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(54) **MULTIVANE CENTRIFUGAL FAN**

(57) A multivane centrifugal fan having a plurality of vanes arranged in a circumferentially spaced state is provided with a plurality of annular plates piled at radially inner portions of the vanes so as to be spaced from one another slightly in the axial direction of the multivane centrifugal fan.

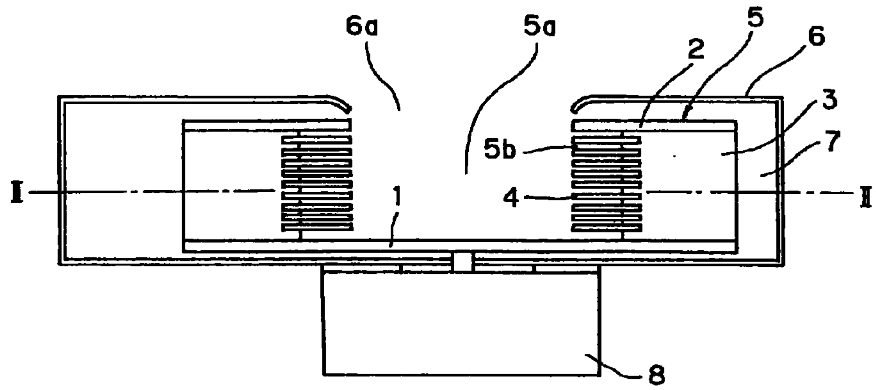

Fig. 1

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Description

[TECHNICAL FIELD]

The present invention relates to a multiblade centrifugal fan.

[BACKGROUND ART]

There have been known multiblade centrifugal fans such as the sirocco fan, the turbo fan and the radial fan. A multiblade centrifugal fan has numerous blades circumferentially spaced from each other.

The noise of a multiblade centrifugal fan is raised by various factors such as the fluid separation at the leading edges of the blades (inside edge of the impeller of the fan) due to the difference between the fluid inlet angle to the leading edges of the blades and the setting angle of the blades, the fluid separation in interblade channels of the impeller of the fan, the difference between the fluid outlet angle from the impeller of the fan and the divergence angle of a casing for accommodating the impeller, the interference between the tongue of the casing and the blades, etc.

As shown in Figure 18(a), the fluid flows substantially radially to the leading edges of the blades in an absolute coordinate system. However, as shown in Figure 18(b), in a relative coordinate system seen from the blades, the fluid flows obliquely to the direction in which the leading edges of the blades extend. In other words, the fluid inlet angle to the leading edges of the blades is different from the setting angle of the blades. Fluid separation at the leading edges of the blades is caused as a result.

One counter measure for reducing the fluid separation at the leading edges of the blades is, as is adopted in the turbo fan, to make the blades backward-curved to decrease the difference between the fluid inlet angle to the leading edges of the blades and the setting angle of the blades. However, the setting angle of the blades must be greatly increased to achieve a substantial decrease in the noise of the fan, which, in turn, degrades such hydrodynamic characteristics of the fan as the P-Q characteristics.

[DISCLOSURE OF INVENTION]

The object of the present invention is therefore to provide a multiblade centrifugal fan wherein the difference between the fluid inlet angle to the leading edges of the blades and the setting angle of the blades is reduced and the noise of the fan caused by the fluid separation at the leading edges of the blades is reduced without changing the setting angle of the blades.

According to the present invention, there is provided a multiblade centrifugal fan, wherein numerous blades are disposed circumferentially spaced from each other, and numerous annular plates are disposed radially inside the blades as stacked in the direction in which the rotation axis of the multiblade centrifugal fan extends with narrow intervening spaces between adjacent ones thereof.

In a multiblade centrifugal fan in accordance with the present invention, fluid flows radially outward through channels formed between the numerous annular plates disposed radially inside the blades as stacked in the direction in which the rotation axis of the multiblade centrifugal fan extends with narrow intervening spaces between adjacent ones thereof. The rotating annular plates apply tangential shear force to the fluid flowing through the channels to accelerate it tangentially, thereby increasing its tangential velocity. The fluid whose tangential velocity has been increased then flows into channels formed between the adjacent blades. In the present multiblade centrifugal fan, the difference between the circumferential velocity of the leading edges of the blades and the tangential velocity of the fluid is smaller than that in a multiblade centrifugal fan having no annular plates. Thus, in the present multiblade centrifugal fan, the difference between the fluid inlet angle to the leading edges of the blades and the setting angle of the blades is smaller than that in a multiblade centrifugal fan having no annular plates.

According to a preferred embodiment of the present invention, the outer peripheries of the annular plates are radially inwardly spaced from the leading edges of the blades.

Even if the outer peripheries of the annular plates are radially inwardly spaced from the leading edges of the blades, the fluid flowing radially outward from the channels between the adjacent annular plates flows to the leading edges of the blades without losing tangential velocity. Thus, the difference between the fluid inlet angle to the leading edges of the blades and the setting angle of the blades decreases. The structure wherein the outer peripheries of the annular plates are radially inwardly spaced from the leading edges of the blades is advantageous in that a multiblade centrifugal fan in accordance with the present invention can be easily obtained by disposing the annular plates in a conventional multiblade centrifugal fan.

According to another preferred embodiment of the present invention, the outer peripheries of the annular plates are in contact with the leading edges of the blades.

According to another preferred embodiment of the present invention, the outer peripheries of the annular plates overlap the leading edges of the blades.

When the outer peripheries of the annular plates are in contact with the leading edges of the blades or overlap the leading edges of the blades, the fluid flowing radially outward from the channels between the adjacent annular plates flows to the leading edges of the blades without losing tangential velocity, thereby reducing the difference between the fluid inlet angle to the leading edges of the blades and the setting angle of the blades, and, in addition, the strength of the multiblade centrifugal fan is increased by the contact or overlapping of the outer peripheries of the annular plates with the leading edges of the blades.

According to another preferred embodiment of the present invention, the blades are radially directed blades.

According to another preferred embodiment of the present invention, the blades are backward-curved blades.

According to another preferred embodiment of the present invention, the blades are forward-curved blades.

Irrespective of whether the blades are radially directed blades (radial fan), backward-curved blades (turbo fan), or forward-curved blades (sirocco fan), the difference between the fluid inlet angle to the leading edges of the blades and the setting angle of the blades can be reduced by disposing numerous annular plates radially inside the blades as stacked in the direction in which the rotation axis of the multiblade centrifugal fan extends with narrow intervening spaces between adjacent ones thereof.

[BRIEF DESCRIPTION OF THE DRAWINGS]

In the drawings:

Figure 1 is a sectional view of a multiblade radial fan in accordance with a preferred embodiment of the present invention.

Figure 2 is a sectional view taken along line II-II in Figure 1

Figure 3 is a sectional view of a multiblade radial fan showing the difference between the fluid inlet angle to the leading edges of the blades and the setting angle of the blades.

Figure 4 is a layout diagram of a measuring apparatus for measuring air volume flow rate and static pressure of a multiblade centrifugal fan.

Figure 5 is a layout diagram of a measuring apparatus for measuring the sound pressure level of a multiblade centrifugal fan.

Figure 6(a) is a plan view of a tested impeller (radial fan) without stacked annular plates and Figure 6(b) is a sectional view taken along line b-b in Figure 6(a).

Figure 7(a) is a plan view of a tested impeller (radial fan) with stacked annular plates and Figure 7(b) is a sectional view taken along line b-b in Figure 7(a).

Figure 8 is a plan view of a tested casing (radial fan).

Figure 9 is a view showing experimentally obtained correlations between minimum specific sound levels K_{Smin} and differences θ between the fluid inlet angle to the leading edges of the blades and the setting angle of the blades.

Figure 10(a) is a sectional view of a tested impeller (sirocco fan) without stacked annular plates and Figure 10(b) is a sectional view of a tested impeller (turbo fan) without stacked annular plates.

Figure 11(a) is a sectional view of a tested impeller (sirocco fan) with stacked annular plates and Figure 11(b) is a sectional view of a tested impeller (turbo fan) with stacked annular plates.

Figure 12 is a plan view of a tested casing (sirocco fan).

Figure 13 is a plan view of a tested casing (turbo fan).

Figure 14 is a view showing a comparison between the noise level of a sirocco fan with stacked annular plates and the noise level of a sirocco fan without stacked annular plates (at an impeller rotation speed of 5100 rpm).

Figure 15 is a view showing a comparison between the noise level of a sirocco fan with stacked annular plates and the noise level of a sirocco fan without stacked annular plates (at an impeller rotation speed of 6120 rpm).

Figure 16 is a view showing a comparison between the noise level of a turbo fan with stacked annular plates and the noise level of a turbo fan without stacked annular plates (at an impeller rotation speed of 5100 rpm).

Figure 17 is a view showing a comparison between the noise level of a turbo fan with stacked annular plates and the noise level of a turbo fan without stacked annular plates (at an impeller rotation speed of 6120 rpm).

Figure 18(a) and Figure 18(b) are sectional views of a multiblade centrifugal fan for xplaining why a difference arises between the fluid inlet angle to the leading edges of the blades and the setting angle of the blades.

[THE BEST MODE FOR CARRYING OUT THE INVENTION]

(A) 1st embodiment

A multiblade radial fan in accordance with an embodiment of the present invention will be described.

(I) Fan structure

In Figures 1 and 2, reference numeral 1 indicates a disk shaped base plate. An annular top plate 2 is disposed above the base plate 1. The top plate 2 is disposed parallel to and coaxially with the base plate 1. Numerous radial blades 3 are disposed as circumferentially spaced from each other to connect the base plate 1 with the top plate 2. A plurality of annular plates 4 are disposed radially inside the radial blades 3. The annular plates 4 are disposed parallel to and coaxially with the base plate 1. The annular plates 4 are stacked with narrow intervenig spaces between adjcent ones thereof. The outer peripheries of the annular plates 4 fit tightly within horizontal slits formed in the inner edges of the radial blades 3.

The base plate 1, the top plate 2, the radial blades 3 and the annular plates 4 constitute an impeller 5. The central openings of the stacked annular plates 4 form a central opening 5a of the impeller 5. Interplate channels 5b are formed between the base plate 1 and the lowermost annular plate 4, the top plate 2 and the uppermost annular plate 4, and adjacent annular plates 4. Interblade channels 5c are formed between adjacent radial blades 3.

The impeller 5 is disposed in a casing 6 having a scroll shaped horizontal cross section. The casing 6 is provided with an inlet opening 6a opposite the central opening 5a of the impeller 5 on the side of its top plate 2. The side wall of the casing 6 is provided with an outlet opening 6b and an outlet channel 7 is formed between the outer periphery of the impeller 5 and the side wall of the casing 6.

A motor 8 is disposed below the casing 6. The motor 8 is fixed to the bottom plate of the casing 6. The output shaft of the motor 8 extends upward through the bottom plate of the casing 6 and is fixed to the center of the lower surface of the base plate 1.

A multiblade radial fan having above described structure operates as follows.

The motor 8 starts. Fluid is drawn into the casing 6 through the inlet opening 6a. The fluid drawn into the casing 6 flows into the interplate channels 5b. The fluid entering the interplate channels 5b flows radially outward through the interplate channels 5b. As indicated by double arrows in Figure 2, the base plate 1, the top plate 2 and the annular plates 4, which are rotating, apply tangential shear force to the fluid flowing through the channels 5b to accelerate it tangentially and apply tangential velocity and centrifugal force to it. The fluid which has passed through the interplate channels 5b flows into the interblade channels 5c. The fluid passing into the interblade channels 5c flows radially outward through the interblade channels 5c. As indicated by single arrows in Figure 2, the radial blades 3, which are rotating, apply force normal to the radial blades 3 to the fluid flowing through the channels 5c to accelerate it still more and apply still larger centrifugal force to it.

The fluid passing through the interblade channels 5c flows out of the outer ends of the interblade channels 5c or the outer periphery of the impeller 5 and into the outlet channel 7. The fluid flowing into the outlet channel 7 flows circumferentially in the outlet channel 7 and flows out the casing 6 through the outlet opening 6b.

In the present multiblade radial fan, the base plate 1, the top plate 2 and the annular plates 4 accelerate the fluid flowing through the interplate channels 5b tangentially thereby increasing the tangential velocity thereof. Thus, in the present multiblade radial fan, the difference between the circumferential velocity of the leading edges of the radial blades 3 and the tangential velocity of the fluid flowing out of the interplate channels 5b and into the interblade channels 5c is smaller than that in a multiblade radial fan without the stacked annular plates 4. Thus, in the present multiblade radial fan, the difference between the fluid inlet angle to the leading edges of the blades 3 and the setting angle of the blades 3 is smaller than that in a multiblade radial fan without the stacked annular plates 4, and the noise caused by the fluid separation at the leading edges of the radial blades 3 is less than that in a multiblade radial fan without the stacked annular plates 4. Outer peripheries of the annular plates 4 fit tightly within horizontal slits formed in the inner edges of the radial blades 3. Thus, the present multiblade radial fan is very sturdy.

(I I) Noise measurement

Noise measurements were carried out on multiblade radial fans in accordance with the present invention and multiblade radial fans without the stacked annular plates.

(1) Difference between the inlet angle of the air to the leading edges of the blades and the setting angle of the blades.

As shown in Figure 3, the radial direction of a multiblade radial fan is defined as 0, the setting angle of the blades of the fan is defined as α, and the fluid inlet angle to the leading edges of the blades is defined as β.

Then, the difference θ between the fluid inlet angle to the leading edges of the blades of the multiblade radial fan and the setting angle of the blades of the multiblade radial fan is given by formula ①.

$$\theta = \beta - \alpha$$
$$= \tan^{-1}\{(r_i / r_0)^2 / \phi\} - \alpha \cdots\cdots ①$$

In the above formula,
r_i : radial position of the leading edges of the blades
r_0 : radial position of the trailing edges of the blades
ϕ : flow coefficient

$$\phi = u_0 / C_0$$

u_0 : mean radial flow velocity of the fluid at position r_0
c_0 : circumferential velocity of the blades at position r_0

The tangential velocity of the fluid flowing through the interplate channel relative to the annular plates was obtained by the method of Hasinger (Hasinger, S. and Kehrt, L., Trans. ASME, J.Eng.Power, 85(1963), 201). In accordance with this method, the tangential velocity V_k of the fluid relative to the annular plates at the outer peripheries of the annular plates is given by formula ②.

$$V_k = c_k X \cdots\cdots ②$$

$$X=(A/12\pi)(r_j / r_k)^2 - ([A/12\pi]-1)(r_j / r_k)^2 \exp([12\pi/A][1-(r_k / r_j)^2])$$

In the above formula,
r_j : inside radius of the annular plates
r_k : outside radius of the annular plates
A : nondimensional constant

$$A = (q\delta/\nu)/(r_j)^2$$

q : flow rate in an interplate channel
δ : space between adjacent annular plates
ν : kinematic viscosity
C_k : circumferential velocity of the annular plates at the position r_k

When stacked annular plates are disposed radially inside the leading edges of the blades and the outer peripheries of the annular plates are in contact with the leading edges of the blades, the difference angle θ at the leading edges of the blades is given by formula ③ derived from the formulas ① and ②.

$$\theta = \tan^{-1}\{[(r_i / r_0)^2 / \phi] X\} - \alpha \cdots\cdots ③$$

(2) Noise measurements

Noise measurements were carried out on multiblade radial fans in accordance with the present invention and multiblade radial fans without the stacked annular plates to obtain correlations between the minimum value of the specific sound level and the difference angle θ.

⟨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 4. The fan unit had an impeller 5, a scroll type casing 6 for accommodating the impeller 5 and a motor 8. An inlet nozzle was disposed on the suction side of the fan unit. A double chamber type air volume flow rate measuring apparatus (product of Rika Seiki Co. Ltd., Type F-401) was disposed on the discharge side of the fan unit. 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 unit. The air flow discharged from the fan unit was rectified by a honeycomb.

The air volume flow rate of the fan unit was measured using orifices located in accordance with the AMCA standard. The static pressure at the outlet of the fan unit was measured through a static pressure measuring hole disposed near the outlet of the fan unit.

② Measuring apparatus for measuring sound pressure level

The measuring apparatus for measuring sound pressure level is shown in Figure 5. An inlet nozzle was disposed on the suction side of the fan unit. 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 discharge side of the fan unit. The inside surface of the static pressure control chamber was covered with sound absorbing 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 unit.

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

The motor 8 was installed in a soundproof box lined with sound absorbing material. Thus, the noise generated by the motor 8 was confined.

The measurement of the sound pressure level was carried out in an anechoic room. The 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 without stacked annular plates

The outside diameter of the tested impellers (diameter at the trailing edges of the radial blades 3) was fixed at 100 mm. The height of the tested impellers was fixed at 24 mm. The thickness of the base plate 1 and the thickness of the top plate 2 were both set at 2 mm. Three different impellers 5 without stacked annular plates 4 were made. Different impellers 5 had a different ratio of the inside diameter (diameter at leading edges of the radial blades 3) to the outside diameter, and a different number of radial blades 3.

The particulars of the three tested impellers 5 (impeller numbers 1, 2 and 3) are shown in Table 1, and Figures 6(a) and 6(b).

② Tested impellers with stacked annular plates

The outside diameter of the tested impellers (diameter at the trailing edges of the radial blades 3) was fixed at 100 mm. The height of the tested impellers was fixed at 24 mm. The thickness of the base plate 1 and the thickness of the top plate 2 were both set at 2 mm. Three different impellers 5 with stacked annular plates 4 were made. Different impellers 5 had a different ratio of the inside diameter (diameter at leading edges of the radial blades 3) to the outside diameter, a different inside diameter of the annular plate 4, and a different number of radial blades 3.

The particulars of the three tested impellers 5 (impeller numbers 4, 5 and 6) are shown in Table 1, and Figures 7(a) and 7(b).

③ Tested casing

The height of the scroll type casing 6 was set at 27 mm. The divergence configuration of the scroll type casing 6 was set as a logarithmic spiral defined by the following formula. The divergence angle γ_c was set at 4.50°.

$$r_c = r_0 \exp(\gamma \tan \gamma_c)$$

In the above formula,

r_c : radius of the side wall of the casing measured from the center of the impeller 5
r_0 : outside radius of the impeller 5
γ : angle measured from a base line, $0 \leqq \gamma \leqq 2\pi$
γ_c : divergence angle
The tested casing 6 is shown in Figure 8.

③ Revolution speed of the impeller 5

The revolution speed of the impeller 5 was generally fixed at 6000 rpm but was varied to a certain extent considering extrinsic factors such as background noise in the anechoic room, condition of the measuring apparatus, etc. The revolution speeds of the impellers 5 when the specific sound level became minimum are shown in Table 1.

(3) Measurement, Data Processing

⟨2⟩ Measurement

The air volume flow rate of the air discharged from the fan unit, the static pressure at the outlet of the fan unit, and the sound pressure level were measured for each of the 6 kinds of the impellers 5 shown in Table 1 when rotated at the revolution speed shown in Table 1, while the air volume flow rate of the air discharged from the fan unit 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 unit, the static pressure at the outlet of the fan unit, and the sound pressure level, a specific sound level K_S defined by the following formula was obtained.

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

In the above formula,

SPL(A) : A-weighted (~ 20 KH_z), 1/3 octave band overall sound pressure level, dB
Q : air volume flow rate of the air discharged from the fan unit, m^3/s
P_t : total pressure at the outlet of the fan unit, mmAq

(4) Test Results

Based on the results of the measurements, a correlation between the specific sound level K_S and the air volume flow rate was obtained for each tested impeller 5.

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 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 unit obtained by the air volume flow rate and static pressure measurement are Q_1 and p_1 respectively, while the specific sound level K_S and the static pressure p at the outlet of the fan unit 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 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 5 varied with variation in the air volume flow rate. The variation of the specific sound level K_S is caused by the casing 6. Thus, it can be assumed that the minimum value of the specific sound level K_S or the minimum specific sound level K_{Smin} represents the noise characteristic of the tested impeller 5 itself free from the effect of the casing 6, and the minimum specific sound level K_{Smin} does not include the sound level caused by the difference between the outlet angle of the fluid flowing out the impeller and the divergence angle of the casing. In all tested impellers, the relation between the ratio of the inside radius of the impeller to the outside radius of the impeller and Karman-Millikan's nondimensional number Z_1, which relation was referred to in the PCT application PCT/JP95/00789 filed by the present applicant, was in the quite region which was proposed in the above PCT application (Karman-Millikan's nondimensional numbers Z_1 are shown in Table 1). Thus, it can be assumed that the minimum specific sound level K_{Smin} does not include the sound level caused by the fluid separation in the interblade channels of the impeller. According to the result of a spectrum analysis of the measured sound level data corresponding to the minimum specific sound level K_{Smin} , the energy of the sound with the same frequency as that of the noise due to the interference between the tongue of the casing and the blades of the impeller was very low. Thus, it can be assumed that the minimum specific sound level K_{Smin} does not include the sound level caused by the interference between the tongue of the casing and the blades of the impeller.

From the above, it can be assumed that the minimum specific sound level K_{Smin} shows the noise characteristics of the impeller caused by the air separation at the leading edges of the blades due to the difference between the air inlet angle to the leading edges of the blades and the setting angle of the blades.

The minimum specific sound level K_{Smin}, the flow coefficient ϕ corresponding to the minimum specific sound levels K_{Smin}, and the difference angle θ corresponding to the minimum specific sound levels K_{Smin} of each tested impeller 5 are shown in Table 1. Correlations between the minimum specific sound levels K_{Smin} and the difference angles θ of the tested impellers 5 are shown in Figure 9. The difference angles θ were calculated on the assumption that the outside diameter of the annular plate 4 ($2\ r_k$) is equal to the inside diameter of the impeller (the diameter at the leading edges of the radial blades 3).

(5) Discussion

From Table 1 and Figure 9, it is clear that the difference angles θ of the impellers 5 with the stacked annular plates (impeller numbers 4, 5 and 6) are smaller than those of the impellers 5 without the stacked annular plates (impeller numbers 1, 2 and 3), and the minimum specific sound level K_{Smin} decreases as the difference angle θ decreases.

From the above described sound level measurement, it was confirmed that the present invention can effectively reduce the noise of a multiblade centrifugal fan caused by the fluid separation at the leading edges of the blades due to the difference between the fluid inlet angle to the leading edges of the blades and the setting angle of the blades.

(B) 2nd embodiment

(1) Noise measurement

Noise measurements were carried out on sirocco fans and turbo fans produced by Rokugo Seisakusho Co. Ltd. The noise measurements were carried out on fans with stacked annular plates and fans without stacked annular plates. From the noise measurements, it was confirmed that the present invention is also effective when applied to sirocco fans and turbo fans.

⟨1⟩ Measuring apparatuses

The same measuring apparatuses for measuring air volume flow rate and static pressure as those used in the 1st embodiment were used. The same measuring apparatuses for measuring sound pressure level as those used in the 1st embodiment were used.

⟨2⟩ Tested impellers, Tested casing

① Tested impellers without stacked annular plates

An impeller of a sirocco fan produced by Rokugo Seisakusho Co. Ltd., (impeller no.1) (outside diameter × inside diameter × height of the interblade channels × number of blades = 102.0 mm × 85.3 mm × 29.0 mm × 32), and an impeller of a turbo fan produced by Rokugo Seisakusho Co. Ltd., (impeller no.2) (outside diameter × inside diameter × height of the interblade channels × number of blades = 99.0 mm × 54.8 mm × 17.0 mm × 10) were used as tested impellers.

The particulars of the no. 1 impeller are shown in Table 2 and Figure 10(a). The particulars of the no. 2 impeller are shown in Table 2 and Figure 10(b).

② Tested impellers with stacked annular plates

An impeller constituted by providing the impeller no. 1 with stacked annular plates (outside diameter × inside diameter × thickness of the annular plates × number of the annular plates × space between the adjacent annular plates × height = 85.0 mm × 65.0 mm × 0.3 mm × 42 × 0.4 mm × 29.0 mm) (impeller no. 3), and an impeller constituted by providing the impeller no. 2 with stacked annular plates (outside diameter × inside diameter × thickness of the annular plates × number of the annular plates × space between the adjacent annular plates × height = 54.0 mm × 40.0 mm × 0.3 mm × 22 × 0.4 mm × 15.0 mm) (impeller no. 4) were used as tested impellers.

The particulars of the no. 3 impeller are shown in Table 2 and Figure 11(a). The particulars of the no. 4 impeller are shown in Table 2 and Figure 11(b).

③ Tested casing

The height of the scroll type casing was set at an impeller height (interblade channel height + base plate thickness + top plate thickness) of + 9 mm for the impeller of the sirocco fan, and an impeller height (interblade channel height + base plate thickness + top plate thickness) of + 8 mm for the impeller of the turbo fan. The divergence configuration of the scroll type casing was set as a logarithmic spiral defined by the following formula. The divergence angle γ_c was set at 4.50°.

$$r_c = r_0 \exp(\gamma \tan \gamma_c)$$

In the above formula,
r_c : radius of the side wall of the casing measured from the center of the impeller 5
r_0 : outside radius of the impeller 5
γ : angle measured from a base line, $0 \leqq \gamma \leqq 2\pi$
γ_c : divergence angle

The tested casing for the impellers no. 1 and no. 3 (sirocco fan) is shown in Figure 12. The tested casing for the impellers no. 2 and no. 4 (turbo fan) is shown in Figure 13.

④ Revolution speed of the impeller

The revolution speed of the impeller was set at 5100 rpm and 6120 rpm.

(2) Measurement, Data Processing

⟨1⟩ Measurement

The air volume flow rate of the air discharged from the fan unit, the static pressure at the outlet of the fan unit, and the sound pressure level were measured for each of the 4 kinds of the impellers shown in Table 2 when rotated at the aforesaid two revolution speeds, while the air volume flow rate of the air discharged from the fan unit was varied using the air volume flow rate control dampers.

(2) Data Processing

Specific sound levels K_S were obtained in the same way as in the 1st embodiment.

(3) Test Results

Based on the results of the measurements, a correlation between the specific sound level K_S and the air volume flow rate Q was obtained for each tested impeller in the same way as in the 1st embodiment. Flow coefficients ϕ were obtained from the air volume flow rates Q based on the following formula ④.

$$\phi = u / v \cdot \cdot \cdot \cdot \cdot ④$$

In the above formula,
$u = Q / S$: radial velocity of the air flow at the outlet of the impeller
$v = r\,\omega$: circumferential velocity of the outer periphery of the impeller
$S = 2\,\pi rh$: area of the outlet of the impeller
Q : air volume flow rate
r : outside radius of the impeller
h : interblade channel height of the impeller
ω : rotation speed of the impeller

Correlations between the specific sound levels K_S and the flow coefficients ϕ were obtained from the correlations between the specific sound levels K_S and the air volume flow rates Q and the flow coefficients ϕ derived from the air volume flow rates Q.

Correlations between the specific sound levels K_S and the flow coefficients ϕ are shown in Figures 14 to 17.

From Figures 14 to 17, it is clear that also in the sirocco fan and the turbo fan the fan noise can be reduced over a wide range of flow coefficient distribution by disposing the stacked annular plates in the impeller.

It seems that the above described noise reduction was achieved by the combined effect of the noise reduction due to the suppression of the fluid separation at the leading edges of the blades, the noise reduction due to the suppression of the fluid separation in the interblade channels following the suppression of the fluid separation at the leading edges of the blades, and the reduction of the interference noise between the tongue of the casing and the blades resulting from more uniform circumferential velocity distribution of the air flow at the outlet of the interblade channels following the suppression of the fluid separation at the leading edges of the blades and in the interblade channels.

As described above, it is clear that also in the sirocco fan and the turbo fan the fan noise caused by the fluid separation at the leading edges of the blades can be reduced by disposing the stacked annular plates in the impeller.

(4) Discussion

It was confirmed that the present invention is effective when applied to the sirocco fan and the turbo fan.

Although preferred embodiments of the present invention and the noise measurements for confirmation of the effectiveness of the present invention were described above, the present invention is not restricted to the above mentioned embodiments.

For example, the outer peripheries of the annular plates 4 of the multiblade radial fan of the 1st embodiment may be radially inwardly spaced from the leading edges of the radial blades 3 or be in contact with the leading edges of the radial blades.

Even if the outer peripheries of the annular plates 4 are radially inwardly spaced from the leading edges of the blades 3, the fluid flowing radially outward from the channels between the adjacent annular plates 4 flows to the leading edges of the blades 3 without losing tangential velocity. Thus, the difference between the fluid inlet angle to the leading edges of the blades 3 and the setting angle of the blades 3 decreases. The structure wherein the outer peripheries of the annular plates 4 are radially inwardly spaced from the leading edges of the blades 3 is advantageous in that a multiblade radial fan in accordance with the present invention can be easily obtained by disposing the annular plates in a conventional multiblade radial fan.

When the outer peripheries of the annular plates 4 are in contact with the leading edges of the blades 3, the same effects can be achieved as when the outer peripheries of the annular plates 4 overlap the leading edges of the blades 3. Specifically, the fluid flowing radially outward from the channels between the adjacent annular plates 4 flows to the leading edges of the blades 3 without losing tangential velocity, thereby reducing the difference between the fluid inlet angle to the leading edges of the blades 3 and the setting angle of the blades 3, and, in addition, the strength of the multiblade radial fan can be increased by brazing, bonding or otherwise fixing the outer peripheries of the annular plates 4 in contact with the leading edges of the blades 3.

In the multiblade radial fan of the 1st embodiment, the stacked annular plates 4 need not be disposed over the entire space between the base plate 1 and the top plate 2 but instead may be disposed only over the portion of the space near the base plate 1 or over the portion of the space near the top plate 2 or over the mid-portion of the space.

[INDUSTRIAL APPLICABILITY OF THE INVENTION]

The present invention provides a multiblade centrifugal fan wherein the difference between the fluid inlet angle to the leading edges of the blades and the setting angle of the blades is reduced and the noise of the fan caused by the fluid separation at the leading edges of the blades is reduced without changing the setting angle of the blades.

TABLE 1

impeller NO.	outside diameter	inside diameter	blade thickness	number of blades	inside diameter of anular plate	space between adjacent annular plates	number of annular plates	thickness of anular plate	Ksmin	rotation speed atKsmin	ϕ at Ksmin	A	X	θ	Z_1
	(mm)	(mm)	(mm)		(mm)	(mm)		(mm)		(rpm)					
1	100.0	90.0	0.5	240					43.0	6000				85.76	0.1618
2	100.0	75.0	0.3	300					38.3	6000				82.91	0.3504
3	100.0	58.0	0.5	120					33.0	8000				76.62	0.5192
4	100.0	58.0	0.5	120	44.0	1.0	14	0.5	32.5	5000	0.07	112.71	0.83	75.86	0.5192
5	100.0	58.0	0.5	120	40.0	0.7	18	0.4	30.4	6000	0.07	78.01	0.69	74.26	0.5192
6	100.0	63.0	0.5	120	55.0	0.4	29	0.3	29.2	6000	0.09	20.79	0.61	69.74	0.4573

TABLE 2

impeller no.	1	2	3	4
outside diameter (mm)	102.0	99.0	102.0	99.0
inside diameter (mm)	85.3	54.8	85.3	54.8
height of interblade channel (mm)	29.0	17.0	29.0	17.0
number of blades	32	10	32	10
outside diameter of annular plate (mm)			85.0	54.0
inside diameter of annular plate (mm)			65.0	40.0
thickness of annular plate (mm)			0.3	0.3
number of annular plates			42	22
space between adjacent annular plates (mm)			0.4	0.4
height of stacked annular plates (mm)			29.0	15.0

Claims

1. A multiblade centrifugal fan, wherein numerous blades are disposed circumferentially spaced from each other, and numerous annular plates are disposed radially inside the blades as stacked in the direction in which the rotation axis of the multiblade centrifugal fan extends with narrow intervening spaces between adjacent ones thereof.

2. A multiblade centrifugal fan of claim 1, wherein the outer peripheries of the annular plates are radially inwardly spaced from the leading edges of the blades.

3. A multiblade centrifugal fan of claim 1, wherein the outer peripheries of the annular plates are in contact with the leading edges of the blades.

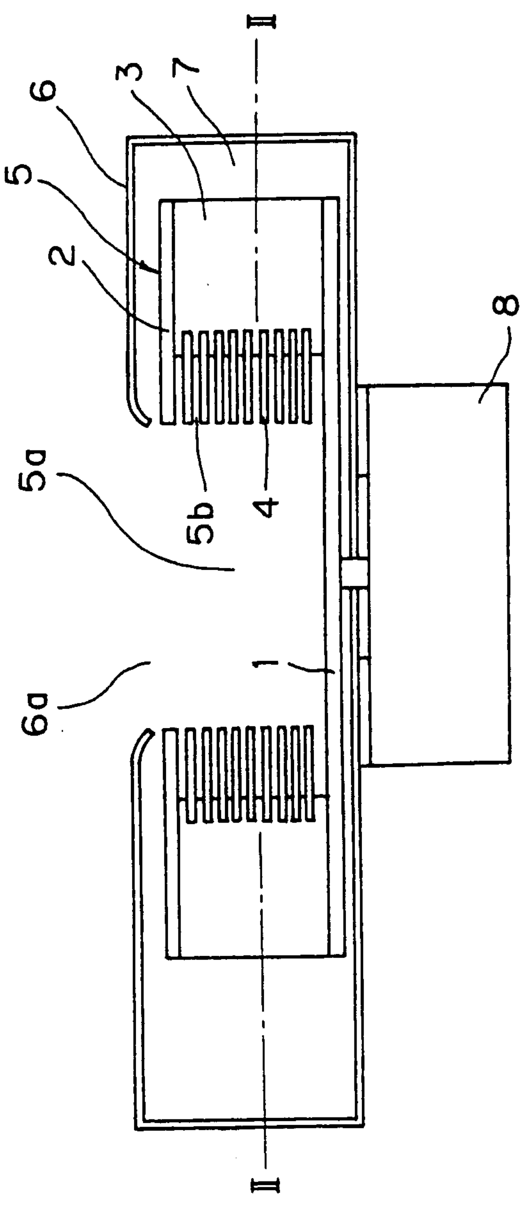
4. A multiblade centrifugal fan of claim 1, wherein the outer peripheries of the annular plates overlap the leading edges of the blades.

5. A multiblade centrifugal fan of claim 1, wherein the blades are radially directed blades.

6. A multiblade centrifugal fan of claim 1, wherein the blades are backward-curved blades.

7. A multiblade centrifugal fan of claim 1, wherein the blades are forward-curved blades.

Fig. 1



F i g. 2

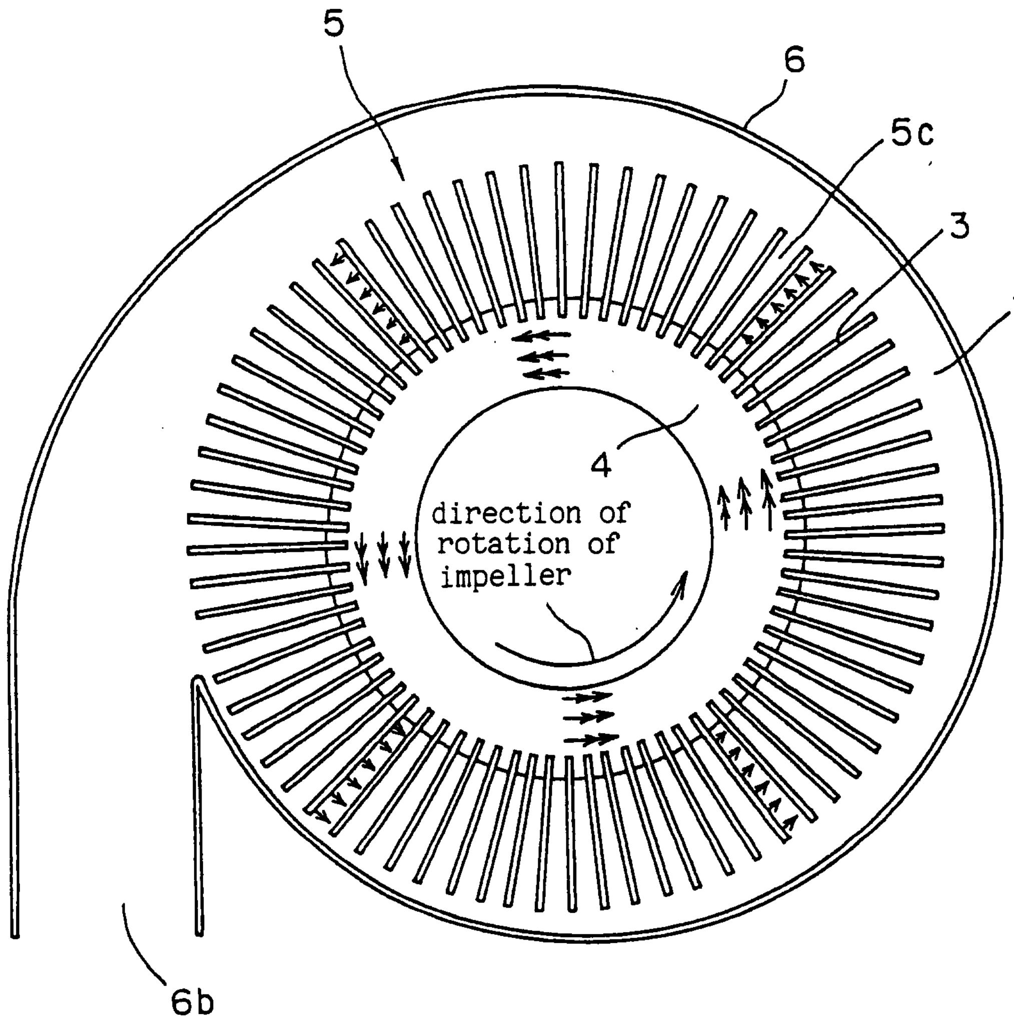
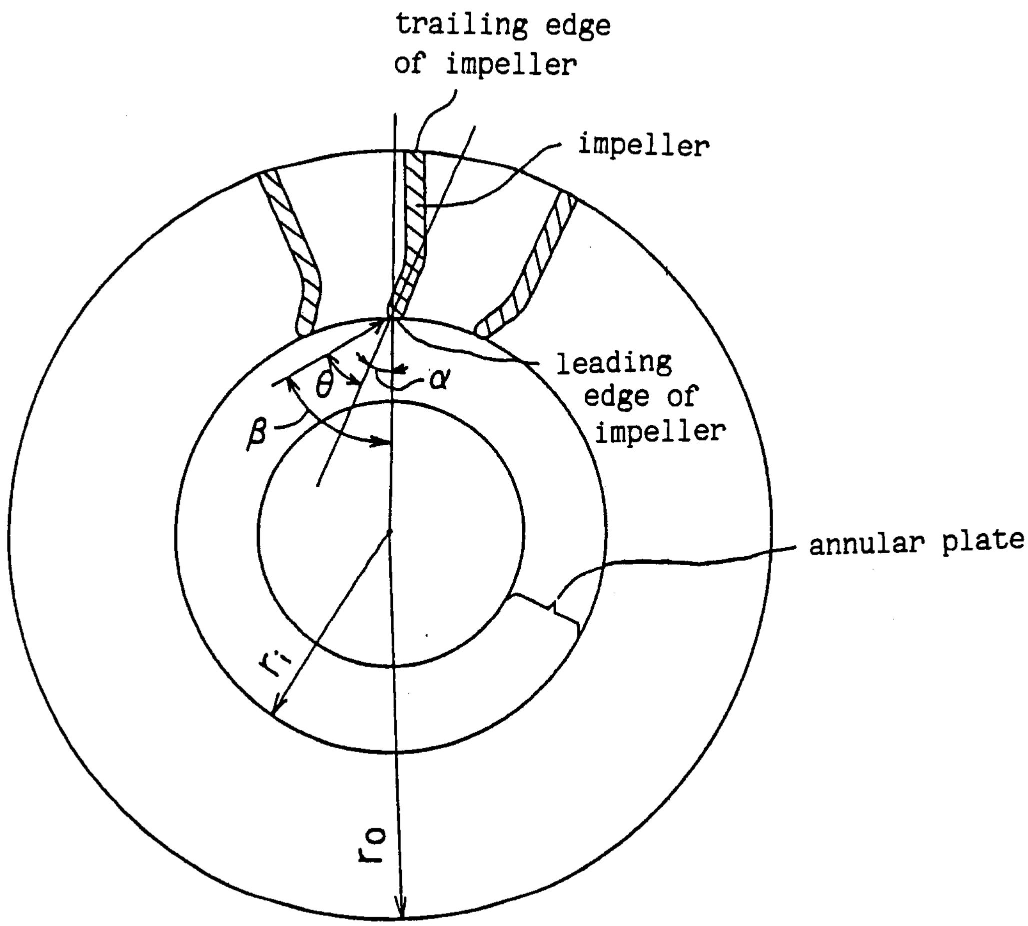
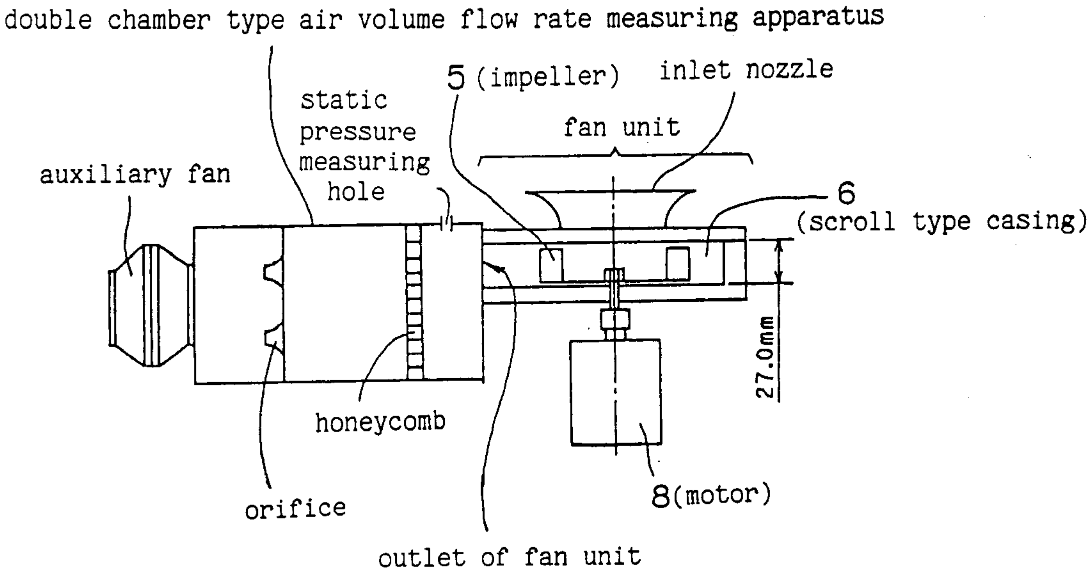


Fig. 3


trailing edge of impeller
impeller
leading edge of impeller
annular plate
θ
α
β
r_i
r_o

F i g. 4


double chamber type air volume flow rate measuring apparatus
static
pressure
measuring
hole
5 (impeller)
inlet nozzle
fan unit
auxiliary fan
6
(scroll type casing)
27.0mm
honeycomb
orifice
8(motor)
outlet of fan unit

F i g. 5

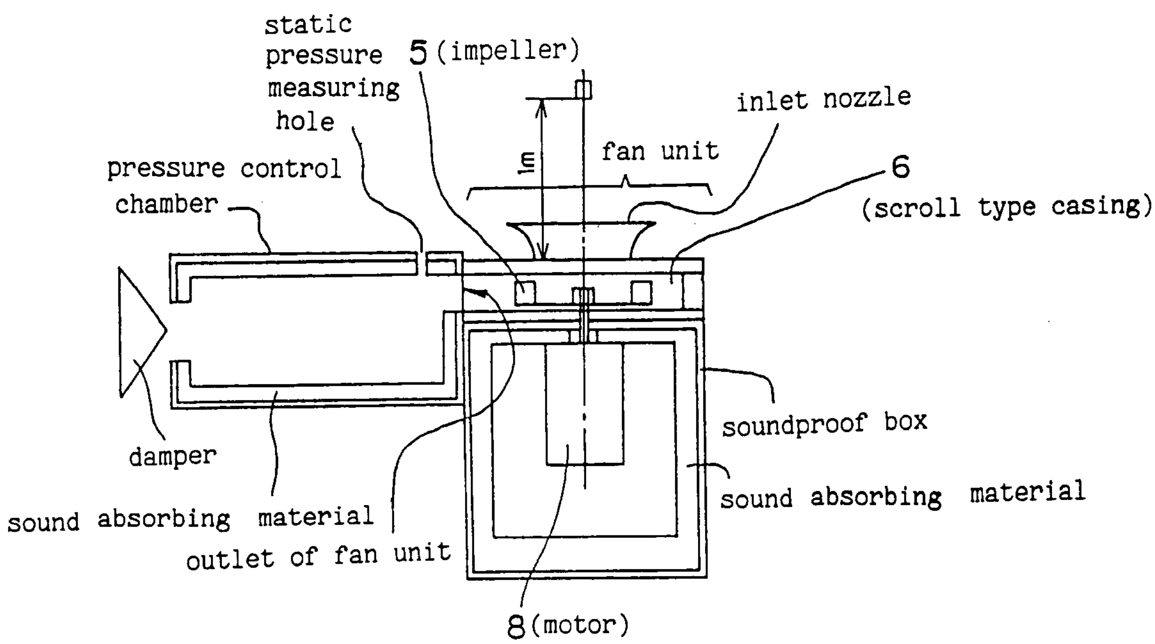

static
pressure
measuring
hole
5 (impeller)
inlet nozzle
1m
fan unit
pressure control
chamber
6
(scroll type casing)
soundproof box
damper
sound absorbing material
sound absorbing material
outlet of fan unit
8 (motor)

Fig. 6 (a)

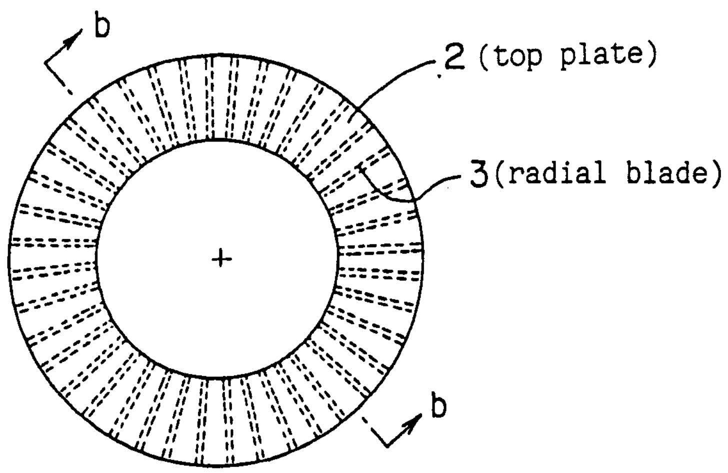

b
2 (top plate)
3 (radial blade)
b

Fig. 6 (b)

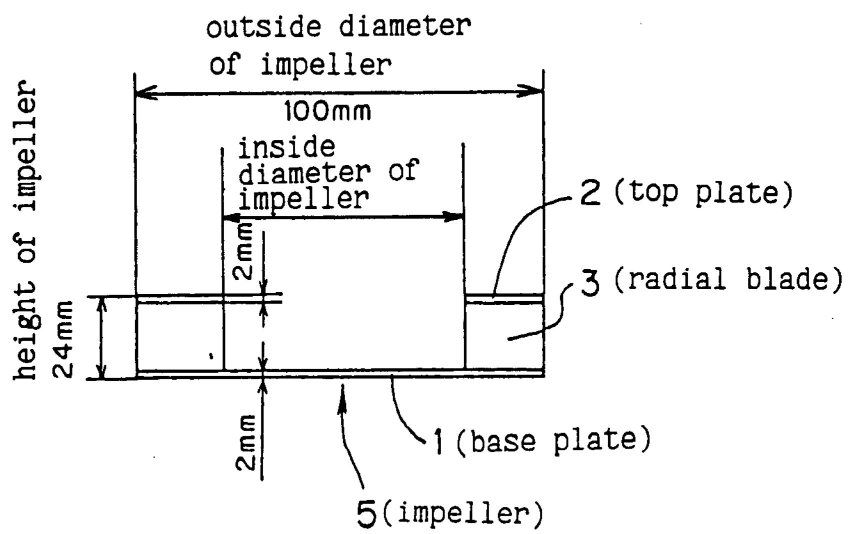

outside diameter
of impeller
100mm
inside
diameter of
impeller
2 (top plate)
3 (radial blade)
height of impeller
24mm
2mm
2mm
1 (base plate)
5 (impeller)

Fig. 7 (a)

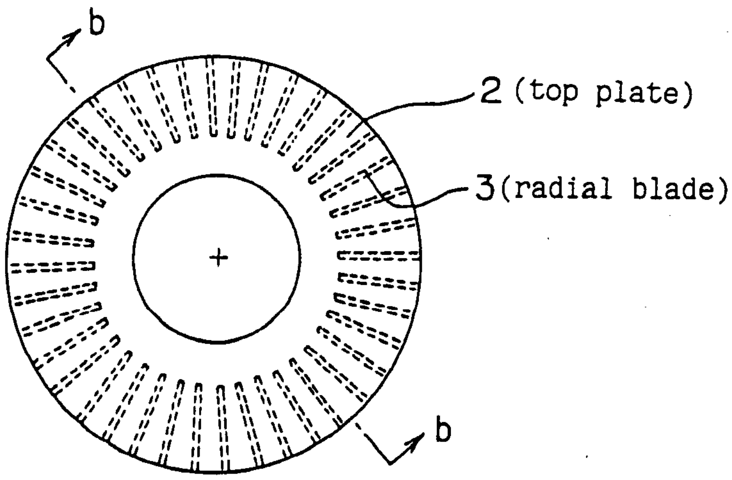

b
2 (top plate)
3 (radial blade)
b

Fig. 7 (b)

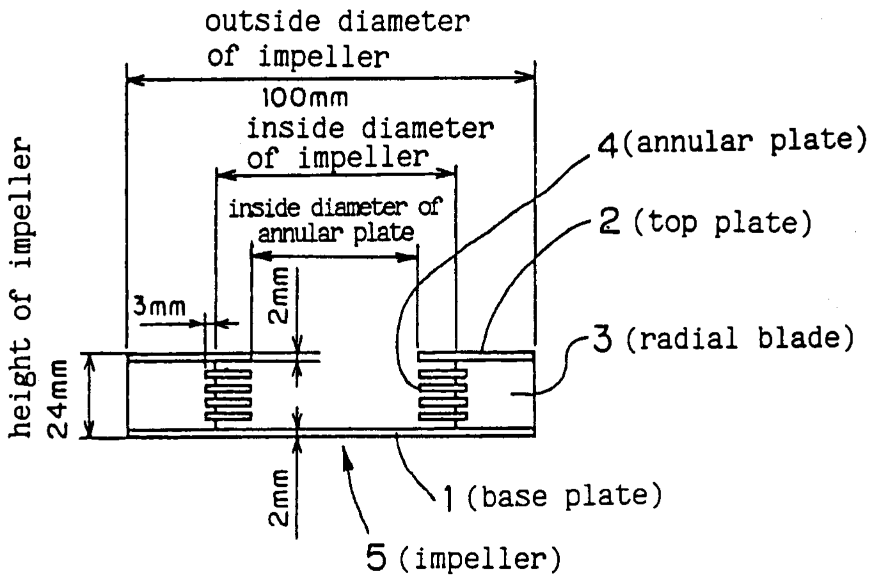

outside diameter
of impeller
100mm
inside diameter
of impeller
inside diameter of
annular plate
4 (annular plate)
2 (top plate)
3 (radial blade)
height of impeller
24mm
3mm
2mm
2mm
1 (base plate)
5 (impeller)

Fig. 8

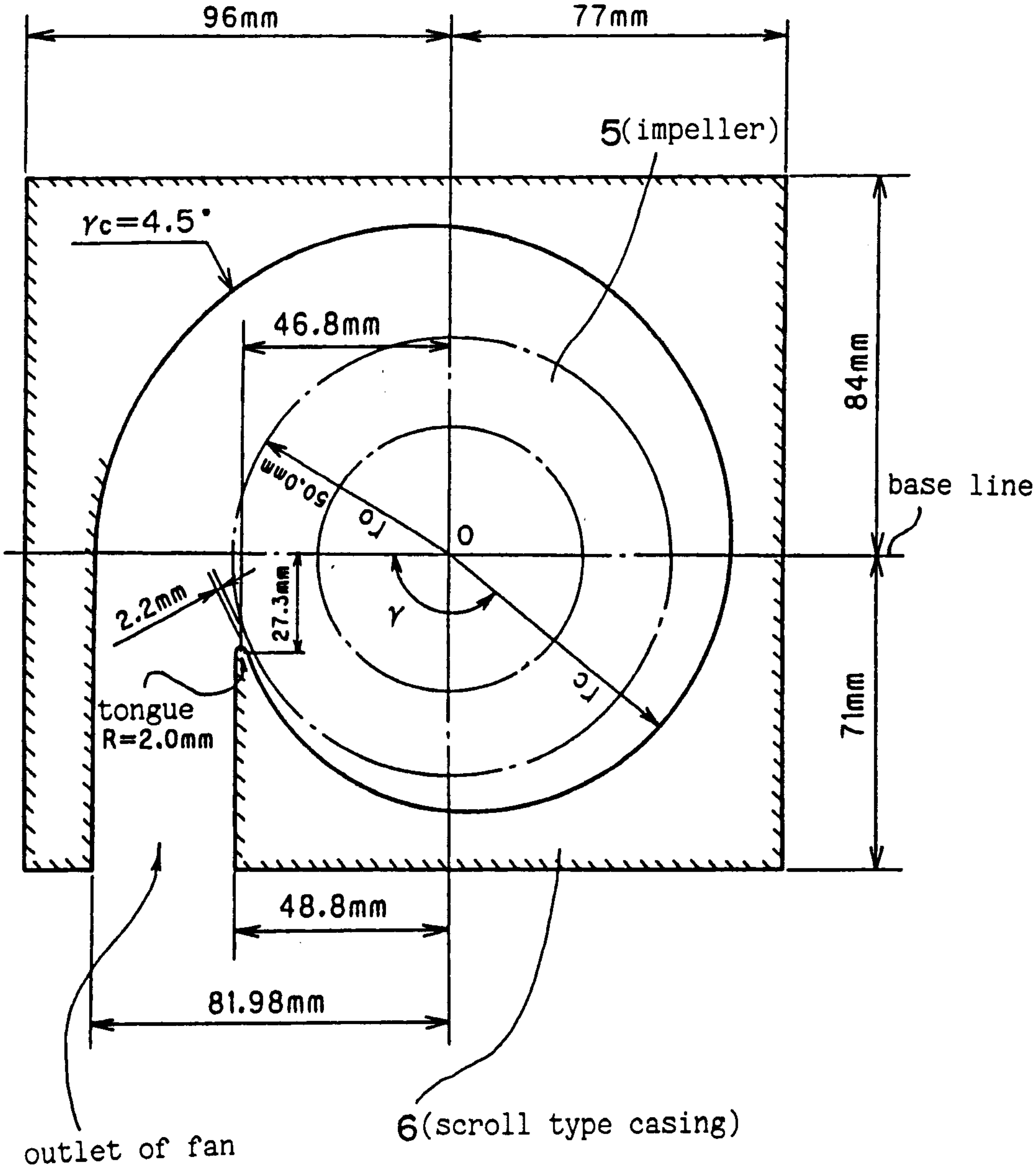

96mm
77mm
5(impeller)
γc=4.5°
46.8mm
84mm
base line
50.0mm
ro
O
2.2mm
27.3mm
γ
rc
tongue
R=2.0mm
71mm
48.8mm
81.98mm
outlet of fan
6(scroll type casing)

Fig. 9

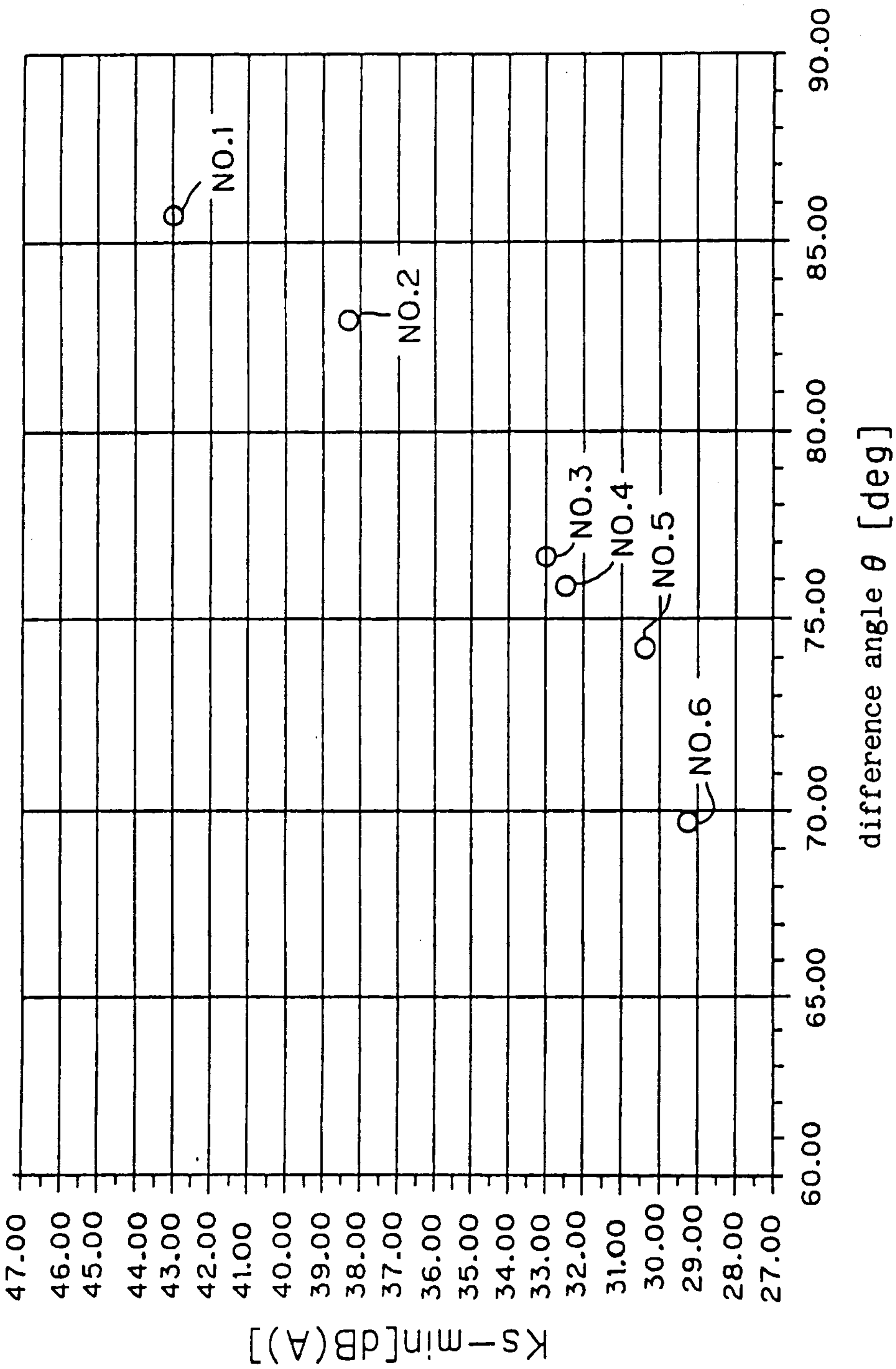

Ks−min[dB(A)]
47.00
46.00
45.00
44.00
43.00
42.00
41.00
40.00
39.00
38.00
37.00
36.00
35.00
34.00
33.00
32.00
31.00
30.00
29.00
28.00
27.00
60.00
65.00
70.00
75.00
80.00
85.00
90.00
difference angle θ [deg]
NO.1
NO.2
NO.3
NO.4
NO.5
NO.6

Fig. 10 (a)

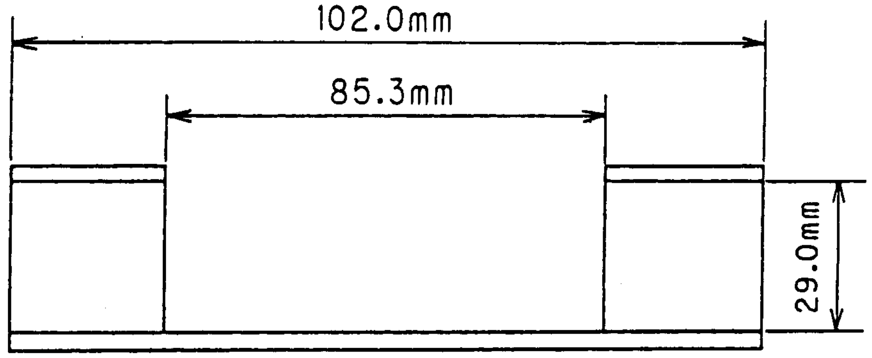


Fig. 10 (b)

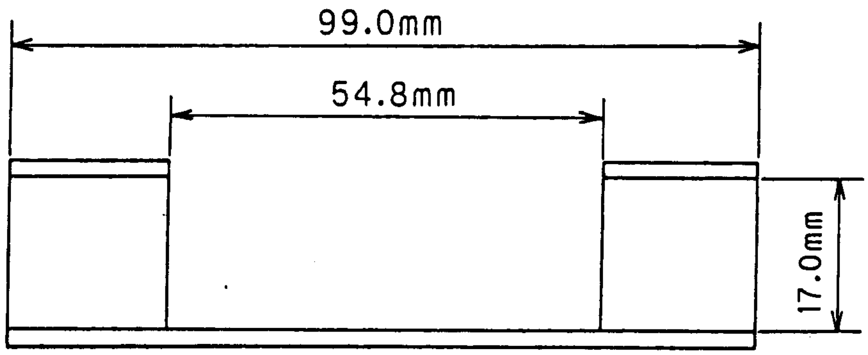


Fig. 11 (a)

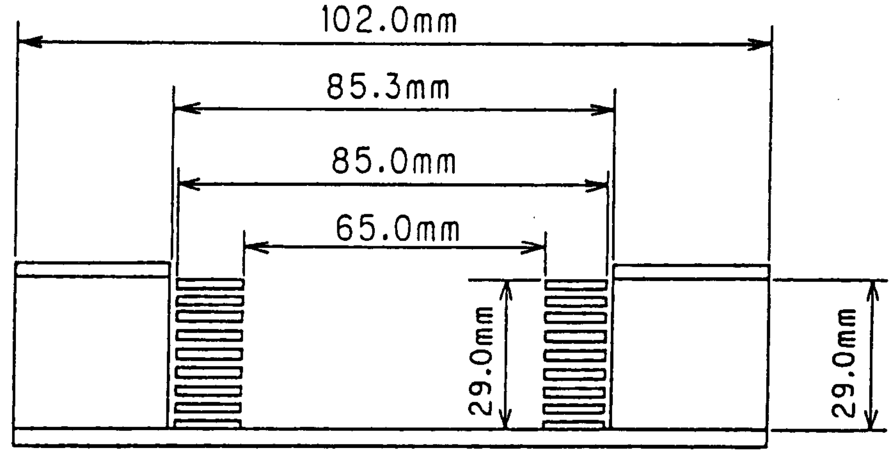


Fig. 11 (b)

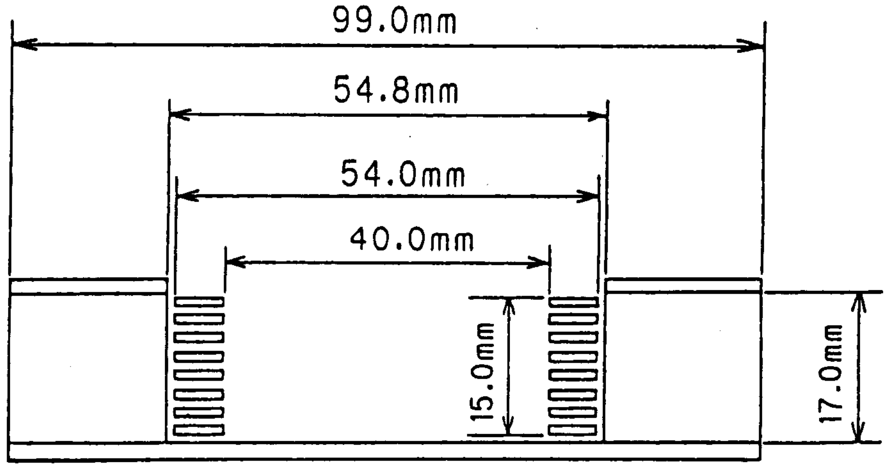


Fig. 12

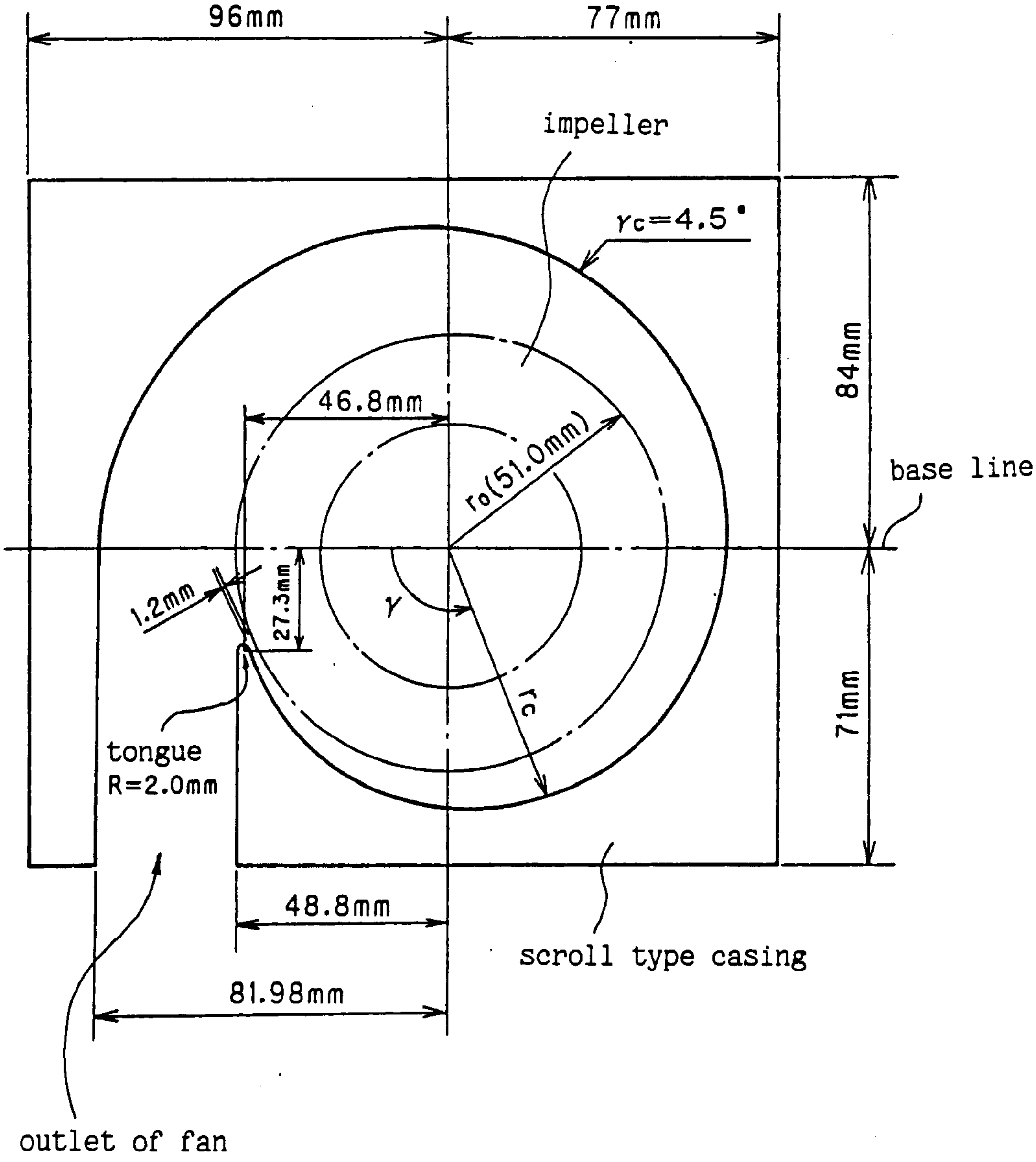
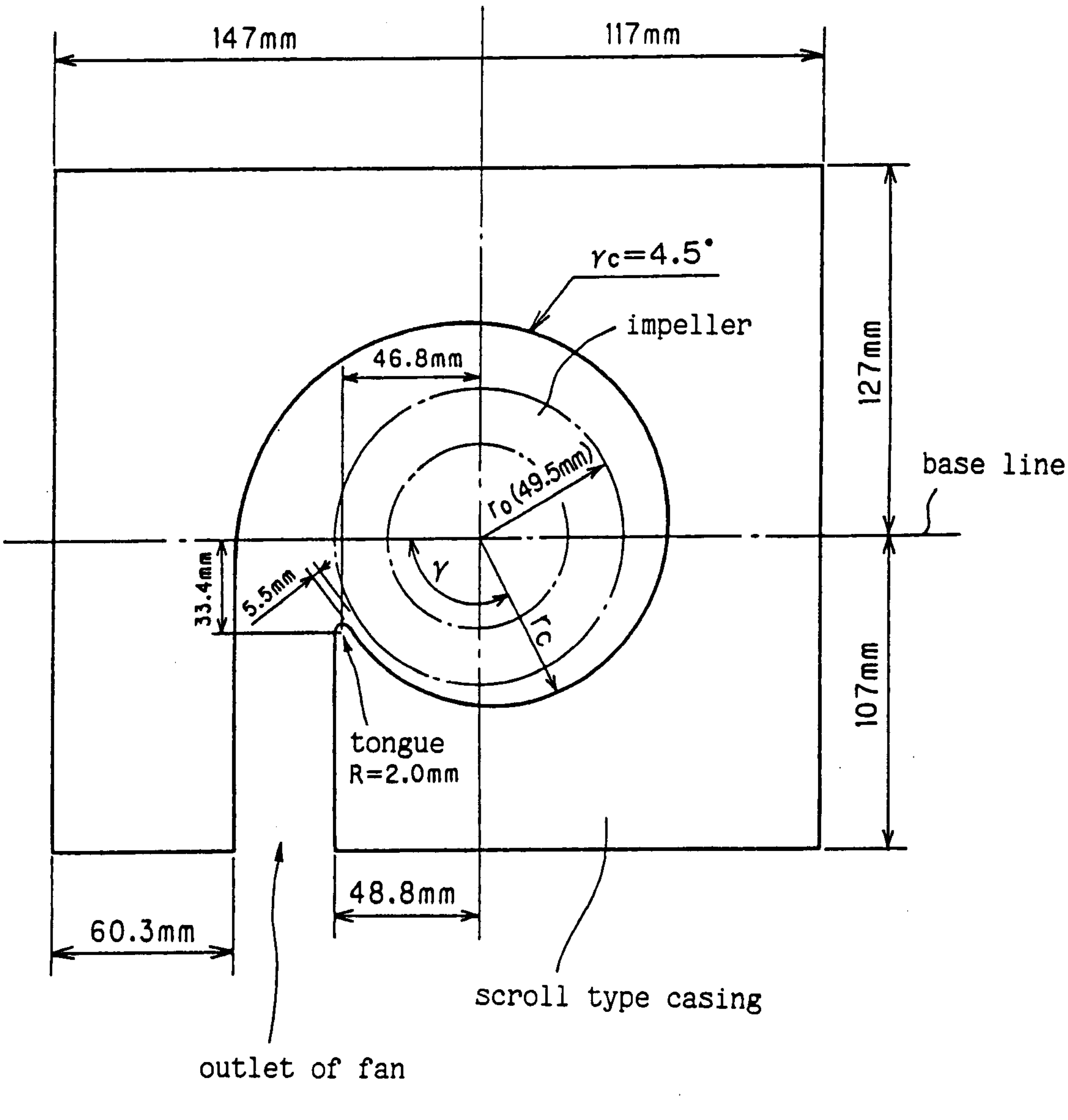
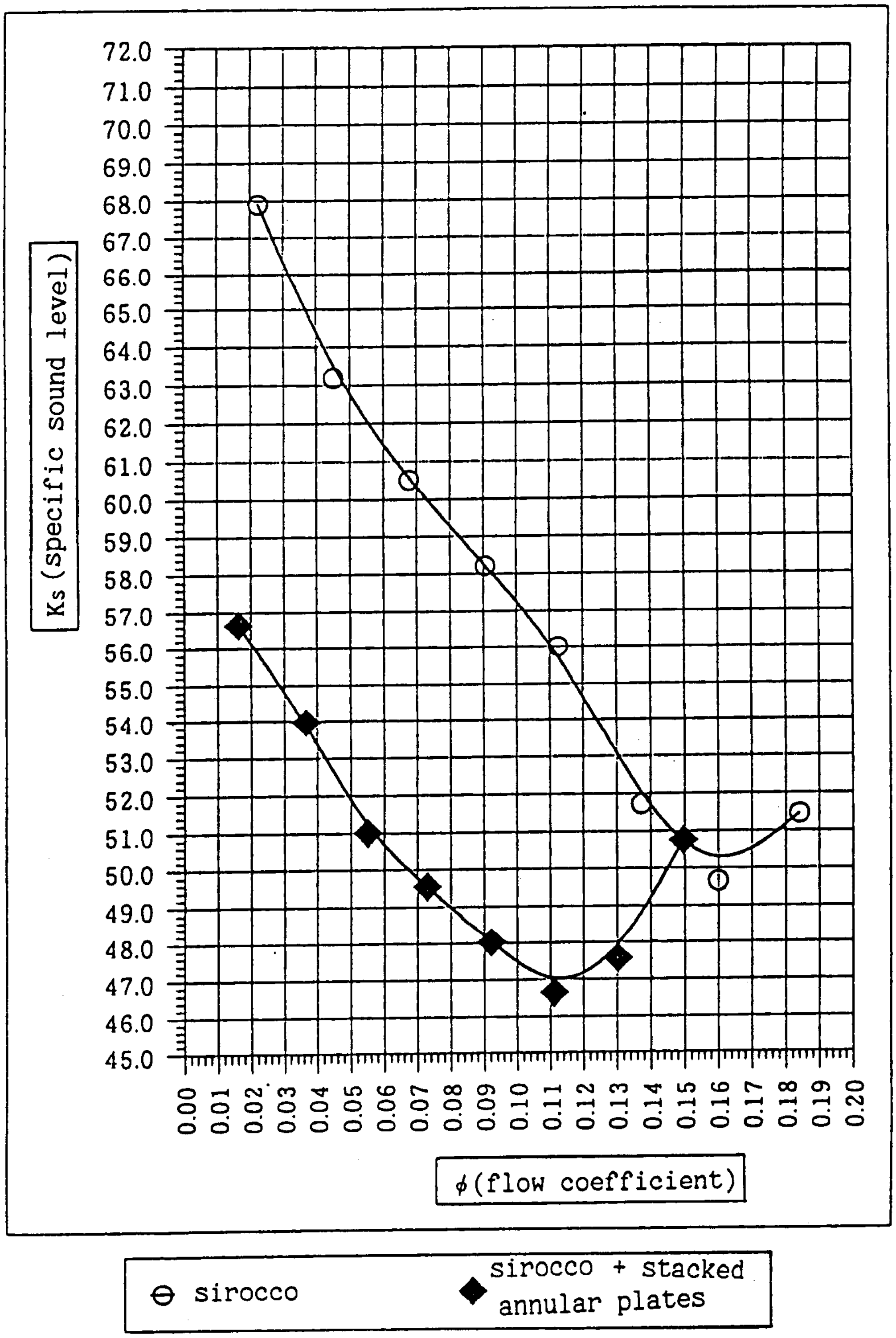

96mm
77mm
impeller
γc=4.5°
84mm
46.8mm
ro(51.0mm)
base line
1.2mm
27.3mm
γ
rc
71mm
tongue
R=2.0mm
48.8mm
scroll type casing
81.98mm
outlet of fan

Fig. 13

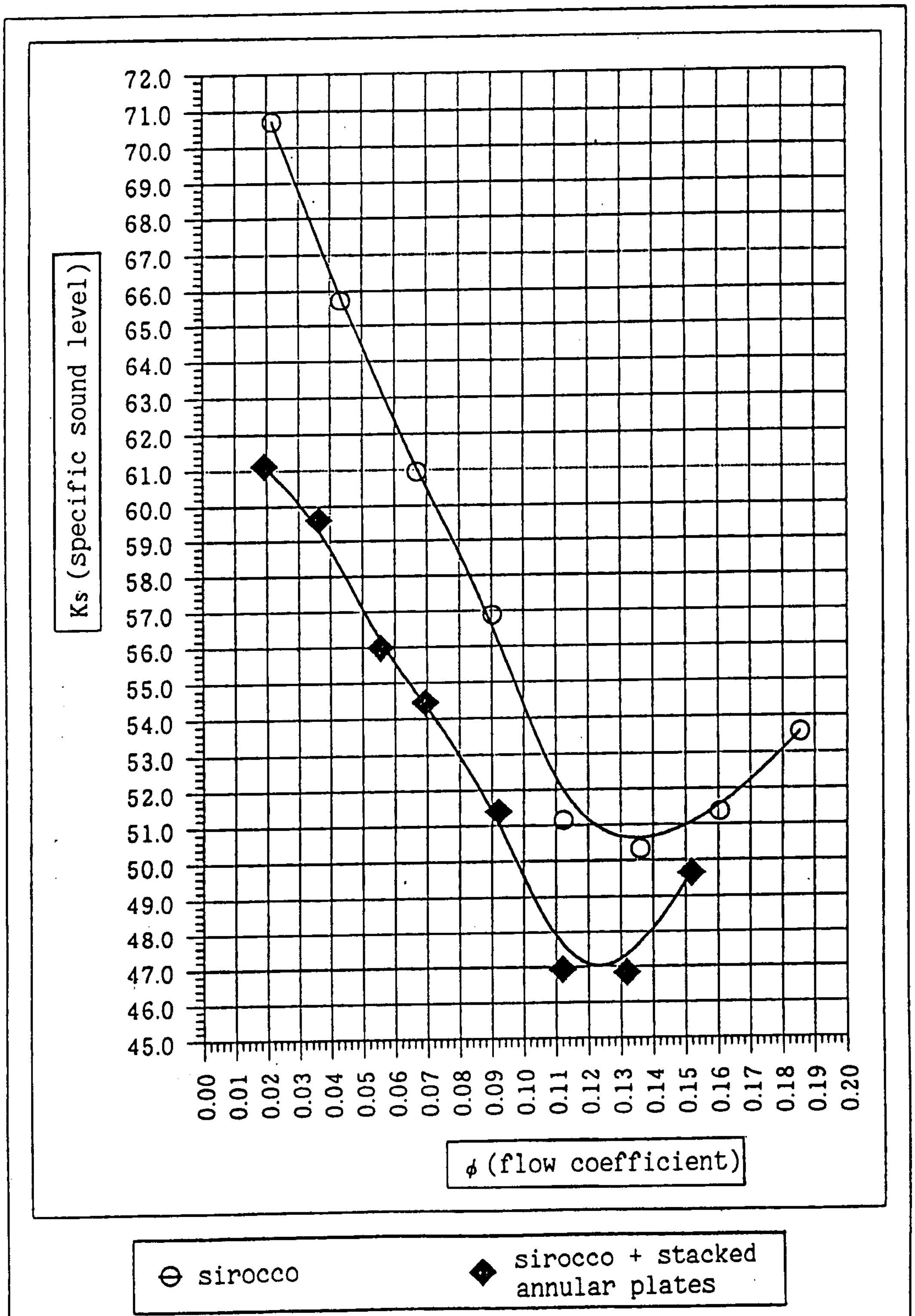

147mm
117mm
γc=4.5°
impeller
46.8mm
127mm
ro(49.5mm)
base line
γ
33.4mm
5.5mm
rc
107mm
tongue
R=2.0mm
48.8mm
60.3mm
scroll type casing
outlet of fan

F i g. 1 4



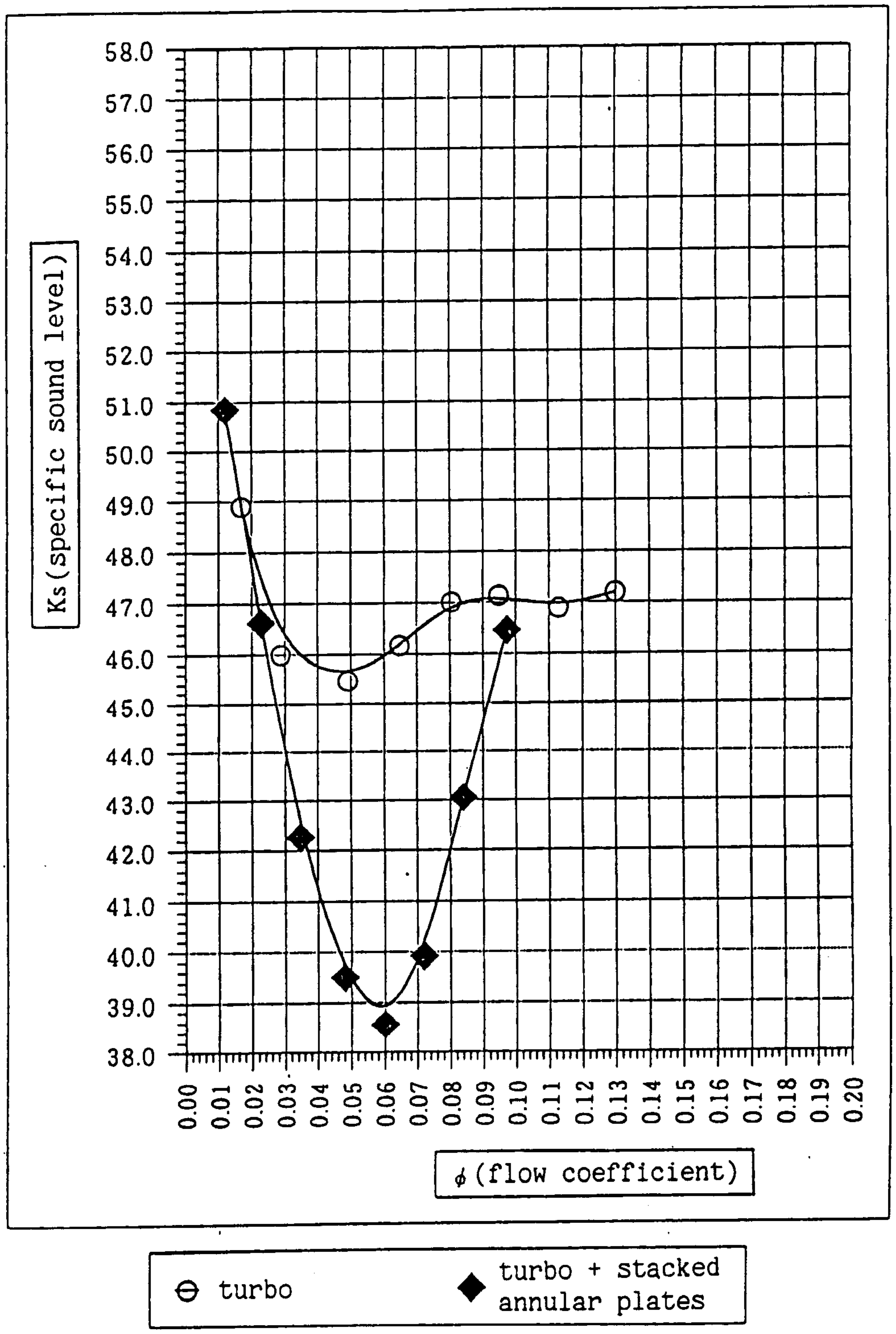
comparison of performance between sirocco fan and sirocco fan + stacked annular plates
rotation speed 5 1 0 0 r p m

Fig. 15



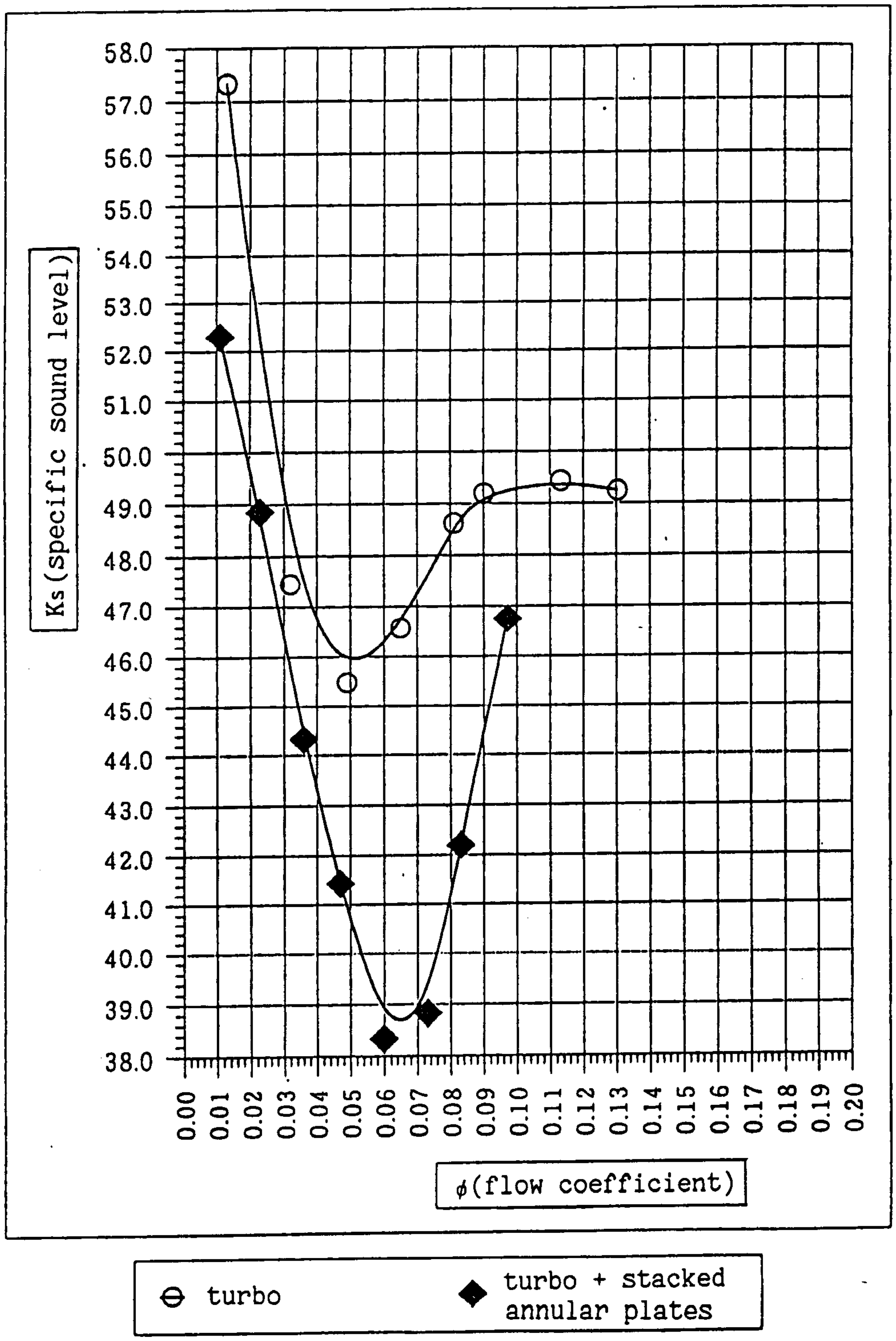
comparison of performance between sirocco fan and sirocco fan + stacked annular plates
rotation speed 6120 rpm

Fig. 16



comparison of performance between turbo fan and turbo fan + stacked annular plates
rotation speed 5 1 0 0 r p m

Fig. 17



comparison of performance between turbo fan and turbo fan + stacked annular plates
rotation speed 6120 rpm

Fig. 18 (a)

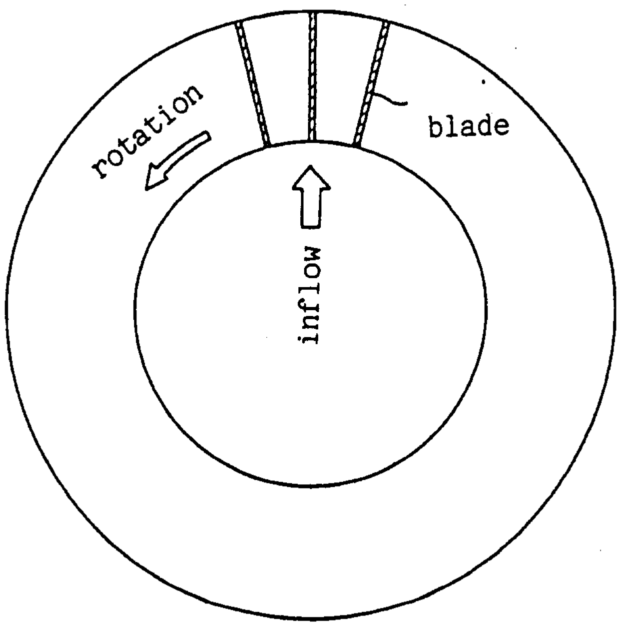
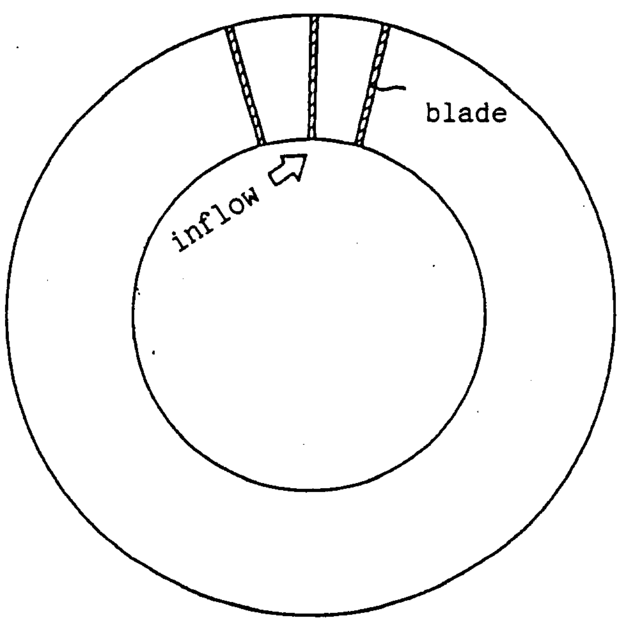


Fig. 18 (b)



INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP95/01307

A. CLASSIFICATION OF SUBJECT MATTER

Int. Cl^6 F04D29/30

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

Int. Cl^6 F04D29/30

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Jitsuyo Shinan Koho 1926 - 1995
Kokai Jitsuyo Shinan Koho 1971 - 1995

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	JP, 56-066494, A (Hitachi, Ltd.), June 4, 1981 (04. 06. 81), Lines 9 to 16, column 4 & US, 4,424,847, A & CA, 1,120,840, A	1 - 7

☐ Further documents are listed in the continuation of Box C. ☐ See patent family annex.

* Special categories of cited documents:
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"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
September 6, 1995 (06. 09. 95)	September 26, 1995 (26. 09. 95)
Name and mailing address of the ISA/ Japanese Patent Office	Authorized officer
Facsimile No.	Telephone No.

Form PCT/ISA/210 (second sheet) (July 1992)