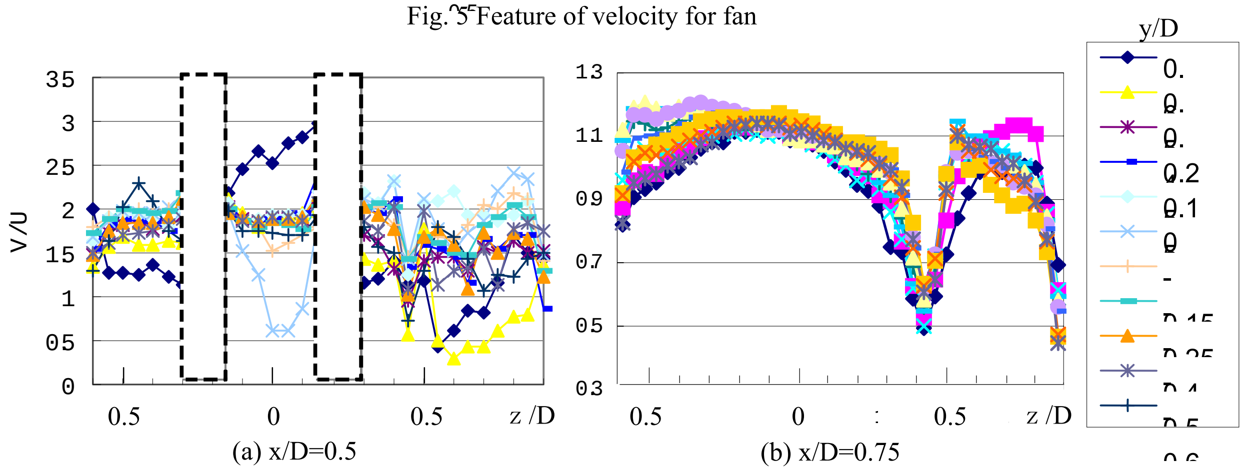


Fig. 5 Feature of velocity for fan



(a) x/D=0.5 (b) x/D=0.75

Fig.6 Spanwise variation of velocity for representative streamwise locations

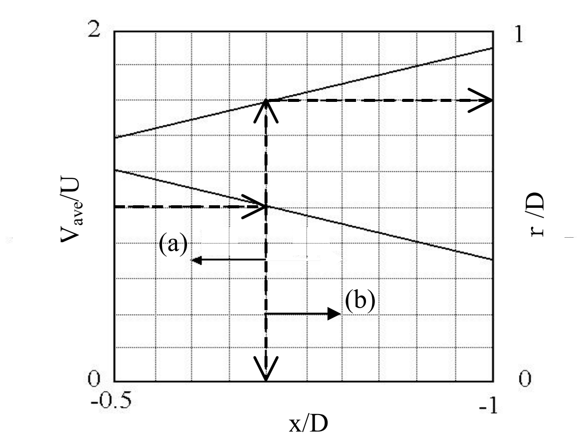


Fig. 7 Forced convection model

(a): forced convection intensity > V_e

(b): forced convection intensity < V_e