

DESCRIPTION

A METHOD FOR DESIGNING A MULTIBLADE RADIAL FAN  
AND

A MULTIBLADE RADIAL FAN

5 [TECHNICAL FIELD]

The present invention relates to a method for designing a multiblade radial fan and also relates to a multiblade radial fan.

[BACKGROUND ART]

10 The radial fan, one type of centrifugal fan, has both its blades and interblade channels directed radially and is thus simpler than other types of centrifugal fans such as the sirocco fan, which has forward-curved blades, and the turbo fan, which has backward-curved blades. The radial fan  
15 is expected to come into wide use as a component of various kinds of household appliances.

However, design criteria for enhancing the quietness of the radial fan have not been established. This is because the radial fan has been applied mainly for handling  
20 corrosive gases, gases including fine particles and the like, taking advantage of the fact that radial fans having only a few blades enable easy repair and cleaning of the interblade channels. Fans used for this purpose do not have to be especially quiet.

25 A number of design criteria have been proposed for enhancing the quietness of centrifugal fans. For example, Japanese Patent Laid-Open Publication Sho 56-6097, Japanese

Patent Laid-Open Publication Sho 56-92397, etc. propose elongating the interblade channels to prevent the air flow in the interblade channels from separating, flowing backward, etc. Japanese Patent Laid-Open Publication Sho 5 63-285295, Japanese Patent Laid-Open Publication Hei 2-33494, Japanese Patent Laid-Open Publication Hei 4-164196, etc. propose optimizing the number of blades of a sirocco fan with a large diameter ratio.

Japanese Patent Laid-Open Publication Sho 56-6097,  
10 Japanese Patent Laid-Open Publication Sho 56-92397, etc. disclose only the concept that the interblade channels should be elongated. They do not disclose any correlation which should be established among various fan specifications for optimizing the quietness of the fan. Thus, the  
15 proposals set out in Japanese Patent Laid-Open Publication Sho 56-6097, Japanese Patent Laid-Open Publication Sho 56-92397, etc. are not practical design criteria for obtaining a quiet fan.

The proposals of Japanese Patent Laid-Open Publication  
20 Sho 63-285295, Japanese Patent Laid-Open Publication Hei 2-33494, Japanese Patent Laid-Open Publication Hei 4-164196, etc. can be applied only to sirocco fans with large diameter ratios. Thus, they are not general purpose design criteria for obtaining a quiet fan.

25 [DISCLOSURE OF INVENTION]

The inventors of the present invention conducted an extensive study and found that there is a definite

correlation between the quietness of a multiblade radial fan and the specifications of the impeller of the multiblade radial fan. The present invention was accomplished based on this finding. The object of the present invention is  
5 therefore to provide methods for systematically determining the specifications of the impeller of a multiblade radial fan under a given condition, based on the above mentioned definite correlation, and optimizing the quietness of the multiblade radial fan. Another object of the present  
10 invention is to provide a multiblade radial fan designed based on the method of the present invention.

According to a first aspect of the present invention, there is provided a method for designing a multiblade radial fan, wherein specifications of the impeller of the  
15 multiblade radial fan are determined so as to satisfy the correlation expressed by the formula  $\nu \geq -0.857Z_1 + 1.009$  (in the formula,  $\nu = r_0/r_1$ ,  $Z_1 = (r_1 - r_0)/[r_1 - nt/(2\pi)]$ ,  $r_0$  : inside radius of the impeller,  $r_1$  : outside radius of the impeller,  $n$  : number of radially directed blades,  $t$  :  
20 thickness of the radially directed blades).

According to the first aspect of the present invention, there is also provided a method for designing a multiblade radial fan, wherein specifications of the impeller of the multiblade radial fan are determined so as  
25 to satisfy the correlation expressed by the formulas  $\nu \geq -0.857Z_1 + 1.009$  and  $0.8 \geq \nu \geq 0.4$  (in the formulas,  $\nu = r_0/r_1$ ,  $Z_1 = (r_1 - r_0)/[r_1 - nt/(2\pi)]$ ,  $r_0$  : inside radius of the

impeller,  $r_1$  : outside radius of the impeller,  $n$  : number of radially directed blades,  $t$  : thickness of the radially directed blades).

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According to the first aspect of the present invention, there is also provided a multiblade radial fan, wherein specifications of the impeller of the multiblade radial fan satisfy the correlation expressed by the formulas  $\nu \geq -0.857Z_1 + 1.009$  and  $0.8 \geq \nu \geq 0.4$  (in the formulas,  $\nu = r_0/r_1$ ,  $Z_1 = (r_1 - r_0) / [r_1 - nt / (2\pi)]$ ,  $r_0$  : inside radius of the impeller,  $r_1$  : outside radius of the impeller,  $n$  : number of radially directed blades,  $t$  : thickness of the radially directed blades).

According to a second aspect of the present invention, there is provided a method for designing a multiblade radial fan, wherein specifications of the impeller of the multiblade radial fan are determined so as to satisfy the correlation expressed by the formula  $(1.009 - \nu) / (1 - \nu) \leq Z_2$  (in the formula,  $\nu = r_0/r_1$ ,  $Z_2 = 0.857 \{t_0 / [(2\pi r_1/n) -$

$t]+1\}$  ,  $r_0$  : inside radius of the impeller,  $r_1$  : outside radius of the impeller,  $n$  : number of radially directed blades,  $t$  : thickness of the radially directed blades,  $t_0$  : reference thickness = 0.5mm).

5           According to the second aspect of the present invention, there is also provided a method for designing a multiblade radial fan, wherein specifications of the impeller of the multiblade radial fan are determined so as to satisfy the correlation expressed by the formulas  $(1.009 - \nu)/(1 - \nu) \leq Z_2$  and  $0.8 \geq \nu \geq 0.4$  (in the formulas,  $\nu = r_0/r_1$ ,  $Z_2 = 0.857 \{t_0/[(2\pi r_1/n)-t]+1\}$  ,  $r_0$  : inside radius of the impeller,  $r_1$  : outside radius of the impeller,  $n$  : number of radially directed blades,  $t$  : thickness of the radially directed blades,  $t_0$  : reference thickness =  
10           0.5mm).  
15           0.5mm).

          According to the second aspect of the present invention, there is also provided a multiblade radial fan, wherein specifications of the impeller of the multiblade radial fan satisfy the correlation expressed by the formula  
20            $(1.009 - \nu)/(1 - \nu) \leq Z_2$  (in the formula,  $\nu = r_0/r_1$ ,  $Z_2 = 0.857 \{t_0/[(2\pi r_1/n)-t]+1\}$  ,  $r_0$  : inside radius of the impeller,  $r_1$  : outside radius of the impeller,  $n$  : number of radially directed blades,  $t$  : thickness of the radially directed blades,  $t_0$  : reference thickness = 0.5mm).

25           According to the second aspect of the present invention, there is also provided a multiblade radial fan, wherein specifications of the impeller of the multiblade

radial fan satisfy the correlation expressed by the formulas  
 $(1.009 - \nu)/(1 - \nu) \leq Z_2$  and  $0.8 \geq \nu \geq 0.4$  (in the  
formulas,  $\nu = r_0/r_1$ ,  $Z_2 = 0.857 \{t_0/[(2\pi r_1/n)-t]+1\}$ ,  $r_0$  :  
inside radius of the impeller,  $r_1$  : outside radius of the  
5 impeller,  $n$  : number of radially directed blades,  $t$  :  
thickness of the radially directed blades,  $t_0$  : reference  
thickness = 0.5mm).

According to another aspect of the present invention,  
there is provided a multiblade radial fan comprising an  
10 impeller having many radially directed blades which are  
circumferentially spaced from each other so as to define  
narrow channels between them, wherein laminar boundary  
layers in the interblade channels are prevented from  
separating.

15 According to a preferred embodiment of the present  
invention, inner end portions of the radially directed  
blades are bent in the direction of rotation of the  
impeller.

#### [BRIEF DESCRIPTION OF THE DRAWINGS]

20 In the drawings:

Figure 1 is a plan view of a divergent channel showing  
the state of a laminar flow in the divergent channel.

Figure 2 is a plan view of divergent channels between  
radially directed blades of the impeller of a multiblade  
25 radial fan.

Figure 3 is an arrangement plan of a measuring  
apparatus for measuring air volume flow rate and static

pressure of a multiblade radial fan.

Figure 4 is an arrangement plan of a measuring apparatus for measuring the sound pressure level of a multiblade radial fan.

5        Figure 5(a) is a plan view of a tested impeller and Figure 5(b) is a sectional view taken along line b-b in Figure 5(a).

Figure 6 is a plan view of a tested casing.

Figure 7 shows experimentally obtained correlation  
10    diagrams between minimum specific sound level  $K_{s_{min}}$  and first Karman-Millikan nondimensional number  $Z_1$  of tested impellers.

Figure 8 is a correlation diagram between diameter ratio and threshold level of first Karman-Millikan  
15    nondimensional number  $Z_1$  of test-impellers.

Figure 9 shows experimentally obtained correlation diagrams between minimum specific sound level  $K_{s_{min}}$  and second Karman-Millikan nondimensional number  $Z_2$  of tested impellers.

20        Figure 10 is a correlation diagram between nondimensional number  $(1.009-r_0/r_1)/(1-r_0/r_1)$  and threshold level of second Karman-Millikan nondimensional number  $Z_2$  of tested impellers.

Figure 11 is a plan sectional view of another type of  
25    radially directed blade.

Figure 12(a) is a perspective view of a double intake multiblade radial fan to which the present invention can be

applied and Figure 12(b) is a sectional view taken along line b-b in Figure 12(a).

[THE BEST MODE FOR CARRYING OUT THE INVENTION]

Preferred embodiments of the present invention will be described.

《 1 》 First Aspect of the Invention

1. Theoretical background

When air flows through radially directed interblade channels of a rotating impeller, laminar boundary layers, which separate easily, develop on the suction surfaces of the blades of the impeller, and turbulent boundary layers, which do not separate easily, develop on the pressure surfaces of the blades of the impeller.

The separations of the laminar boundary layers cause secondary flows in the radially directed interblade channels of the impeller. The secondary flows cause noise and a drop in the efficiency of the impeller.

Thus, for designing a quiet multiblade radial fan, it is important to prevent the separations of the laminar boundary layers which develop on the suction surfaces of the blades.

The following formulas ①, ② have been given for expressing the state of a laminar boundary layer in a static divergent channel by Karman and Millikan (Von Karman, T., and Millikan, C.B., "On the Theory of Laminar Boundary Layers Involving Separation", NACA Rept. No. 504, 1934).

$$U/U_i = 1 \dots \dots \dots \textcircled{1} (0 \leq X/X_e \leq 1)$$

$$U/U_i = 1 + F(X - X_e)/X_e \dots \dots \textcircled{2} (1 \leq X/X_e)$$

In the above formulas, as shown in Figure 1,

X : distance from the fore end of a flat plate (virtual  
5 part)

X<sub>e</sub> : length of a flat plate (virtual part)

U : flow velocity outside of a laminar boundary layer at  
point X

U<sub>i</sub> : maximum flow velocity at point X

10 F :  $F = (X_e/U_i)(dU/dX)$

In the above formulas, the second term of the right  
side of the formula  $\textcircled{2}$  is a nondimensional term which  
expresses the state of the laminar boundary layer in the  
divergent channel. Thus, the second term of the right side  
15 of the formula  $\textcircled{2}$  can be effectively used for designing a  
quiet multiblade radial fan.

If the second term of the right side of the formula  $\textcircled{2}$   
is expressed as Z, and X - X<sub>e</sub> is expressed as x (x = X - X<sub>e</sub>), the  
nondimensional term Z is obtained as

20  $Z = (x/U_i)(dU/dx) \dots \dots \textcircled{3}$

It is fairly hard to obtain analytically or  
experimentally the flow velocity U outside of the laminar  
boundary layer at point X and the maximum flow velocity U<sub>i</sub>  
at point X. Thus, the flow velocity U outside of the  
25 laminar boundary layer at point X is replaced with the mean  
velocity U<sub>m</sub> at point X, and the maximum flow velocity U<sub>i</sub> at  
point X is replaced with the mean velocity U<sub>0</sub> at the inlet

of the divergent channel. Thus, the formula ③ is rewritten as

$$Z=(x/U_0)(dU_m/dx) \cdot \cdot \cdot \textcircled{4}$$

The nondimensional term  $Z$  defined by the formula ④ expresses the state of the laminar boundary layer in a static divergent channel. So, the formula ④ can not be applied directly to a laminar boundary layer in a rotating divergent channel.

Rotation of a divergent channel causes pressure gradient in the circumferential direction between the suction surface of a blade and the pressure surface of the adjacent blade. However, the circumferential pressure gradient between the suction surface of the blade and the pressure surface of the adjacent blade is small in an interblade channel of the impeller of a multiblade radial fan, wherein the ratio between chord length and pitch (distance between the adjacent blades) is large. That is, in the multiblade radial fan, wherein the ratio between chord length and pitch is large, the effect of the rotation on the state of the air flow in the interblade divergent channel is small. Thus, the nondimensional term  $Z$  defined by the formula ④ accurately approximates the state of the laminar boundary layer in the interblade divergent channel of a rotating multiblade radial fan and can be effectively used for designing a quiet multiblade radial fan.

The absolute value of the nondimensional term  $Z$ , defined by the formula ④, at the outer end or the outlet

of the interblade divergent channel of the multiblade radial fan is defined as  $Z_1$ . The term  $Z_1$  is expressed by formula ⑤. Hereinafter, the term  $Z_1$  is called Karman-Millikan's first nondimensional number.

5      $Z_1 = (r_1 - r_0) / [r_1 - nt / (2\pi)] \dots \textcircled{5}$

In the formula ⑤, as shown in Figure 2,

$r_0$  : inside radius of the impeller

$r_1$  : outside radius of the impeller

$n$  : number of radially directed blades

10     $t$  : thickness of the radially directed blades

## 2. Performance Test of Multiblade Radial Fan.

Performance tests were carried out on multiblade radial fans with different values of the term  $Z_1$ .

### [1] Test conditions

#### 15 (1) Measuring apparatuses

① Measuring apparatus for measuring air volume flow rate and static pressure

The measuring apparatus used for measuring air volume flow rate and static pressure is shown in Figure 3. The fan  
20 body had an impeller 1, a scroll type casing 2 for accommodating the impeller 1 and a motor 3. A inlet nozzle was disposed on the suction side of the fan body. A double chamber type air volume flow rate measuring apparatus (product of Rika Seiki Co. Ltd., Type F-401) was disposed on  
25 the discharge side of the fan body. The air volume flow rate measuring apparatus was provided with an air volume flow rate control damper and an auxiliary fan for

controlling the static pressure at the outlet of the fan body. The air flow discharged from the fan body was straightened by a straightening grid.

The air volume flow rate of the fan body was measured  
5 using orifices located in accordance with the AMCA standard.

The static pressure at the outlet of the fan body was measured through a static pressure measuring hole disposed near the outlet of the fan body.

② Measuring apparatus for measuring sound pressure level

10 The measuring apparatus for measuring sound pressure level is shown in Figure 4. A inlet nozzle was disposed on the suction side of the fan body. A static pressure control chamber of a size and shape similar to those of the air volume flow rate measuring apparatus was disposed on the  
15 discharge side of the fan body. The inside surface of the static pressure control chamber was covered with sound absorption material. The static pressure control chamber was provided with an air volume flow rate control damper for controlling the static pressure at the outlet of the fan  
20 body.

The static pressure at the outlet of the fan body was measured through a static pressure measuring hole located near the outlet of the fan body. The sound pressure level corresponding to a certain level of the static pressure at  
25 the outlet of the fan body was measured.

The motor 3 was installed in a soundproof box lined with sound absorption material. Thus, the noise generated

by the motor 3 was confined.

The measurement of the sound pressure level was carried out in an anechoic room. A-weighted sound pressure level was measured at a point on the centerline of the impeller and 1m above the upper surface of the casing.

## (2) Tested impellers, Tested Casing

### ① Tested impellers

As shown in Figures 5(a) and 5(b), the outside diameter and the height of all tested impellers were 100mm and 24mm respectively. The thickness of the circular base plate and the annular top plate of all tested impellers was 2mm. Impellers with four different inside diameters were made. Different impellers had a different number of radially directed flat plate blades disposed at equal circumferential distances from each other. A total of 21 kinds of impellers were made and tested. The particulars and the Karman-Millikan's first nondimensional numbers  $Z_1$  of the tested impellers are shown in Table 1, and Figures 5(a) and 5(b).

### ② Tested casing

As shown in Figure 3, the height of the scroll type casing 2 was 27mm. The divergence configuration of the scroll type casing 2 was a logarithmic spiral defined by the following formula. The divergence angle  $\theta_0$  was  $4.50^\circ$ .

$$r = r_2 [\exp (\theta \tan \theta_0)]$$

In the above formula,

$r$  : radius of the side wall of the casing measured from

the center of the impeller 1

$r_2$  : outside radius of the impeller 1

$\theta$  : angle measured from a base line,  $0 \leq \theta \leq 2\pi$

$\theta_0$  : divergence angle

5        The tested casing 2 is shown in Figure 6.

③ Revolution speed of the impeller 1

      The revolution speed of the impeller 1 was generally fixed at 6000 rpm but was varied to a certain extent considering extrinsic factors such as background noise in  
10 the anechoic room, condition of the measuring apparatus, etc. The revolution speeds of the impeller 1 during measurement are shown in Table 1.

[2] Measurement, Data Processing

(1) Measurement

15        The air volume flow rate of the air discharged from the fan body, the static pressure at the outlet of the fan body, and the sound pressure level were measured for each of the 21 kinds of the impellers 1 shown in Table 1 when rotated at the revolution speed shown in Table 1, while the  
20 air volume flow rate of the air discharged from the fan body was varied using the air volume flow rate control dampers.

(2) Data Processing

      From the measured value of the air volume flow rate of the air discharged from the fan body, the static pressure  
25 at the outlet of the fan body, and the sound pressure level, a specific sound level  $K_s$  defined by the following formula was obtained.

$$K_s = \text{SPL}(A) - 10 \log_{10} Q(P_t)^2$$

In the above formula,

$\text{SPL}(A)$  : A-weighted sound pressure level, dB

$Q$  : air volume flow rate of the air discharged from the  
5 fan body,  $\text{m}^3/\text{s}$

$P_t$  : total pressure at the outlet of the fan body,  $\text{mmAq}$

### 3. Test Results

Based on the results of the measurements, a  
correlation between the specific sound level  $K_s$  and the air  
10 volume flow rate was obtained for each tested impeller 1.

The correlation between the specific sound level  $K_s$   
and the air volume flow rate  $Q$  was obtained on the  
assumption that a correlation wherein the specific sound  
level  $K_s$  is  $K_{s1}$  when the air volume flow rate  $Q$  is  $Q_1$   
15 exists between the specific sound level  $K_s$  and the air  
volume flow rate  $Q$  when the air volume flow rate  $Q$  and the  
static pressure  $p$  at the outlet of the fan body obtained by  
the air volume flow rate and static pressure measurement  
are  $Q_1$  and  $p_1$  respectively, while the specific sound level  $K$   
20  $s$  and the static pressure  $p$  at the outlet of the fan body  
obtained by the sound pressure level measurement are  $K_{s1}$  and  
 $p_1$  respectively. The above assumption is thought to be  
reasonable as the size and the shape of the air volume flow  
rate measuring apparatus used in the air volume flow rate  
25 and static pressure measurement are substantially the same  
as those of the static pressure controlling box used in the  
sound pressure level measurement.

The measurement showed that the specific sound level  $K_s$  of each tested impeller 1 varied with variation in the air volume flow rate. The variation of the specific sound level  $K_s$  is generated by the effect of the casing 2. Thus, it can be assumed that the minimum value of the specific sound level  $K_s$  or the minimum specific sound level  $K_{s_{min}}$  represents the noise characteristic of the tested impeller 1 itself free from the effect of the casing 2.

The minimum specific sound levels  $K_{s_{min}}$  of the tested impellers 1 are shown in Table 1. Correlations between the minimum specific sound levels  $K_{s_{min}}$  and the Karman-Millikan's first nondimensional numbers  $Z_1$  of the tested impellers 1 are shown in Figure 7. Figure 7 also shows correlation diagrams between the minimum specific sound level  $K_{s_{min}}$  and the Karman-Millikan's first nondimensional number  $Z_1$  of each group of the impellers 1 having the same diameter ratio.

As is clear from Figure 7, for the same diameter ratio of the impeller 1, the minimum specific sound level  $K_{s_{min}}$  decreased as the Karman-Millikan's first nondimensional number  $Z_1$  increased. It is also clear from the correlation diagrams shown in Figure 7 that in the groups of the impellers 1 with diameter ratios of 0.75, 0.58 and 0.4, the minimum specific sound level  $K_{s_{min}}$  stayed at a constant minimum value when the Karman-Millikan's first nondimensional number  $Z_1$  became larger than a certain threshold value. The reason why the minimum specific sound

- level  $K_{s_{min}}$  stays at a constant minimum value when the Karman-Millikan's first nondimensional number  $Z_1$  becomes larger than a certain threshold value is thought to be that the increase in the number of the blades causes the
- 5 interblade channels to become more slender, thereby suppressing the separations of the laminar boundary layers in the interblade channels. An analysis using differential calculus was carried out on the air flow in the interblade channel of an impeller 1 with a diameter ratio of 0.58.
- 10 From the analysis, it was confirmed that a separation does not occur in the laminar boundary layer at the measuring point on the horizontal part of the correlation diagram in Figure 7 where  $Z_1$  is 0.5192, while a separation occurs in the laminar boundary layer at the measuring point on the
- 15 inclined part of the correlation diagram in Figure 7 where  $Z_1$  is 0.4813.

- As to the group of the impellers 1 with diameter ratios of 0.90, the threshold value of  $Z_1$  is not clear because the number of the measured points was small. In
- 20 Figure 7, the correlation diagram of the group of the impellers 1 with diameter ratios of 0.90 is assigned a threshold value of  $Z_1$  estimated from the threshold values of  $Z_1$  of the correlation diagrams of other groups of the impellers 1.

- 25 Correlations between the diameter ratio  $\nu$  of the impeller 1 and the threshold value of the Karman-Millikan's first nondimensional number  $Z_1$  were obtained from the

correlation diagrams between the minimum specific sound level  $K_{s_{min}}$  and the Karman-Millikan's first nondimensional number  $Z_1$  of the groups of the impellers 1 with diameter ratios of 0.75, 0.58 and 0.4. The correlations are shown in Figure 8. From Figure 8, there was obtained a correlation diagram  $L_1$  between the diameter ratio  $\nu$  of the impeller 1 and the threshold value of the Karman-Millikan's first nondimensional number  $Z_1$ . The correlation diagram  $L_1$  is defined by the following formula ⑥.

10  $\nu = -0.857Z_1 + 1.009 \dots \textcircled{6}$

In the above formula,

$$\nu = r_0/r_1$$

$$Z_1 = (r_1 - r_0) / [r_1 - nt / (2\pi)]$$

The correlation diagram  $L_1$  can be applied to impellers 1 with diameter ratio  $\nu$  ranging from 0.40 to 0.75. As is clear from Figure 8, the correlation diagram  $L_1$  is straight.

Therefore, there should be practically no problem in applying the correlation diagram  $L_1$  to impellers with diameter ratio  $\nu$  ranging from 0.30 to 0.90.

20 As shown in Figure 8, the hatched area to the right of the correlation diagram  $L_1$  is the quiet region wherein the minimum specific sound level  $K_{s_{min}}$  of an impeller 1 of diameter ratio  $\nu$  stays at a constant minimum value. Thus, the quietness of a multiblade radial fan can be optimized systematically, without resorting to trial and error, by  
25 determining the specifications of the impeller of diameter ratio  $\nu$  so that the Karman-Millikan's first

nondimensional number  $Z_1$  falls in the hatched region in Figure 8, or satisfies the correlation defined by formula ⑦.

$$\nu \geq -0.857Z_1 + 1.009 \dots \textcircled{7}$$

5 In the above formula,

$$\nu = r_0/r_1$$

$$Z_1 = (r_1 - r_0) / [r_1 - nt / (2\pi)]$$

$r_0$  : inside radius of the impeller

$r_1$  : outside radius of the impeller

10  $n$  : number of the radially directed blades

$t$  : thickness of the radially directed blades

Figure 8 also shows the correlation between the diameter ratio  $\nu$  of an impeller 1 with a diameter ratio of 0.90 and the threshold value of the Karman-Millikan's first  
15 nondimensional number  $Z_1$  which is obtained from the correlation diagram shown in Figure 7. As is clear from Figure 8, the correlation between the diameter ratio  $\nu$  of the impeller 1 with a diameter ratio of 0.90 and the threshold value of the Karman-Millikan's first  
20 nondimensional number  $Z_1$  falls on the correlation diagram  $L_1$ .

As will be understood from the above description, the quietness of a multiblade radial fan whose diameter ratio is in the range of from 0.30 to 0.90 can be optimized based on  
25 the formula ⑦. However, as shown in Figure 7, the minimum value of the minimum specific sound level  $K_{smin}$  of an impeller with a diameter ratio  $\nu$  of 0.90 is about 43dB.

In other words, an impeller with a diameter ratio  $\nu$  of 0.90 cannot be made sufficiently quiet. On the other hand, an impeller with a diameter ratio  $\nu$  of 0.30 cannot easily be equipped with many radial blades because of the small  
5 inside radius. It is therefore appropriate to apply the formula ⑦ to impellers with diameter ratios  $\nu$  in the range of from 0.40 to 0.80. Thus, a multiblade radial fan that achieves optimum and sufficient quietness under a given condition and is easy to fabricate can be designed  
10 systematically, without resorting to trial and error, by applying the formula ⑦ to an impeller whose diameter ratio  $\nu$  falls in the range of from 0.40 to 0.80.

As is clear from the formula ⑤, the Karman-Millikan's first nondimensional number  $Z_1$  includes the term  
15 "n" (number of the radially directed blades) and the term "t" (thickness of the radially directed blade) in the form of the product "nt". Thus, the term "n" and the term "t" cannot independently contribute to the optimization of the quietness of the multiblade radial fan. Thus, in  
20 accordance with the first aspect of the invention, the quietness of a multiblade radial fan wherein  $n=100$ ,  $t=0.5\text{mm}$  should be equal to that of a multiblade radial fan wherein  $n=250$ ,  $t=0.2\text{mm}$  because the products "nt" are equal, making the Karman-Millikan's first nondimensional number  $Z_1$  of the  
25 former fan equal to that of the latter. In fact, however, there is some difference in the quietness between the two because of the difference in the shape of the interblade

channels between the two. Therefore, the quietness of a multiblade radial fan should preferably be optimized in accordance with the first aspect of the invention by:

- (1) determining the design value  $Z_1$ , of the the Karman-Millikan's first nondimensional number  $Z_1$  which optimizes the quietness of the multiblade radial fan in accordance with the formula ⑦, and
- (2) selecting the best combination of "n" and "t" from the plurality of combinations of "n" and "t" which achieve the design value  $Z_1$ , based on a sound pressure level measurement.

## 《 2 》 Second Aspect of the Invention

### 1. Theoretical background

As explained above, the first aspect of the invention has a shortcoming in that the term "n" and the term "t" cannot independently contribute to the optimization of the quietness of a multiblade radial fan.

This problem can be overcome by optimizing the quietness of the multiblade radial fan based on a nondimensional number which includes the terms "n" and "t" independently.

For this end, the formula ⑦ is rewritten by replacing the constant values -0.857 and 1.009 with "a" and "b" respectively and then converting it to

$$r_0/r_1 \geq a(r_1 - r_0)/[r_1 - nt/(2\pi)] + b \dots \textcircled{8}$$

A formula ⑨ is derived from the formula ⑧.

$$2\pi r_1 - nt \leq -a(2\pi r_1) [(1 - r_0/r_1)/(b - r_0/r_1)] \dots \textcircled{9}$$

A formula ⑩ is derived from the formula ⑨.

$$(2\pi r_1/n)-t \leq -a(2\pi r_1) [(1-r_0/r_1)/(b-r_0/r_1)]/n \dots \textcircled{10}$$

The term  $(2\pi r_1/n)-t$  making up the left side of the formula ⑩ is the outlet breadth  $\Delta \ell$  of an interblade divergent channel. Thus, the first aspect of the invention indicates that the quietness of a multiblade radial fan is optimized when the outlet breadth  $\Delta \ell$  of the interblade divergent channel satisfies the formula ⑩.

When the left side is equal to the right side in the formula ⑩, the number  $n_c$  of the radially directed blades and the outlet breadth  $\Delta \ell_c$  of the interblade divergent channel are expressed as follows.

$$\begin{aligned} n_c &= (2\pi r_1/t)[1+a(1-r_0/r_1)/(b-r_0/r_1)] \\ \Delta \ell_c &= (2\pi r_1/n_c)-t \\ &= -a[(1-r_0/r_1)/(b-r_0/r_1)]t / [1+a(1-r_0/r_1)/(b-r_0/r_1)] \\ &= -at / [(b-r_0/r_1)/(1-r_0/r_1)+a] \end{aligned}$$

As can be seen from Table 1, the measurements for deriving the first aspect of the invention were carried out mainly on impellers whose blades are 0.5mm thick. Thus, when the thickness "t" of the radially directed blades is "t<sub>0</sub>" (t<sub>0</sub>=0.5mm), the quietness of the multiblade radial fan is optimized provided the outlet breadth  $\Delta \ell$  of the interblade divergent channel satisfies

$$\begin{aligned} \Delta \ell &= (2\pi r_1/n)-t_0 \leq \Delta \ell_c = -at_0 / [(b-r_0/r_1)/(1-r_0/r_1)+a] \\ \text{That is,} \\ (2\pi r_1/n)-t_0 &\leq -at_0 / [(b-r_0/r_1)/(1-r_0/r_1)+a] \dots \textcircled{11} \end{aligned}$$

In the above formula, t<sub>0</sub>=0.5mm.

Now, the following assumption is introduced : even though the thickness "t" of the radially directed blades is not equal to "t<sub>0</sub>" ( t<sub>0</sub>=0.5mm), the quietness of the multiblade radial fan is optimized if the outlet breadth  $\Delta \ell$  of the interblade divergent channel is smaller than the threshold value  $\Delta \ell_0$  of the outlet breadth  $\Delta \ell$  of the interblade divergent channel where the thickness "t" of the radially directed blades is equal to "t<sub>0</sub>" ( t<sub>0</sub>=0.5mm).

Under the above assumption, the condition for optimizing the quietness of the multiblade radial fan is

$$(2\pi r_1/n) - t \leq -at_0/[(b-r_0/r_1)/(1-r_0/r_1)+a] \dots \textcircled{12}$$

In the above formula, t<sub>0</sub>=0.5mm.

A formula  $\textcircled{13}$  is derived from the formula  $\textcircled{12}$ .

$$(b-r_0/r_1)/(1-r_0/r_1) \leq -a \{t_0/[(2\pi r_1/n) - t] + 1\} \dots \textcircled{13}$$

Hereinafter, the right side of the formula  $\textcircled{13}$  is called Karman-Millikan's second nondimensional number Z<sub>2</sub>. The Karman-Millikan's second nondimensional number Z<sub>2</sub> includes the number "n" and the thickness "t" of the radially directed blades independently. Thus, the Karman-Millikan's second nondimensional number Z<sub>2</sub> does not include the problem of the Karman-Millikan's first nondimensional number Z<sub>1</sub>.

The formula  $\textcircled{13}$  is expressed as follows by using the Karman-Millikan's second nondimensional number Z<sub>2</sub>.

$$(b-r_0/r_1)/(1-r_0/r_1) \leq Z_2 \dots \textcircled{14}$$

In the above formula,

$$Z_2 = -a \{t_0/[(2\pi r_1/n) - t] + 1\}$$

$$a = -0.857$$

$$b = 1.009$$

$t_0$  : specific thickness of the radially directed blades  
= 0.5mm

5  $r_0$  : inside radius of the impeller

$r_1$  : outside radius of the impeller

$n$  : number of the radially directed blades

$t$  : thickness of the radially directed blades

Thus, if tests show that the quietness of a multiblade  
10 radial fan is optimized when the Karman-Millikan's second  
nondimensional number  $Z_2$  satisfies the formula (14), a second  
aspect of the invention is established wherein the  
specifications of a multiblade radial fan are determined  
based on the formula (14). The second aspect of the  
15 invention is more generalized than the first aspect of the  
invention wherein the specifications of a multiblade radial  
fan are determined based on the formula (7).

## 2. Performance Test of Multiblade Radial Fan.

Performance tests were carried out on multiblade  
20 radial fans with different values of the term  $Z_2$  in the  
same way as described earlier in connection with the first  
aspect of the invention. The particulars, Karman-Millikan's  
first nondimensional number  $Z_1$ , Karman-Millikan's second  
nondimensional numbers  $Z_2$ , the minimum specific sound  
25 levels  $K_{s_{min}}$ , and the rotation speeds of the tested  
impellers are listed in Table 2. The measured correlations  
between the minimum specific sound levels  $K_{s_{min}}$  and the

Karman-Millikan's second nondimensional numbers  $Z_2$  of the tested impellers are shown in Figure 9. A correlation diagram between the minimum specific sound level  $K_{smin}$  and the Karman-Millikan's second nondimensional number  $Z_2$  was  
5 obtained for each group of impellers with the same diameter ratio. The correlation diagrams are also shown in Figure 9.

As is clear from Figure 9, for the same impeller diameter ratio, the minimum specific sound level  $K_{smin}$   
10 decreases as the Karman-Millikan's second nondimensional number  $Z_2$  increases. As is clear from the correlation diagrams in Figure 9, in the impellers 1 with diameter ratios of 0.75, 0.58 and 0.4, the minimum specific sound levels  $K_{smin}$  stay at constant minimum values when the  
15 Karman-Millikan's second nondimensional numbers  $Z_2$  exceed certain threshold values. Though the threshold value of the impeller 1 with a diameter ratio of 0.90 is not clear owing to the small number of measured points, a correlation diagram of the impeller 1 with a diameter ratio of 0.90  
20 having a threshold value estimated from those of the other correlation diagrams is also shown in Figure 9.

The formula (14) is shown in Figure 10. The hatched area on the right of the correlation diagram  $L_2$  is the assumed quiet region.

25 Correlations between the nondimensional numbers  $(b - r_0/r_1)/(1 - r_0/r_1)$  derived from the specifications of the impellers and the threshold values of the Karman-Millikan's

second nondimensional numbers  $Z_2$  were obtained from the correlation diagrams, shown in Figure 9, between the minimum specific sound levels  $K_{s_{min}}$  and the Karman-Millikan's second nondimensional numbers  $Z_2$  of the groups of the impellers with diameter ratios of 0.75, 0.58 and 0.4. The correlations are shown in Figure 10. As is clear from Figure 10, the experimentally obtained correlations between the nondimensional numbers  $(b-r_0/r_1)/(1-r_0/r_1)$  derived from the specifications of the impellers and the threshold values of the Karman-Millikan's second nondimensional numbers  $Z_2$  fall on the correlation diagram  $L_2$ . A correlation between the nondimensional number  $(b-r_0/r_1)/(1-r_0/r_1)$  and the threshold value of the Karman-Millikan's second nondimensional number  $Z_2$  of the impeller with a diameter ratio of 0.90 was obtained from the correlation diagram shown in Figure 9. This is also shown in Figure 10. As is clear from Figure 10, the correlation between the nondimensional number  $(b-r_0/r_1)/(1-r_0/r_1)$  and the threshold value of the Karman-Millikan's second nondimensional number  $Z_2$  of the impeller with a diameter ratio of 0.90 also falls on the correlation diagram  $L_2$ .

Thus, it was experimentally confirmed that the quietness of a multiblade radial fan is optimized when the Karman-Millikan's second nondimensional number  $Z_2$  satisfies the formula ⑭.

Thus, the quietness of a multiblade radial fan with a given impeller diameter ratio, can be optimized

systematically, without resorting to trial and error, by determining the specifications of the impeller so that the Karman-Millikan's second nondimensional number  $Z_2$  falls in the hatched region in Figure 10, or satisfies the  
5 correlation defined by formula (14).

The formula (14) can be applied to impellers with diameter ratios in the range of from 0.40 to 0.90. As shown in Figure 9, However, the minimum value of the minimum specific sound level  $K_{s_{min}}$  of the impeller with a diameter  
10 ratio of 0.90 is about 43dB. In other words, an impeller with a diameter ratio of 0.90 cannot be made sufficiently quiet. It is therefore appropriate to apply the formula (14) to impellers with diameter ratios in the range of from 0.40 to 0.80.

15 Thus, a multiblade radial fan that achieves optimum and sufficient quietness under a given condition can be designed systematically, without resorting to trial and error, by applying the formula (14) to an impeller whose diameter ratio falls in the range from 0.40 to 0.80.

20 Radially directed plate blades are used in the above embodiments. As shown in Figure 11, the inner end portions of the radially directed plate blades can be bent in the direction of rotation of the impeller to decrease the inlet angle of the air flow against the radially directed plate  
25 blades. This prevents the generation of turbulence in the air flow on the suction side of the inner end portion of the radially directed plate blades and further enhances the

quietness of the multiblade radial fan. The bend can be made on every blade, or at intervals of a predetermined number of blades.

The present invention can be applied to a double suction type multiblade radial fan such as the fan 10 shown in Figures 12(a) and 12(b). The double suction type multiblade radial fan 10 has a cup shaped circular base plate 11, a pair of annular plates 12a, 12b disposed on the opposite sides of the base plate 11, a large number of radially directed plate blades 13a disposed between the base plate 11 and the annular plate 12a, and a large number of radially directed plate blades 13b disposed between the base plate 11 and the annular plate 12b.

Multiblade radial fans in accordance with the present invention can be used in various kinds of apparatuses in which centrifugal fans such as sirocco fans and turbo fans, and cross flow fans, etc. have heretofore been used and, specifically, can be used in such apparatuses as hair driers, hot air type driers, air conditioners, air purifiers, office automation equipments, dehumidifiers, deodorization apparatuses, humidifiers, cleaning machines and atomizers.

#### [INDUSTRIAL APPLICABILITY]

According to the first aspect of the present invention, the specifications of the impeller of a multiblade radial fan are determined so as to satisfy the correlation expressed by the formula  $\nu \geq -0.857Z_1 + 1.009$  (in

the formula,  $\nu = r_0/r_1$ ,  $Z_1 = (r_1 - r_0)/[r_1 - nt/(2\pi)]$ ,  $r_0$  :  
inside radius of the impeller,  $r_1$  : outside radius of the  
impeller,  $n$  : number of radially directed blades,  $t$  :  
thickness of the radially directed blades ), whereby the  
5 minimum specific sound level of the multiblade radial fan  
is minimized. Thus, in accordance with the first aspect of  
the present invention, a multiblade radial fan that  
achieves optimum quietness under a given condition can be  
designed systematically, without resorting to trial and  
10 error.

According to a modification of the first aspect of the  
present invention, specifications of the impeller of a  
multiblade radial fan are determined so as to satisfy the  
correlation expressed by the formulas  $\nu \geq -0.857Z_1 + 1.009$  and  
15  $0.8 \geq \nu \geq 0.4$  (in the formulas,  $\nu = r_0/r_1$ ,  $Z_1 = (r_1 - r_0)/[r_1 -$   
 $nt/(2\pi)]$ ,  $r_0$  : inside radius of the impeller,  $r_1$  :  
outside radius of the impeller,  $n$  : number of radially  
directed blades,  $t$  : thickness of the radially directed  
blades ), whereby the minimum specific sound level of the  
20 multiblade radial fan is minimized. Thus, in accordance  
with the modification of the first aspect of the present  
invention, a multiblade radial fan that achieves optimum  
and sufficient quietness under a given condition and can be  
easily fabricated can be designed systematically, without  
25 resorting to trial and error.

According to the second aspect of the present  
invention, specifications of the impeller of a multiblade

radial fan are determined so as to satisfy the correlation expressed by the formula  $(1.009 - \nu)/(1 - \nu) \leq Z_2$  (in the formula,  $\nu = r_0/r_1$ ,  $Z_2 = 0.857 \{t_0/[(2\pi r_1/n)-t]+1\}$ ,  $r_0$ : inside radius of the impeller,  $r_1$ : outside radius of the  
5 impeller,  $n$ : number of radially directed blades,  $t$ : thickness of the radially directed blades,  $t_0$ : reference thickness = 0.5mm), whereby the minimum specific sound level of the multiblade radial fan is minimized. Thus, in accordance with the second aspect of the present invention,  
10 a multiblade radial fan that achieves optimum quietness under a given condition can be designed systematically, without resorting to trial and error.

According to a modification of the second aspect of the present invention, there is provided a method for  
15 designing a multiblade radial fan, wherein specifications of the impeller of a multiblade radial fan are determined so as to satisfy the correlation expressed by the formulas  $(1.009 - \nu)/(1 - \nu) \leq Z_2$  and  $0.8 \geq \nu \geq 0.4$  (in the formulas,  $\nu = r_0/r_1$ ,  $Z_2 = 0.857 \{t_0/[(2\pi r_1/n)-t]+1\}$ ,  $r_0$ :  
20 inside radius of the impeller,  $r_1$ : outside radius of the impeller,  $n$ : number of radially directed blades,  $t$ : thickness of the radially directed blades,  $t_0$ : reference thickness = 0.5mm), whereby the minimum specific sound level of the multiblade radial fan is minimized. Thus, in  
25 accordance with the modification of the second aspect of the present invention, a multiblade radial fan that achieves optimum and sufficient quietness under a given condition and

can be easily fabricated can be designed systematically, without resorting to trial and error.

The inner end portions of the radially directed plate blades can be bent in the direction of rotation of the  
5 impeller to decrease the inlet angle of the air flow against the radially directed plate blades. This prevents the generation of turbulence in the air flow on the suction side of the inner end portion of the radially directed plate blades and further enhances the quietness of the  
10 multiblade radial fan. The bend can be made on every blade, or at intervals of a predetermined number of blades.

The present invention can be applied to a double suction type multiblade radial fan.

Multiblade radial fans in accordance with the present  
15 invention can be used in various kinds of apparatuses in which centrifugal fans such as sirocco fans and turbo fans, and cross flow fans, etc. have heretofore been used, specifically in such apparatuses as hair driers, hot air type driers, air conditioners, air purifiers, office  
20 automation equipments, dehumidifiers, deodorization apparatuses, humidifiers, cleaning machines and atomizers.

T A B L E 1

| impeller<br>NO.       | outside<br>dia-<br>meter<br>(mm) | inside<br>dia-<br>meter<br>(mm) | thick-<br>ness of<br>radially<br>directed<br>blades<br>(mm) | number<br>of<br>radially<br>directed<br>blades | Z <sub>1</sub> | k <sub>z</sub> min<br>(dB) | revolution<br>speed<br>(rpm) |
|-----------------------|----------------------------------|---------------------------------|-------------------------------------------------------------|------------------------------------------------|----------------|----------------------------|------------------------------|
| diameter ratio : 0.90 |                                  |                                 |                                                             |                                                |                |                            |                              |
| 1                     | 100.0                            | 90.0                            | 0.5                                                         | 100                                            | 0.1189         | 46.0                       | 6000.0                       |
| 2                     | 100.0                            | 90.0                            | 0.5                                                         | 120                                            | 0.1236         | 47.3                       | 5000.0                       |
| 3                     | 100.0                            | 90.0                            | 0.5                                                         | 240                                            | 0.1618         | 43.0                       | 5000.0                       |
| diameter ratio : 0.75 |                                  |                                 |                                                             |                                                |                |                            |                              |
| 4                     | 100.0                            | 75.0                            | 0.5                                                         | 40                                             | 0.2670         | 47.4                       | 3000.0                       |
| 5                     | 100.0                            | 75.0                            | 0.5                                                         | 60                                             | 0.2764         | 41.8                       | 6000.0                       |
| 6                     | 100.0                            | 75.0                            | 0.5                                                         | 80                                             | 0.2865         | 40.3                       | 6000.0                       |
| 7                     | 100.0                            | 75.0                            | 0.5                                                         | 100                                            | 0.2973         | 38.7                       | 5000.0                       |
| 8                     | 100.0                            | 75.0                            | 0.5                                                         | 120                                            | 0.3090         | 39.8                       | 7200.0                       |
| 9                     | 100.0                            | 75.0                            | 0.5                                                         | 144                                            | 0.3243         | 39.2                       | 7200.0                       |
| 10                    | 100.0                            | 75.0                            | 0.3                                                         | 300                                            | 0.3504         | 38.7                       | 6000.0                       |
| diameter ratio : 0.58 |                                  |                                 |                                                             |                                                |                |                            |                              |
| 11                    | 100.0                            | 58.0                            | 0.5                                                         | 10                                             | 0.4268         | 45.0                       | 5000.0                       |
| 12                    | 100.0                            | 58.0                            | 0.5                                                         | 40                                             | 0.4486         | 42.1                       | 6000.0                       |
| 13                    | 100.0                            | 58.0                            | 0.5                                                         | 60                                             | 0.4643         | 40.1                       | 5000.0                       |
| 14                    | 100.0                            | 58.0                            | 0.5                                                         | 80                                             | 0.4813         | 38.7                       | 6000.0                       |
| 15                    | 100.0                            | 58.0                            | 0.5                                                         | 100                                            | 0.4995         | 36.2                       | 6000.0                       |
| 16                    | 100.0                            | 58.0                            | 0.5                                                         | 120                                            | 0.5192         | 33.4                       | 8000.0                       |
| 17                    | 100.0                            | 58.0                            | 0.3                                                         | 144                                            | 0.4870         | 33.4                       | 7200.0                       |
| diameter ratio : 0.40 |                                  |                                 |                                                             |                                                |                |                            |                              |
| 18                    | 100.0                            | 40.0                            | 0.5                                                         | 40                                             | 0.6408         | 37.0                       | 6000.0                       |
| 19                    | 100.0                            | 40.0                            | 0.5                                                         | 100                                            | 0.7136         | 35.7                       | 6000.0                       |
| 20                    | 100.0                            | 40.0                            | 0.3                                                         | 120                                            | 0.6777         | 33.3                       | 5000.0                       |
| 21                    | 100.0                            | 40.0                            | 0.5                                                         | 120                                            | 0.7416         | 33.3                       | 6000.0                       |

T A B L E 2

| impeller<br>NO.       | outside<br>dia-<br>meter<br>(mm) | inside<br>dia-<br>meter<br>(mm) | thick-<br>ness of<br>radially<br>directed<br>blades<br>(mm) | number<br>of<br>radially<br>directed<br>blades | Z <sub>1</sub> | Z <sub>2</sub> | k <sub>smi</sub> <sup>a</sup><br>(dB) | revo-<br>lution<br>speed<br>(rpm) |
|-----------------------|----------------------------------|---------------------------------|-------------------------------------------------------------|------------------------------------------------|----------------|----------------|---------------------------------------|-----------------------------------|
| diameter ratio : 0.90 |                                  |                                 |                                                             |                                                |                |                |                                       |                                   |
| 1                     | 100.0                            | 90.0                            | 0.5                                                         | 100                                            | 0.119          | 1.019          | 46.0                                  | 6000.0                            |
| 2                     | 100.0                            | 90.0                            | 0.5                                                         | 120                                            | 0.124          | 1.059          | 47.3                                  | 5000.0                            |
| 3                     | 100.0                            | 90.0                            | 0.5                                                         | 240                                            | 0.162          | 1.387          | 43.0                                  | 5000.0                            |
| diameter ratio : 0.75 |                                  |                                 |                                                             |                                                |                |                |                                       |                                   |
| 4                     | 100.0                            | 75.0                            | 0.5                                                         | 40                                             | 0.267          | 0.915          | 47.4                                  | 3000.0                            |
| 5                     | 100.0                            | 75.0                            | 0.5                                                         | 60                                             | 0.276          | 0.947          | 41.8                                  | 6000.0                            |
| 6                     | 100.0                            | 75.0                            | 0.5                                                         | 80                                             | 0.286          | 0.982          | 40.3                                  | 6000.0                            |
| 7                     | 100.0                            | 75.0                            | 0.5                                                         | 100                                            | 0.297          | 1.019          | 38.7                                  | 5000.0                            |
| 8                     | 100.0                            | 75.0                            | 0.5                                                         | 120                                            | 0.309          | 1.059          | 39.8                                  | 7200.0                            |
| 9                     | 100.0                            | 75.0                            | 0.5                                                         | 144                                            | 0.324          | 1.112          | 37.6                                  | 7200.0                            |
| 10                    | 100.0                            | 75.0                            | 0.3                                                         | 300                                            | 0.350          | 1.430          | 38.7                                  | 6000.0                            |
| diameter ratio : 0.58 |                                  |                                 |                                                             |                                                |                |                |                                       |                                   |
| 11                    | 100.0                            | 58.0                            | 0.5                                                         | 10                                             | 0.427          | 0.871          | 45.0                                  | 5000.0                            |
| 12                    | 100.0                            | 58.0                            | 2.0                                                         | 30                                             | 0.519          | 0.908          | 41.0                                  | 11200.0                           |
| 13                    | 100.0                            | 58.0                            | 0.5                                                         | 40                                             | 0.449          | 0.915          | 42.1                                  | 6000.0                            |
| 14                    | 100.0                            | 58.0                            | 0.3                                                         | 60                                             | 0.446          | 0.944          | 37.6                                  | 7000.0                            |
| 15                    | 100.0                            | 58.0                            | 0.5                                                         | 60                                             | 0.464          | 0.947          | 35.0                                  | 5000.0                            |
| 16                    | 100.0                            | 58.0                            | 1.0                                                         | 60                                             | 0.519          | 0.958          | 36.1                                  | 6000.0                            |
| 17                    | 100.0                            | 58.0                            | 0.3                                                         | 80                                             | 0.455          | 0.975          | 36.9                                  | 7000.0                            |
| 18                    | 100.0                            | 58.0                            | 0.5                                                         | 80                                             | 0.481          | 0.982          | 34.5                                  | 6000.0                            |
| 19                    | 100.0                            | 58.0                            | 0.3                                                         | 200                                            | 0.519          | 1.194          | 32.6                                  | 6000.0                            |
| 20                    | 100.0                            | 58.0                            | 0.5                                                         | 120                                            | 0.519          | 1.059          | 32.3                                  | 8000.0                            |
| 21                    | 100.0                            | 58.0                            | 0.3                                                         | 240                                            | 0.545          | 1.282          | 30.7                                  | 7000.0                            |
| 22                    | 100.0                            | 58.0                            | 0.3                                                         | 180                                            | 0.507          | 1.153          | 32.0                                  | 6000.0                            |
| 23                    | 100.0                            | 58.0                            | 0.5                                                         | 144                                            | 0.545          | 1.112          | 32.0                                  | 6000.0                            |
| diameter ratio : 0.40 |                                  |                                 |                                                             |                                                |                |                |                                       |                                   |
| 24                    | 100.0                            | 40.0                            | 0.5                                                         | 40                                             | 0.641          | 0.915          | 37.0                                  | 6000.0                            |
| 25                    | 100.0                            | 40.0                            | 0.5                                                         | 100                                            | 0.714          | 1.019          | 35.7                                  | 6000.0                            |
| 26                    | 100.0                            | 40.0                            | 0.3                                                         | 120                                            | 0.678          | 1.042          | 33.3                                  | 5000.0                            |
| 27                    | 100.0                            | 40.0                            | 0.5                                                         | 120                                            | 0.742          | 1.059          | 33.3                                  | 6000.0                            |



CLAIMS

- (1) A method for designing a multiblade radial fan, wherein specifications of an impeller of a multiblade radial fan are determined so as to satisfy a correlation expressed by a formula  $\nu \geq -0.857Z_1 + 1.009$  (in the formula,  $\nu = r_0/r_1$ ,  $Z_1 = (r_1 - r_0)/[r_1 - nt/(2\pi)]$ ,  $r_0$  : inside radius of the impeller,  $r_1$  : outside radius of the impeller,  $n$  : number of radially directed blades,  $t$  : thickness of the radially directed blades ).
- 5
- (2) A method for designing a multiblade radial fan, wherein specifications of an impeller of a multiblade radial fan are determined so as to satisfy a correlation expressed by formulas  $\nu \geq -0.857Z_1 + 1.009$  and  $0.8 \geq \nu \geq 0.4$  (in the formulas,  $\nu = r_0/r_1$ ,  $Z_1 = (r_1 - r_0)/[r_1 - nt/(2\pi)]$ ,  $r_0$  : inside
- 10
- radius of the impeller,  $r_1$  : outside radius of the impeller,  $n$  : number of radially directed blades,  $t$  : thickness of the radially directed blades ).
- 15
- (3) A multiblade radial fan, wherein specifications of an impeller of a multiblade radial fan satisfy a correlation expressed by a formula  $\nu \geq -0.857Z_1 + 1.009$  (in the formula,  $\nu = r_0/r_1$ ,  $Z_1 = (r_1 - r_0)/[r_1 - nt/(2\pi)]$ ,  $r_0$  : inside radius of the
- 20
- impeller,  $r_1$  : outside radius of the impeller,  $n$  : number of radially directed blades,  $t$  : thickness of the radially directed blades ).
- 25
- (4) A multiblade radial fan, wherein specifications of an impeller of a multiblade radial fan satisfy a correlation expressed by formulas  $\nu \geq -0.857Z_1 + 1.009$  and  $0.8 \geq \nu \geq 0.4$



(in the formulas,  $\nu = r_0/r_1$ ,  $Z_1 = (r_1 - r_0)/[r_1 - nt/(2\pi)]$ ,  
 $r_0$  : inside radius of the impeller,  $r_1$  : outside radius of  
the impeller,  $n$  : number of radially directed blades,  $t$  :  
thickness of the radially directed blades ).

5 (5) A method for designing a multiblade radial fan,  
wherein specifications of an impeller of a multiblade radial  
fan are determined so as to satisfy a correlation expressed  
by a formula  $(1.009 - \nu)/(1 - \nu) \leq Z_2$  (in the formula,  $\nu =$   
 $r_0/r_1$ ,  $Z_2 = 0.857 \{t_0/[(2\pi r_1/n) - t] + 1\}$ ,  $r_0$  : inside radius of  
10 the impeller,  $r_1$  : outside radius of the impeller,  $n$  :  
number of radially directed blades,  $t$  : thickness of the  
radially directed blades,  $t_0$  : reference thickness =  
0.5mm).

(6) A method for designing a multiblade radial fan,  
15 wherein specifications of an impeller of a multiblade radial  
fan are determined so as to satisfy a correlation expressed  
by formulas  $(1.009 - \nu)/(1 - \nu) \leq Z_2$  and  $0.8 \geq \nu \geq 0.4$  (in  
the formulas,  $\nu = r_0/r_1$ ,  $Z_2 = 0.857 \{t_0/[(2\pi r_1/n) - t] + 1\}$ ,  
 $r_0$  : inside radius of the impeller,  $r_1$  : outside radius of  
20 the impeller,  $n$  : number of radially directed blades,  $t$  :  
thickness of the radially directed blades,  $t_0$  : reference  
thickness = 0.5mm).

(7) A multiblade radial fan, wherein specifications of an  
impeller of a multiblade radial fan satisfy a correlation  
25 expressed by a formula  $(1.009 - \nu)/(1 - \nu) \leq Z_2$  (in the  
formula,  $\nu = r_0/r_1$ ,  $Z_2 = 0.857 \{t_0/[(2\pi r_1/n) - t] + 1\}$ ,  $r_0$  :  
inside radius of the impeller,  $r_1$  : outside radius of the



impeller,  $n$  : number of radially directed blades,  $t$  :  
thickness of the radially directed blades,  $t_0$  : reference  
thickness = 0.5mm).

(8) A multiblade radial fan, wherein specifications of an  
5 impeller of a multiblade radial fan satisfy a correlation  
expressed by formulas  $(1.009 - \nu)/(1 - \nu) \leq Z_2$  and  $0.8 \geq \nu \geq$   
0.4 (in the formulas,  $\nu = r_0/r_1$ ,  $Z_2 = 0.857 \{t_0/[(2\pi r_1/n) -$   
 $t] + 1\}$ ,  $r_0$  : inside radius of the impeller,  $r_1$  : outside  
radius of the impeller,  $n$  : number of radially directed  
10 blades,  $t$  : thickness of the radially directed blades,  $t_0$  :  
reference thickness = 0.5mm).

(9) A multiblade radial fan comprising an impeller having  
many radially directed blades which are circumferentially  
spaced from each other so as to define narrow channels  
15 between them, wherein laminar boundary layers in the  
interblade channels are prevented from separating.

(10) A multiblade radial fan of any one of claims 3, 4, 7,  
8 and 9, wherein inner end portions of the radially directed  
blades are bent in the direction of rotation of the  
20 impeller.