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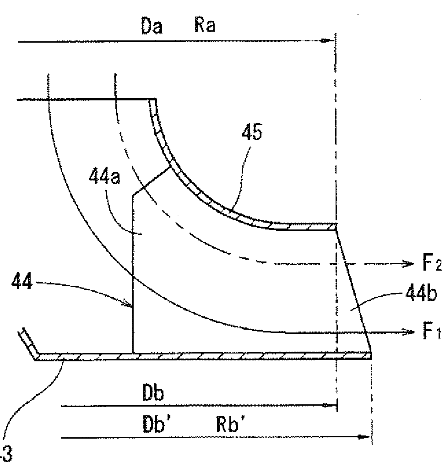
## (54) CENTRIFUGAL FAN

(57) A centrifugal fan 1 includes a plurality of blades 44 having a three-dimensional shape, a main plate 43, an annular side plate 45, and a motor 41 for rotating the blades 44 via the main plate 43. One end face of each blade 44 in the span direction is fixed to the main plate 43, such that the blades 44 are spaced at predetermined intervals along the circumferential direction. The annular side plate 45 is provided at the other end faces of the blades 44 in the span direction. The radial length of a trailing edge 44b of each blade 44 at a part close to the main plate 43 is set longer than the radial length of the trailing edge 44b of each blade 44 at a part close to the side plate 45.

Fig.12

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## Description

TECHNICAL FIELD

**[0001]** The present invention relates to a centrifugal fan, and more particularly, to the structure of impeller blades for a centrifugal fan.

BACKGROUND ART

**[0002]** In general, centrifugal fans such as backwardly curved blade fans generate undesirable noise. Accordingly, reduction of noise in centrifugal fans has been sought. Thus, various techniques have been developed for reducing air flow noise from centrifugal fans. A typical design approach for reducing air flow noise is to increase the outer diameter of the impeller of a fan.

**[0003]** When the obtained air volume is the same, an increased diameter of the fan impeller reduces the number of rotations of the fan impeller. This reduces the flow velocity of the air flow discharged by the fan impeller. The level of the air flow noise is proportional to the sixth power of the flow velocity.

**[0004]** However, a simple increase of the outer diameter of the fan impeller increases the size of the fan as a whole including an air passageway. Also, an increase in the outer diameter of the impeller will require an increase in the manufacturing costs for the fan, and readjustment of the parameters of the blades (the inlet angle, the outlet angle) and the layout in the apparatus.

**[0005]** In this respect, some techniques have been introduced that use thick blades, that is, airfoil blades to reduce flow separation around the blades, in an attempt to reduce noise at the lowest costs possible (for example, refer to Patent Document 1).

**[0006]** However, the use of the airfoil blades does not necessarily eliminate the flow separation over the entire surfaces of the blades. That is, noise caused by trailing vortices at the trailing edges of the blades is not reduced. The flow velocity of the discharged air flow varies along the span direction of each blade. That is, an uneven velocity distribution of air flow exists between the main plate and the side plate.

**[0007]** According to some other techniques, dimples or serrations are formed over the entire surface of each blade to reduce noise (for example, refer to Patent Document 2). In such a configuration, separation of air flow is further efficiently reduced, so that turbulence due to separation will be segmented. This further suppresses the development of trailing vortices. As a result, the noise caused by any trailing vortex is reduced.

**[0008]** However, this configuration can neither prevent an uneven velocity distribution of air flow on each blade between the main plate and the side plate along the span direction of the blade, nor suppress the generation of trailing vortices. Therefore, the noise caused by trailing vortices cannot be effectively prevented. Further, the configuration disadvantageously complicates the manufacturing process for surface machining and drawing of the blades.

**[0009]** Meanwhile, various techniques are known that improve the velocity distribution of air flow at the outlet of an impeller as described above. According to one of these techniques, a part of each blade adjacent thereto is cut out to reduce the length of the blade, thereby reducing the flow separation at this part. This equalizes the velocity distribution at the outlet of the impeller (refer to Patent Document 3).

**[0010]** However, because of the cut out portion in the part of each blade adjacent to the side plate, the length of the blade is less than a blade having no cut out portion. This reduces the amount of work that each blade applies to an air flow.

**[0011]** This technique is effective in a centrifugal fan in which the blade width is sufficiently small in relation to the outer shape, as described in Patent Document 3. Specifically, cutting out of the blades prevent separation of drawn-in air flow in a part of each blade in the vicinity of the side plate, and the air flow is allowed to flow along the part. However, the technique is not necessarily effective in a normal centrifugal fan, which has wide blades, that is, not in a case where the influence of separation near the side plate is not dominant.

**[0012]** Also, if a part of the blade near the side plate is cut out, air flow that passes through the part interferes with corners of a part of the blade near the main plate. This can generate noise having discretely-distributed frequency spikes.

**[0013]** Another technique has been known in which the blades are perpendicular to the main plate and the side plate, and the thickness of each blade is changed from the end face adjacent to the main plate to the end face adjacent to the side plate. Accordingly, fluctuation of the flow velocity distribution at the outlet of the blades is suppressed. In this case, the outer diameter of the main plate of the impeller and the outer diameter of the side plate of the impeller are made differently at the outlet of the impeller (outer end faces of the blades), and the size of each blade is increased toward the main plate (for example, refer to Patent Document 4), while maintaining the shape. This reduces the velocity fluctuation of the flow at the blade wake.

**[0014]** However, at the outlet of the impeller, the ratio of the outer diameter of the main plate of the impeller to the outer diameter of the side plate of the impeller is set to 1.2 to 1.6. Therefore, when the blades are enlarged with a small increase in the length, a noise reduction effect cannot be achieved. In contrast, when the length is excessively increased, the flow characteristics will be degraded.

**[0015]** As a result, the outer diameter of the main plate of the impeller is twenty percent greater than the outer diameter of the side plate. This structure is not advantageous over the technique in which the size of the fan is simply increased. Enlargement of the size by twenty percent or more goes entirely against desired goals that have been sought in the prior art, which are the size reduction and lowered noise, in the first place.

**[0016]** In the meantime, an impeller has been proposed that uses blades having a three-dimensional shape (for example, refer to Patent Document 2). Specifically, each blade extends in the rotation direction of the fan while being twisted from the main plate toward the side plate.

**[0017]** Compared with the impellers having the above described blades with two-dimensional shapes, an impeller having blades with three-dimensional shapes improves the load distribution on the blade surface and pressure fluctuations of air flow through between the blades.

**[0018]** A centrifugal fan employing blades having a three-dimensional shape and an air conditioner using the centrifugal fan will now be described with reference to Figs. 25 to 32.

**[0019]** Fig. 25 illustrates an air conditioner 1 using a centrifugal fan having an impeller. The air conditioner 1 is a ceiling embedded type, and includes a casing 2 accommodating various components, and a decorative panel 3 provided underneath the casing 2. More specifically, the casing 2 of the air conditioner 1 is fitted in an opening formed in a ceiling U of an air conditioned room, and the decorative panel 3 is arranged along the ceiling U.

**[0020]** The casing 2 is a box-like body having a lower opening. When viewed in the plan view, the casing 2 has an octagonal shape with short sides and long sides arranged alternately. The casing 2 has a substantially octagonal top plate 21 with alternately arranged long sides and short sides, and a side wall plate 22 extending downward from the periphery of the top plate 21.

**[0021]** The decorative panel 3 is shaped as a rectangular plate in the plan view. The panel 3 has an air inlet 31 and air outlets 32. The air inlet 31 is substantially located at the center of the decorative panel 3 and draws air from the air conditioned room. The air outlets 32 blows out air from the casing 2 into the air conditioned room. The decorative panel 3 is arranged such that each of its sides corresponds to one of the long sides of the top plate 21 of the casing 2.

**[0022]** The air inlet 31 is a substantially square opening. In contrast, each air outlet 32 is an elongated rectangular opening along a side of the decorative panel 3. The air inlet 31 includes an air intake grill 33 and a filter 34 for removing dust in the air drawn into the air inlet 31.

**[0023]** Further, a horizontal flap 35 is provided at each air outlet 32. The horizontal flap 35 can be swung about an axis extending along the longitudinal direction of the air outlet 32. The horizontal flap 35 is a rectangular blade-like member extending along the longitudinal direction of the air outlet 32. Each horizontal flap 35 is supported at both ends in the longitudinal direction with a support pins (not shown) so as to be rotatable. The horizontal flap 35 is thus capable of changing the direction of flow of air blown out of the air outlet 32.

**[0024]** An air blower 4 and a heat exchanger 6 are arranged in the casing 2. The air blower 4 draws air into the casing 2 from the air conditioned room through the air inlet 31 of the decorative panel 3. The heat exchanger 6 is arranged to surround the air blower 4.

**[0025]** The air blower 4 is a backwardly curved blade fan, which is a centrifugal fan to which the present invention is applied. The air blower 4 includes a fan motor (impeller driving means) 41 and an impeller 42 coupled to the shaft (rotary shaft) of the fan motor 41. The fan motor 41 is provided at the center of the top plate 21 of the casing 2 and is oriented downward.

**[0026]** The heat exchanger 6 is a cross-inner-tube type heat exchanger, which is formed by bending a tube to surround the air blower 4. The heat exchanger 6 is connected to an outdoor unit (not shown) installed outside the house through refrigerant piping. The heat exchanger 6 functions as an evaporator during the cooling operation, and as a condenser during the heating operation. The heat exchanger 6 performs heat exchange with air that is drawn into the casing 2 by the air blower 4 through the air inlet 31, so as to cool the air during the cooling operation, and heat the air during the heating operation.

**[0027]** A drain pan 7 is provided below the heat exchanger 6. The drain pan 7 receives drain water, which is produced when water in air is condensed on the surface of the heat exchanger 6. The drain pan 7 is attached to the lower part of the casing 2. The drain pan 7 has an air inlet portion 71 communicating with the air inlet 31 of the decorative panel 3, air outlet portions 72 corresponding to the air outlets 32 of the decorative panel 3, and a drain water receiving groove 73 that is formed to surround the lower part of the heat exchanger 6 to receive drain water.

**[0028]** A bellmouth 5 is provided at the air inlet portion 71 of the drain pan 7. The bellmouth 5 guides air drawn in through the air inlet 31 of the decorative panel 3 to the impeller 42 of the air blower 4.

**[0029]** The structure of the impeller 42 of the centrifugal air blower 4 will now be described with reference to Figs. 26 and 27. Fig. 26 is a perspective view showing the appearance of the impeller 42. Fig. 27 is a side view of the impeller 42 shown in Fig. 3.

**[0030]** The impeller 42 is mainly formed by a disk-shaped main plate 43, an annular side plate 45 separated from the main plate 43, and a plurality of blades 44 arranged between the main plate 43 and the side plate 45. The main plate 43 is coupled to a shaft 41a of the fan motor 41. The blades 44 are arranged along the main plate 43 at predetermined angular intervals about the shaft 41 a of the fan motor 41.

**[0031]** The main plate 43 is a synthetic resin member. A substantially truncated cone shaped bulge 43a is formed at the center of the main plate 43. The bulge 43a protrudes toward the air inlet 31. A main plate cover 46 is fixed at a position spaced from the lower surface of the main plate 43. The main plate cover 46 covers a cooling air hole. Radially arranged guide blades 46a are provided on the side of the main plate cover 46 that faces the main plate 43. The guide blades 46a cause part of the air blown out of the impeller 42 to flow about the main plate 43 by means of the difference between the static pressure in the space between the main plate 43 and the casing 2 and the static pressure in the space between the main plate 43 and the side plate 45. This part of the blown out air passes through the space about the fan motor 41 to cool the fan motor 41. Thereafter, the air is blown toward the space inside the impeller 42 through the cooling air hole of the main plate 43 and the guide blades 46a of the main plate cover 46.

**[0032]** The side plate 45 has a diameter that gradually decreases from the outer circumference toward the center opening. The side plate 45 is a bell shaped resin member that protrudes toward the air inlet 31.

**[0033]** The structure of the blades 44 of the impeller 42 will be described with reference to Figs. 28 to 32. Fig. 28 is a perspective view of the blade 44 as seen from the left rear. Fig. 29 is a projection view of the blade 44 of Fig. 28, as viewed from above. Fig. 30 is a side view of the blade 44 of Fig. 28 with section lines 31A-31A to 31E-31E. Figs. 31 (a) to 31 (e) are cross-sectional views taken along lines 31A-31A to 31E-31E. Fig. 32 is a schematic view illustrating an operation of the blade 44.

**[0034]** The blades 44 are resin members that are molded separately from the main plate 43 and the side plate 45. One end face of each blade 44 is fixed to the main plate 43, and another end face is fixed to the side plate 45. When viewed from the side of the impeller 42, the end of the blade 44 that faces the side plate 45 is inclined rearward than the end close to the main plate 43 as shown in Fig 28. The blade 44 is formed such that these ends cross each other to substantially form an X as shown in Fig. 29. In other words, the blade 44 has a three-dimensional shape that extends parallel to the rotary shaft while being twisted between the main plate 43 and the side plate 45.

**[0035]** The edge at the leading side in the rotation direction of the blade 44, which is a three-dimensional blade, or a leading edge 44a extends from the end close to the main plate 43 to a predetermined position close to the side plate 45 with substantially a constant radius. The blade 44 as an inclined edge from the predetermined position close to the side plate 45, while retreating outward with a gradually decreasing radius. The edge of the blade 44 on the trailing side of the rotation direction R, or the trailing edge 44b, has a shape formed by a straight line that extends parallel to the rotation from the position close to the main plate 43 and the position close to the side plate 45.

**[0036]** Compared to blade shapes formed based on a two-dimensional blade element as in Patent Documents 1, 3 and 4, the load distribution on the surface of the blade 44 and the pressure fluctuation of the air flow through between the blades 44 are significantly improved. Accordingly, at least noise caused by pressure fluctuation of air flow is effectively reduced.

**[0037]** However, since the trailing edge 44b of the blade 44 has a shape formed by connecting the ends between the main plate 43 and the side plate 45, noise is still generated by the influence of the trailing vortex. As shown in Fig. 32, the amount of air flow $F_2$, which flows along the curvature of the side plate 45 is small in the entire air flow drawn in from a part of the blade 44 in the vicinity of the side plate 45. In contrast, main flow $F_1$, which is drawn in from the center of the side plate 45, has a great amount. The main flow $F_1$ flows in an area close to the main plate 43 in a concentrated manner because of its velocity vector. Therefore, the flow velocity distribution of the blown out air flow at the outlet of the blade 44 is not constant along the span direction of the blade 44. Also, it is not possible to reduce the trailing vortex generated at a position downstream of the trailing edge 44b of the blade 44. As a result, noise is still generated by the influence of the trailing vortex.

Patent Document 1: Japanese Laid-Open Patent Publication No. 2002-339897
Patent Document 2: Japanese Laid-Open Patent Publication No. 2005-155510
Patent Document 3: Japanese Laid-Open Patent Publication No. 5-60096
Patent Document 4: Japanese Laid-Open Patent Publication No. 2001-132687

DISCLOSURE OF THE INVENTION

**[0038]** Accordingly, it is an objective of the present invention to provide a three-dimensional blade for a centrifugal fan, and a centrifugal fan that further reduces noise.

**[0039]** To achieve the foregoing objectives, and in accordance with a first aspect of the present invention, a centrifugal fan is provided that includes a plurality of blades having a three-dimensional shape, a main plate, an annular side plate, and blade actuating means for rotating the blades via the main plate. One end face of each blade in the span direction is fixed to the main plate, such that the blades are spaced at predetermined intervals along the circumferential direction. The annular side plate is provided at the other end faces of the blades in the span direction. The radial length of an outer end portion of each blade at a part close to the main plate is set longer than the radial length of an outer end portion of the blade at a part close to the side plate.

**[0040]** This configuration significantly improves the velocity distribution of a main flow portion in an air flow that flows in a concentrated manner at a part of the blade close to the main plate. Accordingly, the static pressure-flow rate characteristics of the fan are improved in the entire flow rate range, which increases the discharged air volume. Also, the specific noise characteristics of the air blower are significantly improved, so that the influence of a trailing vortex generated at the trailing edge of a part of each three-dimensional blade close to the main plate is relatively reduced. As a result, noise caused by trailing vortices is effectively reduced.

**[0041]** It is preferable that the radial length of the outer end portion of each blade close to the main plate be set longer than the radial length of the outer end portion of each blade close to the side plate, so that air flow flowing past each blade effectively receives work from the blade at a part of the blade close to the main plate, and that the velocity of the air flow along the span direction of each blade is effectively developed at a part of the blade close to the main plate.

**[0042]** This configuration significantly improves the velocity distribution of a main flow portion in air flow that flows in a concentrated manner at a part of the blade close to the main plate. Accordingly, the static pressure-flow rate characteristics of the fan are improved over the entire flow rate range, which increases the discharged air volume. Also, the specific noise characteristics of the air blower are significantly improved, so that the influence of a trailing vortex generated at the trailing edge of a part of each three-dimensional blade close to the main plate is relatively reduced. As a result, noise caused by trailing vortices is effectively reduced.

**[0043]** It is preferable that the radial length of the outer end portion of each blade close to the main plate be set longer than the radial length of the outer end portion of each blade close to the side plate by extending the trailing edge of the blade rearward with respect to the air flow.

**[0044]** Extending the trailing edge of the blade rearward with respect to the air flow significantly improves the velocity distribution in the main flow portion of air flow that flows in a concentrated manner at a part of the blade close to the main plate. Accordingly, the static pressure-flow rate characteristics of the fan are improved in the entire flow rate range, which increases the blown air volume. Also, the specific noise characteristics of the air blower are significantly improved, so that the influence of a trailing vortex generated at the trailing edge of a part of each three-dimensional blade close to the main plate is relatively reduced. As a result, noise caused by trailing vortices is effectively reduced.

**[0045]** It is preferable that the trailing edge be extended such that its length is gradually increased toward the main plate.

**[0046]** According to this configuration, the trailing edge of each blade has a tapered shape that becomes wider from the side plate toward the main plate. Therefore, the shape of the trailing edge of each blade is a suitable shape that is adapted to changes in the velocity distribution of the main flow that flows in a concentrated manner at a part of the blade close to the main plate. The substantially tapered shape, which becomes gradually wider from the side plate toward the main plate, may be a shape that changes linearly or a shape that changes in a curve.

**[0047]** It is preferable that the trailing edge be extended in a curved manner, and that the trailing edge have a bulge in a part close to the main plate such that there is at least one inflection point in the curved part.

**[0048]** In the centrifugal fan, air flow in the vicinity of the main plate forms a laminar flow shear layer due to the viscosity and the influence of the wall surface of the main plate. This can narrow the passage of the main flow and lower the fan performance. However, the above configuration suppresses the development of the shear layer, thereby improving the fan performance.

**[0049]** The trailing edge is preferably extended rearward by a length that corresponds to the velocity distribution of the main flow of the air flow at the trailing edge.

**[0050]** According to this configuration, the shape of the extended trailing edge is further suitable for changes in the velocity distribution of the main flow, so that the fan performance is further improved.

**[0051]** It is preferable that a step that is extended forward by a predetermined length be formed at a part of the leading edge of the blade close to the main plate.

**[0052]** According to this configuration, when air flow drawn into the impeller through the air inlet is blown outward from the trailing edges of the blades, the air flow is effectively prevented from separating from the negative pressure surfaces of the blades. This further effectively reduces noise of the air blower.

**[0053]** The diameter of the main plate is preferably increased in accordance with the extension of the blades.

**[0054]** By extending the radius of the main plate in accordance with the extension of the trailing edge of each blade, the structural strength of the impeller of the centrifugal fan is also increased.

**[0055]** The centrifugal fan is preferably configured as an air blower of the indoor unit of an air conditioner.

**[0056]** The air blower of an air conditioner indoor unit is required to have a large air volume and to be silent because of its intended use. Therefore, a centrifugal fan according to the present invention is optimal as the air blower of an air conditioner indoor unit because of its small size, high air flow performance, and low noise.

**[0057]** As described above, the present invention provides a compact and quiet centrifugal fan that generates a large air volume and is therefore suitable as an air blower for an air conditioner indoor unit.

## BRIEF DESCRIPTION OF THE DRAWINGS

**[0058]**

Fig. 1 is a perspective view showing an air conditioner using a centrifugal fan according to one embodiment of the present invention;

Fig. 2 is a longitudinal cross-sectional view showing the air conditioner of Fig. 1;

Fig. 3 is a perspective view showing the impeller of the centrifugal fan of Fig. 2;

Fig. 4 is a side view showing the impeller of Fig. 3;

Fig. 5 is a bottom view showing the impeller of Fig. 3;

Fig. 6 is a side view showing a blade of Fig. 3;

Fig. 7 is a projection view of the blade of Fig. 6, as viewed from the side corresponding to the side plate;

Fig. 8 is a perspective view of the blade of Fig. 6, as viewed from above and the left rear;

Fig. 9 is a side view of the blade of Fig. 6, with section lines 10A-10A to 10E-10E;

Figs. 10(a) to 10(e) are cross-sectional views corresponding to respective section lines of Fig. 9;

Fig. 11 is a cross-sectional view showing the comparison between the shape of the trailing edge of the blade according to the present embodiment and the shape of the trailing edge of a prior art blade;

Fig. 12 is a diagram showing the operation (characteristics) of the blade of Fig. 6;

Fig. 13 is a graph showing changes in the static pressure coefficient of four examples using flow rate coefficient as a parameter, to measure the effects of the blade according to the present embodiment, where (a) represents a case of the prior art fan of Figs. 25 to 32, (b) represents a case where the outer diameter of the prior art fan is increased as a whole by 5%, (c) represents a case where the outer diameter of only a lower part (part close to the main plate) of the blade of the present embodiment is increased by 5%, and (d) represents a case where the outer diameter of only a lower part (part close to the main plate) of the blade of the present embodiment is increased by 10%;

Fig. 14 is a graph showing changes in air flow noise of the four examples of Fig. 13 using the discharged air volume as a parameter to measure the effects of the blade according to the present embodiment;

Fig. 15 is a graph showing changes in specific noise of the four examples of Fig. 13 using the blown air volume as parameter, to measure the effects of the blade according to the present embodiment;

Fig. 16 is a diagram showing operation of a blade according to a first modification, in which the shape of the trailing edge of the blade according to the present embodiment is changed in a curved manner;

Fig. 17 is a diagram showing operation of a blade according to a second modification, in which the shape of the trailing edge of the blade according to the present embodiment is changed to a shape that reduces shearing force;

Fig. 18 is a longitudinal cross-sectional view showing an air conditioner that uses a centrifugal fan according to a third modification, in which steps are provided at the leading edge of each blade according to the present embodiment, in a part that faces the main plate;

Fig. 19 is a side view showing a blade of the impeller of the air conditioner;

Fig. 20 is a perspective view of the blade of Fig. 19, as viewed from above and the left rear;

Fig. 21 is a diagram showing the operation of the blade of Fig. 19;

Fig. 22 is a graph showing changes in specific noise of four examples using a flow rate coefficient as a parameter, to measure the effects of the blade of Fig. 19, where (a) represents a case of the prior art fan of Figs. 25 to 32 without steps, (b) represents a case where steps are formed in the prior art blade, (c) represents a case of the present embodiment of Figs. 1 to 12, and (d) represents the third modification of the present embodiment shown in Figs. 18 to 21;

Fig. 23 is a graph showing changes in static pressure coefficient of the four examples using a flow rate coefficient as a parameter, to measure the effects of the blade of Fig. 19;

Fig. 24 is a diagram showing operation of a blade according to a fourth modification of the present embodiment;

Fig. 25 is a longitudinal cross-sectional view showing an air conditioner using a prior art centrifugal fan;

Fig. 26 is a perspective view showing the impeller of the centrifugal fan of Fig. 25;

Fig. 27 is a side view showing a blade of Fig. 26;

Fig. 28 is a perspective view of the impeller of Fig. 26, as viewed diagonally from the rear;

Fig. 29 is a projection view of the blade of Fig. 28;

Fig. 30 is a side view of the blade of Fig. 28, with section lines;

Figs. 31(a) to 31(e) are cross-sectional views respectively along lines 31A-31A to 31 E-31 E of Fig. 30; and

Fig. 32 is a diagram showing the operation of the blade of Fig. 28.

## BEST MODE FOR CARRYING OUT THE INVENTION

**[0059]** Hereinafter, a centrifugal fan according to one embodiment of the present invention and an air conditioner using the centrifugal fan will be described with reference to the drawings.

### (1) Overall Structure of Air Conditioner

**[0060]** Fig. 1 is a perspective view showing the appearance of an air conditioner 1 using a centrifugal fan according to one embodiment of the present invention (ceiling part is omitted). The air conditioner 1 is a ceiling embedded type, and includes a casing 2 accommodating various components, and a decorative panel 3 provided underneath the casing 2. More specifically, the casing 2 of the air conditioner 1 is fitted in an opening formed in a ceiling U of an air conditioned room as shown in Fig. 2 (a longitudinal cross-sectional view of the air conditioner 1), and a decorative panel 3 is arranged along the ceiling U.

**[0061]** The casing 2 is a box-like body having a lower opening. When viewed in the plan view, the casing 2 has an octagonal shape with short sides and long sides arranged alternately. The casing 2 has a substantially octagonal top plate 21 with alternately arranged long sides and short sides, and a side wall plate 22 extending downward from the periphery of the top plate 21.

**[0062]** The decorative panel 3 is shaped as a rectangular plate in the plan view. The panel 3 has an air inlet

31 and air outlets 32. The air inlet 31 is substantially located at the center of the decorative panel 3 and draws air from the air conditioned room. The air outlets 32 discharges air from the casing 2 into the air conditioned room. The decorative panel 3 is arranged such that each of its sides corresponds to one of the long sides of the top plate 21 of the casing 2.

**[0063]** The air inlet 31 is a substantially square opening. In contrast, each air outlet 32 is an elongated rectangular opening along a side of the decorative panel 3. The air inlet 31 includes an air intake grill 33 and a filter 34 for removing dust in the air drawn into the air inlet 31.

**[0064]** Further, a horizontal flap 35 is provided at each air outlet 32. The horizontal flap 35 can be swung about an axis extending along the longitudinal direction of the air outlet 32. The horizontal flap 35 is a rectangular blade-like member extending along the longitudinal direction of the air outlet 32. Each horizontal flap 35 is supported at both ends in the longitudinal direction with a support pins (not shown) so as to be rotatable. The horizontal flap 35 is thus capable of changing the direction of flow of air discharged from the air outlet 32.

**[0065]** An air blower 4 and a heat exchanger 6 are arranged in the casing 2. The air blower 4 draws air into the casing 2 from the air conditioned room through the air inlet 31 of the decorative panel 3. The heat exchanger 6 is arranged to surround the air blower 4.

**[0066]** The air blower 4 is a backwardly curved blade fan, which is a centrifugal fan to which the present invention is applied. The air blower 4 includes a fan motor (impeller driving means) 41 and an impeller 42 coupled to the shaft (rotary shaft) of the fan motor 41. The fan motor 41 is provided at the center of the top plate 21 of the casing 2 and is oriented downward. The detailed structure of the impeller 42 will be discussed below.

**[0067]** The heat exchanger 6 is a cross-inner-tube type heat exchanger, which is formed by bending a tube to surround the air blower 4. The heat exchanger 6 is connected to an outdoor unit (not shown) installed outside the house through refrigerant piping. The heat exchanger 6 functions as an evaporator during the cooling operation, and as a condenser during the heating operation. The heat exchanger 6 performs heat exchange with air that is drawn into the casing 2 by the air blower 4 through the air inlet 31, so as to cool the air during the cooling operation, and heat the air during the heating operation.

**[0068]** A drain pan 7 is provided below the heat exchanger 6. The drain pan 7 receives drain water, which is produced when water in air is condensed on the surface of the heat exchanger 6. The drain pan 7 is attached to the lower part of the casing 2. The drain pan 7 has an air inlet portion 71 communicating with the air inlet 31 of the decorative panel 3, air outlet portions 72 corresponding to the air outlets 32 of the decorative panel 3, and a drain water receiving groove 73 that is formed to surround the lower part of the heat exchanger 6 to receive drain water.

**[0069]** A bellmouth 5 is provided at the air inlet portion 71 of the drain pan 7. The bellmouth 5 guides air drawn in through the air inlet 31 of the decorative panel 3 to the impeller 42 of the air blower 4.

**[0070]** As described above, the air conditioner 1 of the present embodiment has an air passage from the air inlet 31 of the decorative panel 3 to the four air outlets 32 via the filter 34, the bellmouth 5, the drain pan 7, the air blower 4, and the heat exchanger 6. The air conditioner 1 draws in air in the air conditioned room and performs heat exchange between the drawn in air and the refrigerant at the heat exchanger 6. Thereafter, the air conditioner 1 discharges air, the temperature of which has been adjusted, to the four sides of the air conditioned room through the air passage.

(2) Structure of Impeller of Centrifugal Fan

**[0071]** The structure of the impeller 42 of the centrifugal fan 4 will be described with reference to Figs. 2 to 5. Fig. 3 is a perspective view showing the appearance of the impeller 42. Fig. 4 is a side view of the impeller 42 shown in Fig. 3.Further, Fig. 5 is a diagram showing the impeller 42 installed as shown in Fig. 4, as viewed from above.

**[0072]** The impeller 42 is mainly formed by a disk-shaped main plate 43, an annular side plate 45 separated from the main plate 43, and a plurality of blades 44 arranged between the main plate 43 and the side plate 45. The main plate 43 is coupled to a shaft 41a of the fan motor 41. The blades 44 are arranged along the main plate 43 at predetermined angular intervals about the shaft 41 a of the fan motor 41. The rotation direction of the impeller 42 is shown by arrow R in Fig. 5.

**[0073]** The main plate has an outer diameter Db'. The main plate 43 is a synthetic resin member. A substantially truncated cone shaped bulge 43a is formed at the center of the main plate 43. The bulge 43a protrudes toward the air inlet 31. A main plate cover 46 is fixed at a position spaced from the lower surface of the main plate 43. The main plate cover 46 covers a cooling air hole. Radially arranged guide blades 46a are provided on the side of the main plate cover 46 that faces the main plate 43. The guide blades 46a cause part of the air blown out of the impeller 42 to flow about the main plate 43 by means of the difference between the static pressure in the space between the main plate 43 and the casing 2 and the static pressure in the space between the main plate 43 and the side plate 45. This part of the discharged air passes through the space about the fan motor 41 to cool the fan motor 41. Thereafter, the air is blown to the space inside the impeller 42 through the cooling air hole of the main plate 43 and the guide blades 46a of the main plate cover 46.

**[0074]** The side plate has an outer diameter Da. The side plate 45 has a shaper that gradually decreases from the outer circumference toward the center opening. The side plate 45 is a bell shaped resin member that protrudes toward the air inlet 31.

(3) Structure of Impeller Blade

**[0075]** The structure of the blades 44 of the impeller 42 will be described with reference to Figs. 6 to 11. Fig. 6 is a side view of the blade 44 as seen from the direction of the negative pressure surface. Fig. 7 is a projection view of the blade 44 of Fig. 6, as viewed from above. Fig. 8 is a perspective view of the blade 44 of Fig. 6, as viewed from above and the left rear. Fig. 9 is a side view of the blade 44 of Fig. 6, with section lines 10A-10A to 10E-10E from a lower part to an upper part (from the end close to the main plate 43 to the end close to the side plate 45). Figs. 10(a) to 10(e) are cross-sectional views taken along lines 10A-10A to 10E-10E of Fig 9, respectively. Fig. 11 is a cross-sectional view showing a characteristic of the blade 44 (difference from the prior art shape shown in Figs. 25 to 32). Fig. 11 is a comparison between Fig. 10 and the cross-sectional view of Fig. 31 taken along line 31B-31B.

**[0076]** The blades 44 are resin members that are molded separately from the main plate 43 and the side plate 45. One end face of each blade 44 is fixed to the main plate 43, and another end face is fixed to the side plate 45. When viewed from the side of the impeller 42, the end of the blade 44 that faces the side plate 45 is inclined rearward than the end close to the main plate 43 as shown in Fig 28. The blade 44 is formed such that these ends cross each other to substantially form an X as shown in Fig. 29. In other words, the blade 44 has a three-dimensional shape that extends parallel to the rotary shaft while being twisted between the main plate 43 and the side plate 45.

**[0077]** The edge at the leading side in the rotation direction of the blade 44, which is a three-dimensional blade, or a leading edge 44a extends from the end close to the main plate 43 to a predetermined position close to the side plate 45 with substantially a constant radius. The blade 44 as an inclined edge from the predetermined position close to the side plate 45, while retreating outward with a gradually decreasing radius. The end of the blade 44 on the opposite side of the rotation direction R, or the trailing edge 44b, has a shape different from that of the prior art blade. Unlike the prior art shown in Figs. 25 to 32, the trailing edge 44b does not have a shape formed by a straight line (perpendicular to the main plate 43) that extends parallel to the rotation axis from the end close to the main plate 43 and the end close to the side plate 45. For example, as shown in Figs. 6 to 11, the trailing edge 44b is extended rearward with respect to air flow such that the degree of extension increases from the end close to the side plate 45 toward the main plate 43. Accordingly, the radial length (Rb' in Fig. 5) of the trailing edge 44b of the main plate 43 of the blade 44 through which main flow $F_1$ of the centrifugal fan flows is set longer than the radial length (Ra in Fig. 5) of the trailing edge 44b at a part close to the side plate 45 of the blade 44. Thus, the velocity distribution of air flow along the span direction of the blade 44 effectively develops in a part close to the main plate 43, at which the flow rate is greater.

**[0078]** The trailing edge 44b is extended without changing the basic shape of the three-dimensional blade, but rather in conformity with the shape of the three-dimensional blade. In this case, using the original outer diameter Db of the main plate 43 (the outer diameter Da of the side plate 45) as a reference, the outer diameter Db' (Fig. 12) of the expanded main plate 43 is preferably determined such that the amount of extension of the trailing edge 44b is less than or equal to 10% of the original diameter Db. In other words, the amount of extension of the trailing edge 44b is preferably less than or equal to 10% of the outer diameter Da of the side plate 45.

**[0079]** In the case of the configuration of Figs. 6 to 12, the outer diameter Db' of the main plate 43 is enlarged by approximately 5% compared to the outer diameter Db of the prior art main plate 43 shown in Figs. 28 to 32.

**[0080]** When extending the trailing edge 44b, the inlet angle, the outlet angle, the attaching angle, and the skew angle of a blade element of the blade 44 that is close to the main plate 43 are maintained to the same values as those in the original blade 44 shown in Figs. 28 to 32. Therefore, as shown in, for example, the perspective view of Fig. 8, the blade 44 of the present embodiment is enlarged at the attachment position on the side corresponding to the trailing edge 44b bulges in the direction of rotation.

**[0081]** Also, in accordance with the extension of the radius at the end of the blade 44 close to the main plate 43, the outer diameter of the main plate 43 is increased.

**[0082]** As described above, since the radius of the main plate 43 is increased in accordance with the extension of the trailing edge 44b of each blade 44, the structural strength of the impeller 42 of the centrifugal fan is also increased.

**[0083]** In accordance with the above described configuration, the velocity distribution of the main flow $F_1$, which flows in a concentrated manner in the vicinity of the main plate 43, is significantly improved compared to the flow $F_2$ in the vicinity of the side plate 45. Accordingly, the static pressure-flow rate characteristics (P-Q characteristics) of the blade 44 is improved over the entire flow rate range, which increases the discharged air volume. Also, the specific noise characteristics are significantly improved, so that the influence of a trailing vortex generated at the trailing edge 44b of a part of each three-dimensional blade 44 close to the main plate 43 is relatively reduced. As a result, the noise caused by trailing vortices is effectively reduced.

**[0084]** In the present embodiment, the trailing edge 44b is extended further rearward toward the main plate 43. As shown in Fig. 12, the trailing edge 44b is also extended so that it becomes gradually longer from the side plate 45 toward the main plate 43. That is, the trailing edge 44b of the blade 44 has a tapered shape that is linearly enlarged from the side plate 45 toward the main plate 43. Accordingly, the velocity distribution at the outlet of the blade 44 effectively develops in the main flow $F_1$. Therefore, the influence of the wake flow generated at a part of the trailing edge 44b close to the main plate 43 is relatively weakened.

**[0085]** Thus, since the trailing edge 44b is enlarged into a tapered shape, the shape of the trailing edge 44b is more suitable for changes in the velocity distribution of the main flow $F_1$, in which the flow rate increases toward the main plate 43. That is, the shape of the blade 44 is optimized for the velocity distribution of air flow, so that the fan performance is improved.

**[0086]** Therefore, by using a centrifugal fan according to the present embodiment, a compact and low noise air conditioner having a large air volume can be produced at low cost.

**[0087]** Particularly, in the impeller of the centrifugal fan according to the present embodiment, each blade has the three-dimensional shape as described above (see Figs. 7 and 8). Thus, compared to blade shapes formed based on two-dimensional blade element as in the prior art, the load distribution on the surface of the blade 44 and the pressure fluctuation of air flow through between the blades 44 are significantly improved.

**[0088]** Therefore, if such a three-dimensional blade is combined with the blade 44 of the present embodiment and operates as described above, trailing vortices are further effectively eliminated.

**[0089]** As a matter of course, several techniques have been known that improve the velocity distribution of air flow at the outlet of the blade 44, as described in the prior art section. For example, Patent Document 3 discloses a blade that is shortened by cutting out a part close to the side plate. This reduces the flow separation at the part of the blade close to the side plate, thereby equalizing the velocity distribution at the outlet of the blade.

**[0090]** However, because of the cut out portion in the part of each blade adjacent to the side plate, the length of the blade is less than a blade having no cut out portion. This reduces the amount of work that each blade applies to the air flow. According to the present embodiment, the blade 44 itself is not shortened. On the contrary, the area of the blade 44 is increased. The present embodiment there does not have such a drawback and effectively increases the work of the blade 44.

**[0091]** This technique is effective in a centrifugal fan in which the blade width is sufficiently small in relation to the outer shape, as described in Patent Document 3. Specifically, cutting out of the blades prevent separation of drawn-in air flow in a part of each blade in the vicinity of the side plate, and the air flow is allowed to flow along the part. However, the technique is not necessarily effective in a normal centrifugal fan, which has wide blades, that is, not in a case where the influence of separation near the side plate is not dominant.

**[0092]** Also, if a part of the blade near the side plate is cut out, air flow that passes through the part interferes with corners of a part of the blade near the main plate. This can generate noise having discretely-distributed frequency spikes. According to the present embodiment, the basic shape of the blade 44 itself is not changed at all. Thus, the present embodiment does not have these drawbacks of the prior art.

**[0093]** Another technique has been known in which the blades are perpendicular to the main plate and the side plate, and the thickness of each blade is changed from the end face adjacent to the main plate to the end face adjacent to the side plate. Accordingly, fluctuation of flow velocity distribution at the outlet of the blades is suppressed. In some prior art configurations, the outer diameter of a part of the blade close to the main plate and the outer diameter of a part of the blade close to the side plate are made different, so that the shape of each blade is increased toward the main plate (for example, refer to Patent Document 4). This reduces the velocity fluctuation of the flow at the blade wake.

**[0094]** However, the ratio of the diameter of the impeller at the side close to the main plate to the diameter of the impeller at the side close to the side face is set to 1.2 to 1.6. Therefore, when the blades are enlarged with a small increase in the length, noise reduction effect cannot be achieved. In contrast, when the length is excessively increased, the flow characteristics will be degraded.

**[0095]** As a result, the diameter of the impeller at the end close to the main plate is twenty percent greater than the diameter at the side plate. This structure is not advantageous over the technique in which the size of the fan is simply increased. Enlargement of the size by twenty percent or more goes entirely against desired goals that have been sought in the first place in the prior art, which are size reduction and lowered noise.

**[0096]** However, such drawbacks are effectively eliminated by the present embodiment.

**[0097]** In the present embodiment, the trailing edge 44b is extended further rearward toward the main plate 43. As shown in Fig. 12, the trailing edge 44b is also extended so that it becomes gradually longer from the side plate 45 toward the main plate 43.

**[0098]** According to this configuration, the trailing edge 44b of each blade 44 has a tapered shape that becomes wider from the side plate 45 toward the main plate 43. As shown in Fig. 12, the trailing edge 44b is more suitable for changes in the velocity distribution of the main flow $F_1$, which flows in a concentrated manner in the vicinity of the main plate 43. In this case, in accordance with the extension of the radius at a part of the trailing edge 44b of the blade 44 close to the main plate 43, the outer diameter of the main plate 43 is increased.

**[0099]** As described above, since the radius of the main plate 43 is increased in accordance with the extension of the trailing edge 44b of each blade 44, the structural strength of the impeller 42 of the centrifugal fan is also increased.

<Examples>

**[0100]** The flow characteristics of two prior art Examples a, b and two Examples c, d according to the present invention were measured. Example c was an impeller having three-dimensional blades, or the blades 44 of the present embodiment in which the enlargement factor of the outer diameter Db' of the main plate 43 (extension factor of the width of the trailing edge 44b rearward with respect to air flow) was 5%. Example d was an impeller having three-dimensional blades, or the blades 44 of the present embodiment in which the enlargement factor of the outer diameter Db' of the main plate 43 was 10%. The prior art Example a, or comparison example, was an impeller having the prior art blades shown in Figs. 27 to 32. The prior art Example b was equivalent to the fan of the prior Example a in which the diameter of the fan was increased by 5%.

**[0101]** In the Examples c and d, a part of each blade 44 close to the main plate 43 (the lower part as viewed in Figs. 6 to 9) was extended by 5% and 10% rearward with respect to the air flow, respectively. The main flow $F_1$, which dominantly determined the fan performance, passed through this part. This increased the amount of work applied to air flow by the blades 44, so that the volume of air blown discharged from the impeller was effectively increased. Thus, compared to the prior art Example a or the prior art Example b, in which the diameter of the fan was increased by 5% as a whole, the static pressure-flow rate characteristics (P-Q characteristics) were increased over the entire flow rate range in the Examples c and d as shown in Fig. 13. Also, in the Examples c and d, the air flow noise was reduced except for a low air volume range as shown in Fig. 14.

**[0102]** Further, the improvement of the two properties significantly improved the specific noise characteristics of the Examples c and d, for example, as shown in Fig. 15. The properties of the Examples c and d were clearly improved in comparison with the prior art Example b, in which the diameter of the fan was evenly increased.

**[0103]** In comparison with the blade of Patent Document 4, the size of which was increased without changing the shape, the same advantages were achieved in a range of the enlargement factor less than or equal to 10% (5%). Therefore, the impeller according to the present embodiment is superior in terms of increased air volume and lowered noise, and is capable of being reduced in size.

<First Modification>

**[0104]** The shape of enlargement (extension) of the trailing edge 44b of the blade 44 is not limited to the linear shape (tapered shape corresponding to a linear function) as described above. For example, as shown in Fig. 16, the extension may be shaped as a curve of a quadratic function for a parabola.

**[0105]** This curved shape of the trailing edge 44b is selected in consideration of the deviation of the velocity distribution of air flow discharged from the impeller. Specifically, the trailing edge 44b of the blade 44 is extended further in the radial direction toward the main plate 43 from the side plate 45.

**[0106]** According to this configuration, the trailing edge 44b of each blade 44 has a tapered shape of a quadratic function that becomes wider in an arcuate manner from the side plate 45 toward the main plate 43. The trailing edge 44b of the blade 44 is suitable for the velocity distribution of the main flow $F_1$, which flows in a concentrated manner in the vicinity of the main plate 43 as shown in Fig. 16.

**[0107]** That is, the shape of the trailing edge 44b of the blade 44 is further effectively optimized for the velocity distribution of the main flow $F_1$, so that the fan performance is improved.

<Second Modification>

**[0108]** In the centrifugal fan as described above, air flow in the vicinity of the main plate 43 forms a laminar flow shear layer due to the influence of the viscosity at the wall surface of the main plate. This can narrow the passage of the main flow $F_1$ and lower fan performance.

**[0109]** To deal with such a problem, the trailing edge 44b may be formed such that there is at least one inflection point in the curved part as shown in Fig 17. In this case, the trailing edge 44b has a bulge in the vicinity of the main plate 43. Taking into consideration the velocity distribution of the air flow at the outlet, the enlargement of the trailing edge 44b at the end face close to the main plate 43 is slightly shorter than the maximally extended part (but, at least greater than or equal to the end of the trailing edge 44b close to the side plate 45).

**[0110]** This configuration suppresses the development of the shear layer, thereby improving the fan performance.

<Third Modification>

**[0111]** As shown in Figs. 18 to 21, the leading edge 44a of the blade 44 may be formed stepwise (in the present embodiment, two steps) toward the inside of the impeller 42. Specifically, the leading edge 44a may have first and second steps 44c, 44d. When air flow drawn into the impeller through the air inlet 31 and the bellmouth 5 shown in Fig. 1 is blown to the outside by the blade 44, the first and second steps 44c, 44d prevent the air flow from separating from the negative pressure surface of the blade 44. In this manner, the first and second steps 44c, 44d contribute to further reduction of air flow noise of the air blower 4.

**[0112]** The negative pressure surface refers to a surface of the blade 44 that faces inward of the impeller 42. The surface opposite to the negative pressure surface, that is, a surface of the blade 44 that faces outward of the impeller 42 is referred to as a positive pressure surface.

**[0113]** The lengths La·Lb, Lc·Ld of the first and second steps 44, 44d are set to be 0.09 to 0.18 times the original chord lengths $L_1$, $L_2$, $L_3$ of the blade 44 in the span direction. That is, the length of the first step 44c, which is the lower step, varies along the span direction in a range between 0.15 times the length $L_1$ (La) and 0.2 times the length $L_1$ (Lb), and the length of the second step 44d, which is the upper step, varies along the span direction in a range between 0.08 times the length $L_2$ (Lc) and 0.1 times the length $L_3$ (Ld).

**[0114]** The blade 44 according to the third modification with trapezoid steps will be discussed as an Example d. The above described example of Figs. 1 to 12 with none of the first and second steps 44c, 44c on the leading edge 44a will be expressed as an example c (without trapezoid steps). A comparison Example b is the one with the first and second steps 44c, 44d formed in the prior art blade. When Example c and Example d are compared, the specific noise characteristics are lowered in the entire flow region. The Example c achieves a reduction of at least 1.1 [dBA] at the minimum specific noise characteristic point. This is because when the impeller is rotating, the protruding shapes of the first and second steps 44c, 44d of the leading edge 44a generate a longitudinal vortex in the air flow prior to flowing into the blade 44 as a reference, and the longitudinal vortex prevents the air flow from separating from the leading edge 44a.

**[0115]** The centrifugal fan according to the embodiment shown in Figs. 1 to 12 is designed such that air flow that passes through the impeller flows below the blade 44 compared to the prior art Example a. Therefore, the longitudinal vortex generated by the first and second steps 44c, 44d of the leading edge 44a further effectively prevents separation.

**[0116]** As shown in Fig. 23, the Example c is inferior to the comparison Example b in terms of the aerodynamic characteristics in the low air volume region. In the Example d, the first and second steps 44c, 44d improved the characteristics in the low air volume region compared to the comparison Example b. Accordingly, in comparison with the comparative Example b, the aerodynamic characteristics of the Example d were improved in the entire air volume range. Consequently, the centrifugal fan using the Example d according to the third modification of the present invention was capable of generating a greater amount of air volume at the same static pressure.

**[0117]** As a result, the centrifugal fan according to the third modification of the present embodiment generates lower noise than a conventional centrifugal fan, and is capable of increasing the air volume at the same static pressure. This enables development of a compact and quiet fan.

**[0118]** Accordingly, the use of the centrifugal fan realizes a compact air conditioner that runs quietly while maintaining greater air volume.

<Fourth Modification>

**[0119]** As shown in Figs. 24, the first and second steps 44c, 44d of the third modification may be combined with the second modification.

<Application of Centrifugal Fan of Present Invention>

**[0120]** Generally, biased velocity distribution of the discharged air flow at a part of each blade close to the main plate is commonly seen in any type of centrifugal fan. Therefore, the present invention can be applied to impellers in centrifugal fans of various types (for example, backwardly-curved blade types, sirocco types, and straight radial blade types). In these cases, the fan performance is sufficiently improved.

## Claims

1. A centrifugal fan comprising a plurality of blades having a three-dimensional shape, a main plate, an annular side plate, and blade actuating means for rotating the blades via the main plate, one end face of each blade in the span direction being fixed to the main plate, such that the blades are spaced at predetermined intervals along the circumferential direction, the annular side plate being provided at the other end faces of the blades in the span direction, the radial length of an outer end portion of each blade at a part close to the main plate being set longer than the radial length of an outer end portion of the blade at a part close to the side plate.

2. The centrifugal fan according to claim 1, wherein the radial length of the outer end portion of each blade close to the main plate is set longer than the radial length of the outer end portion of each blade close to the side plate, so that air flow flowing past each blade effectively receives work from the blade at a part of the blade close to the main plate, and that the velocity of the air flow along the span direction of each blade is effectively developed at a part of the blade close to the main plate.

3. The centrifugal fan according to claim 1 or 2, wherein the radial length of the outer end portion of each blade close to the main plate is set longer than the radial length of the outer end portion of each blade close to the side plate by extending the trailing edge of the blade rearward with respect to the air flow.

4. The centrifugal fan according to claim 3, wherein the trailing edge is extended such that its length is gradually increased toward the main plate.

5. The centrifugal fan according to claim 4, wherein the trailing edge is extended in a curved manner, and the trailing edge has a bulge in a part close to the main plate such that there is at least one inflection point in the curved part.

**6.** The centrifugal fan according to claim 3, wherein the trailing edge is extended rearward by a length that corresponds to the velocity distribution of the main flow of the air flow at the trailing edge.

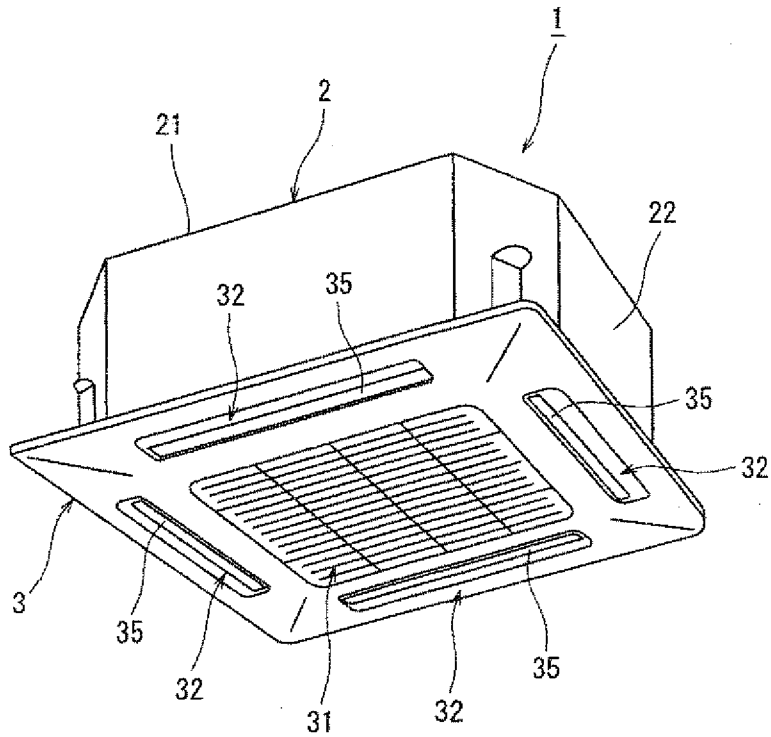
**7.** The centrifugal fan according to claim 6, wherein the amount of extension of the trailing edge is less than or equal to 10% of the outer diameter of the side plate.

**8.** The centrifugal fan according to any one of claims 1 to 7, wherein a step that is extended forward by a predetermined length is formed at a part of the leading edge of the blade close to the main plate.

**9.** The centrifugal fan according to any one of claims 1 to 8, wherein the diameter of the main plate is increased in accordance with the extension of the blades.

**10.** The centrifugal fan according to any one of claims 1 to 9, wherein the centrifugal fan is configured as an air blower of an indoor unit of an air conditioner.

# Fig.1



# Fig.2

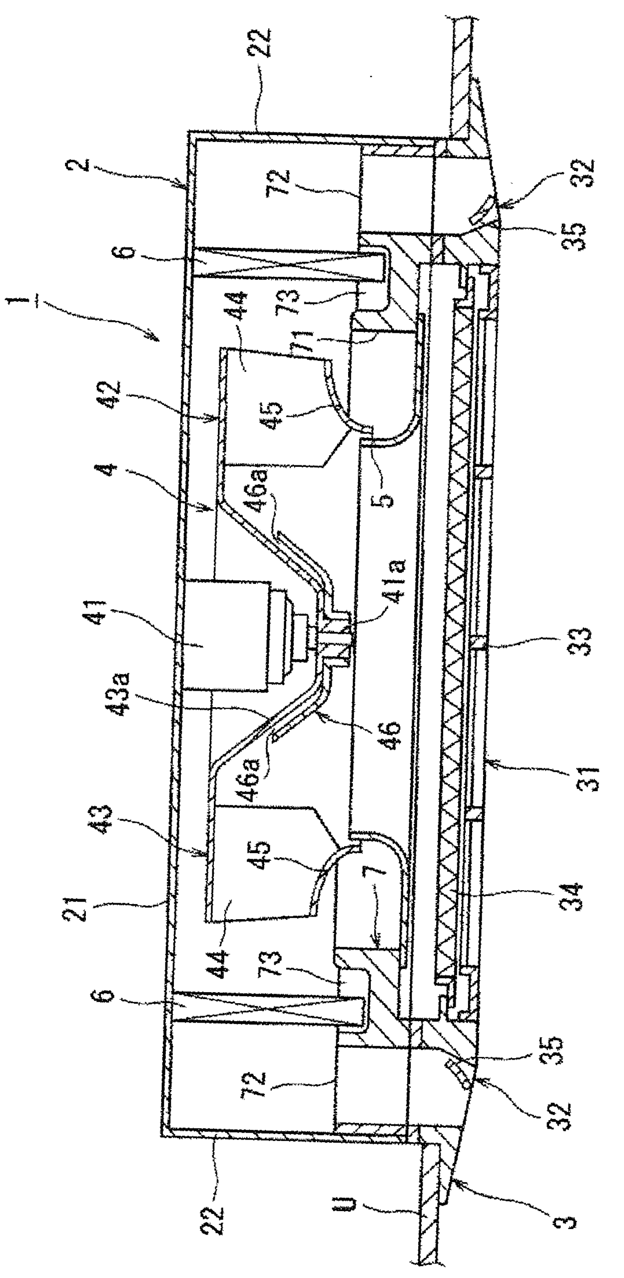


Fig.3

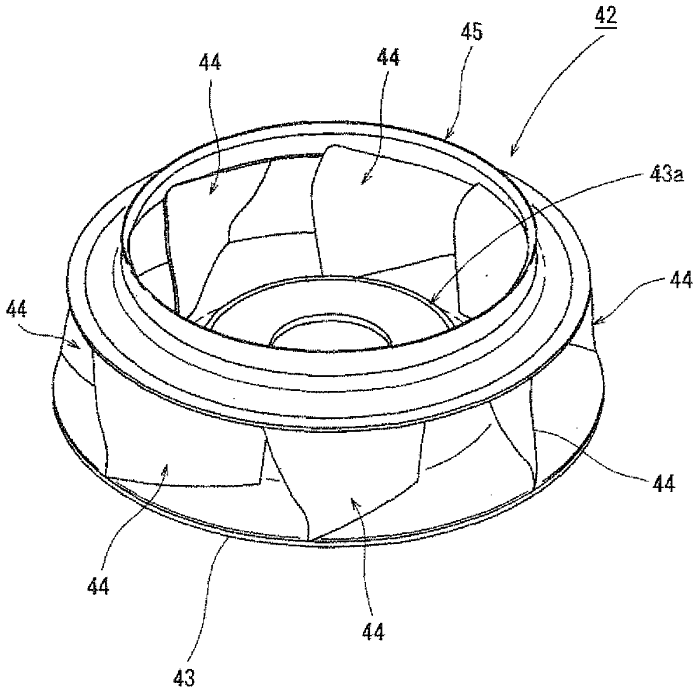
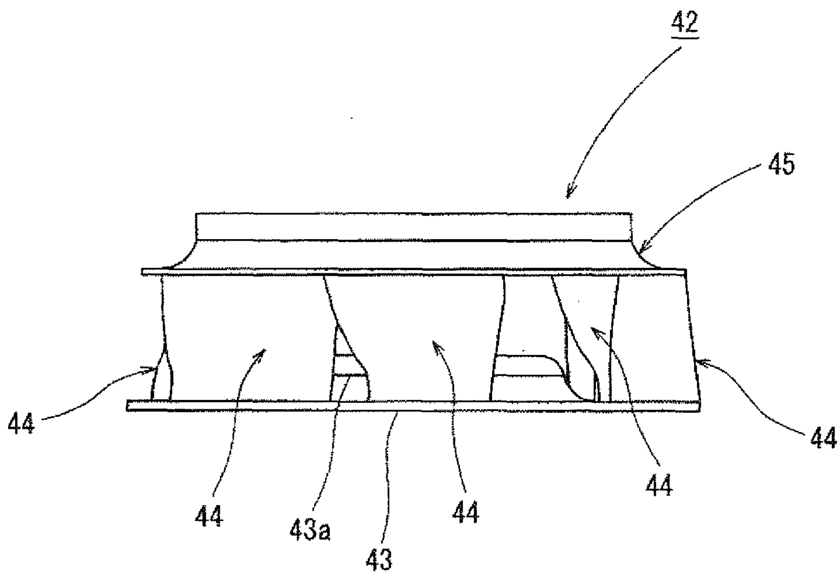
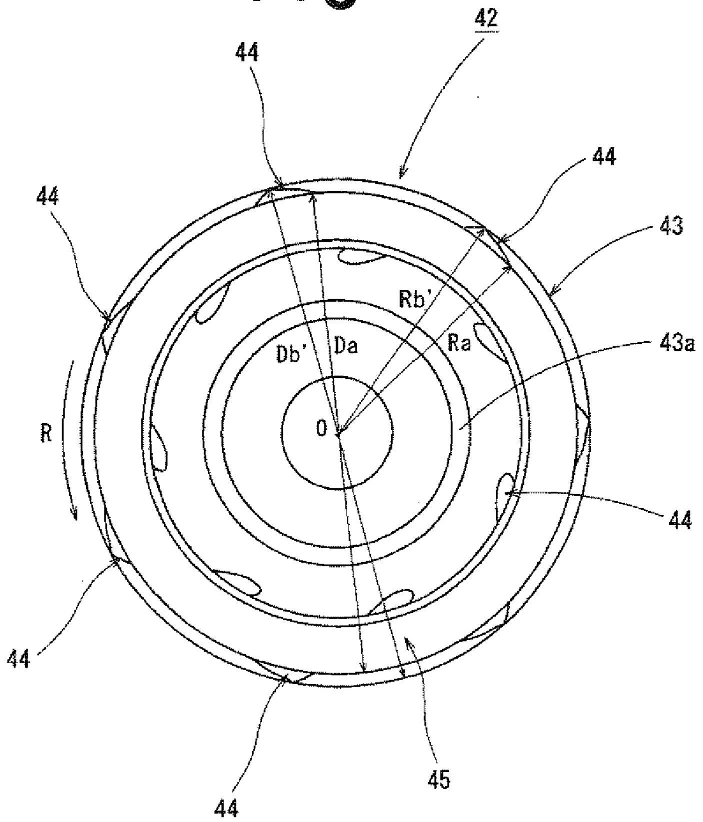


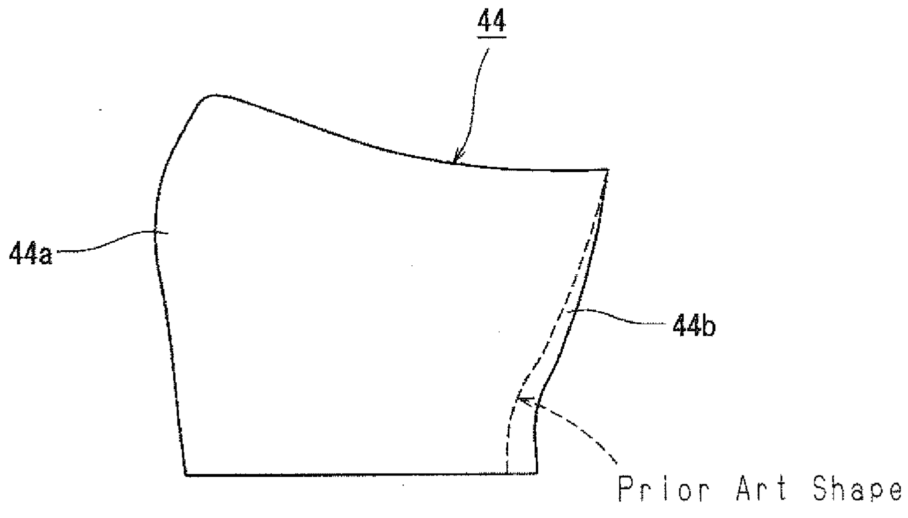

Fig.4



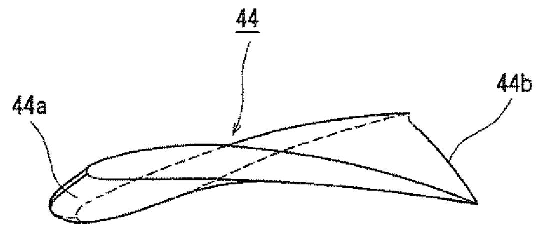
# Fig.5


42
44
44
44
43
Rb'
Db'
Da
Ra
43a
0
R
44
44
44
45

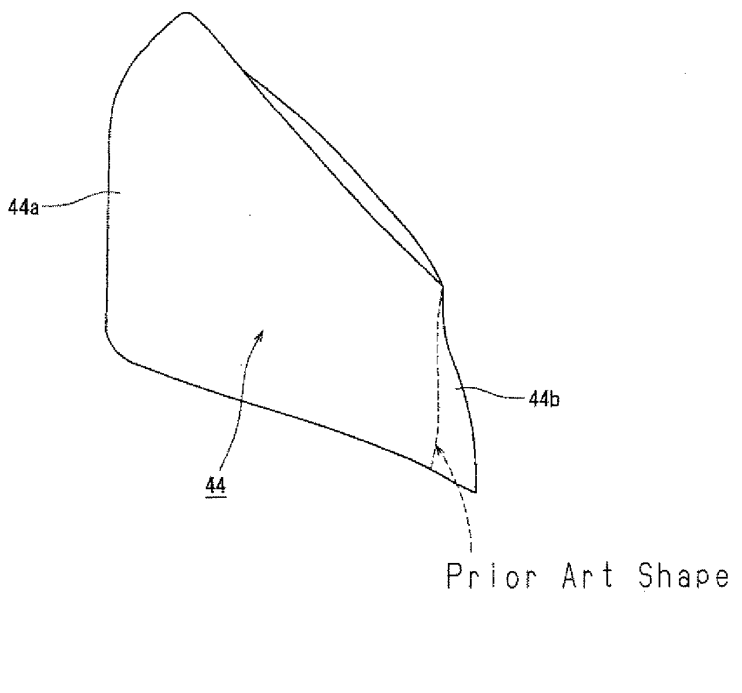
# Fig.6


44
44a
44b
Prior Art Shape

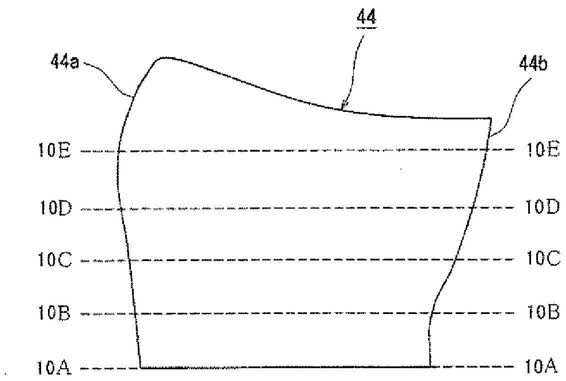
# Fig.7



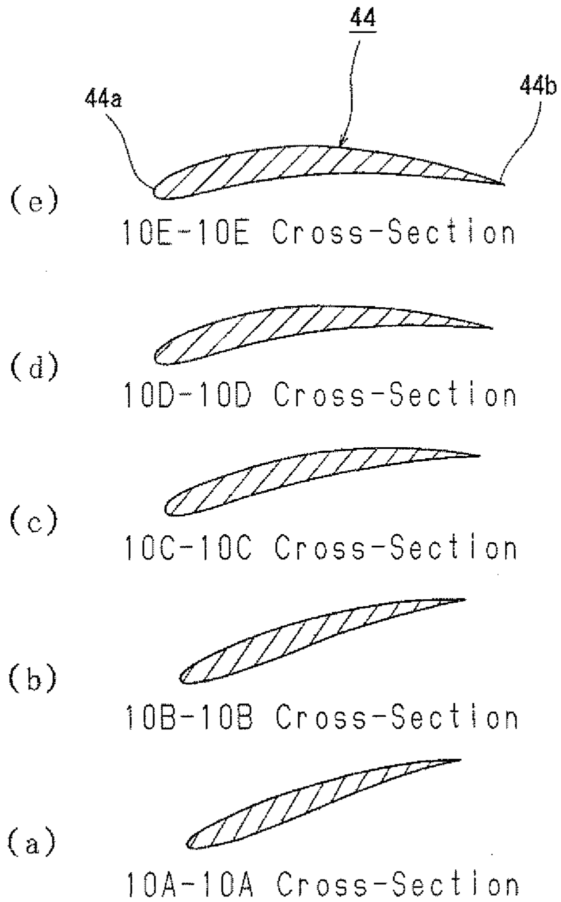

# Fig.8



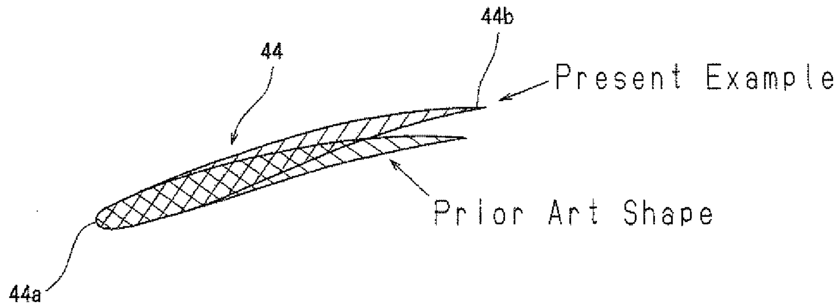
# Fig.9


44
44a
44b
10E
10E
10D
10D
10C
10C
10B
10B
10A
10A

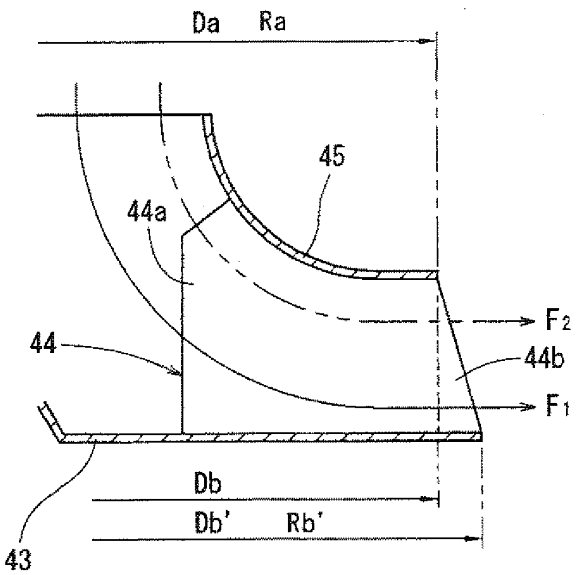
# Fig.10


44
44a
44b
(e)
10E-10E Cross-Section
(d)
10D-10D Cross-Section
(c)
10C-10C Cross-Section
(b)
10B-10B Cross-Section
(a)
10A-10A Cross-Section

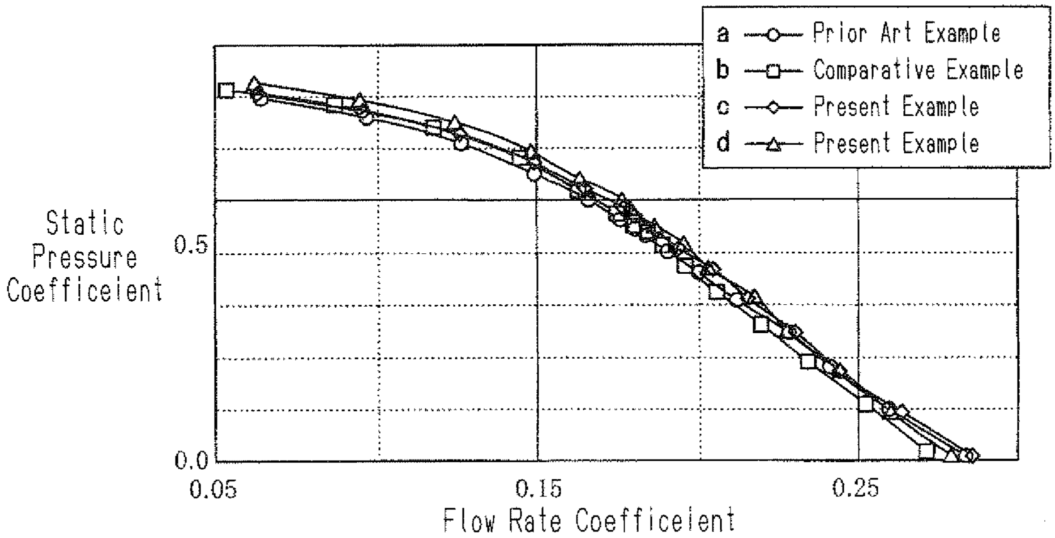
# Fig.11


44b
44
Present Example
Prior Art Shape
44a

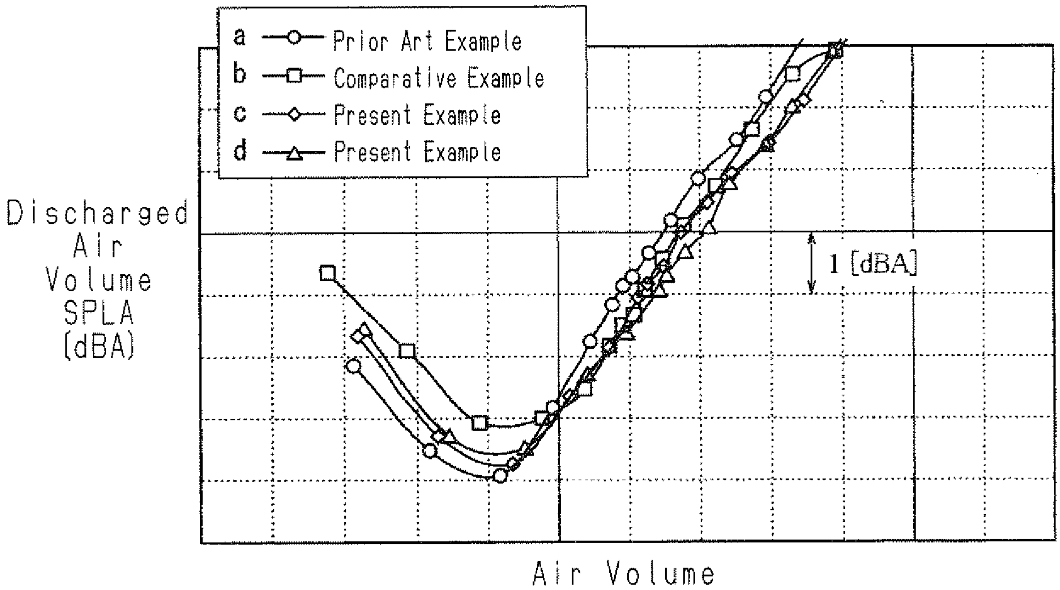
# Fig.12


Da Ra
45
44a
F2
44
44b
F1
Db
Db' Rb'
43

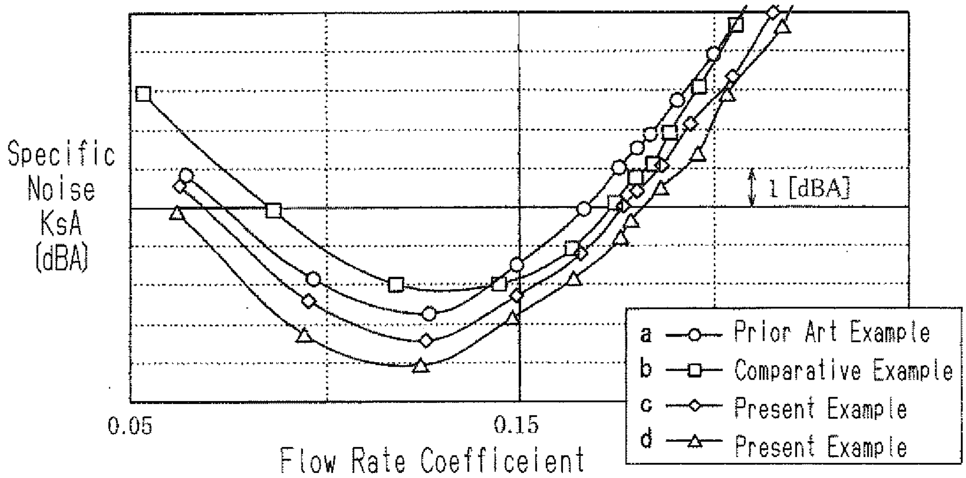
# Fig.13


a Prior Art Example
b Comparative Example
c Present Example
d Present Example
Static Pressure Coefficeient
0.5
0.0
0.05
0.15
0.25
Flow Rate Coefficeient

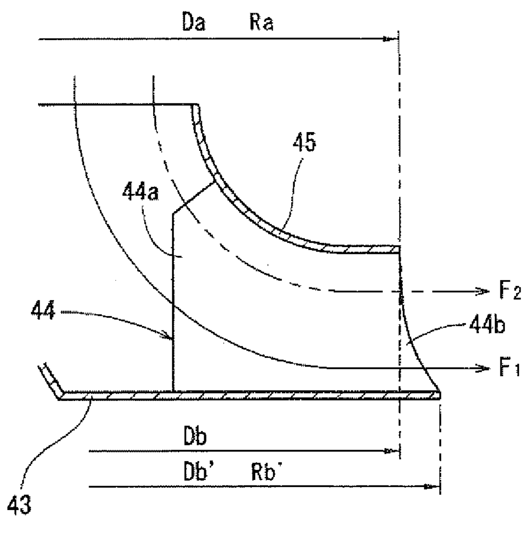
# Fig.14


a Prior Art Example
b Comparative Example
c Present Example
d Present Example
Discharged Air Volume SPLA (dBA)
1 [dBA]
Air Volume

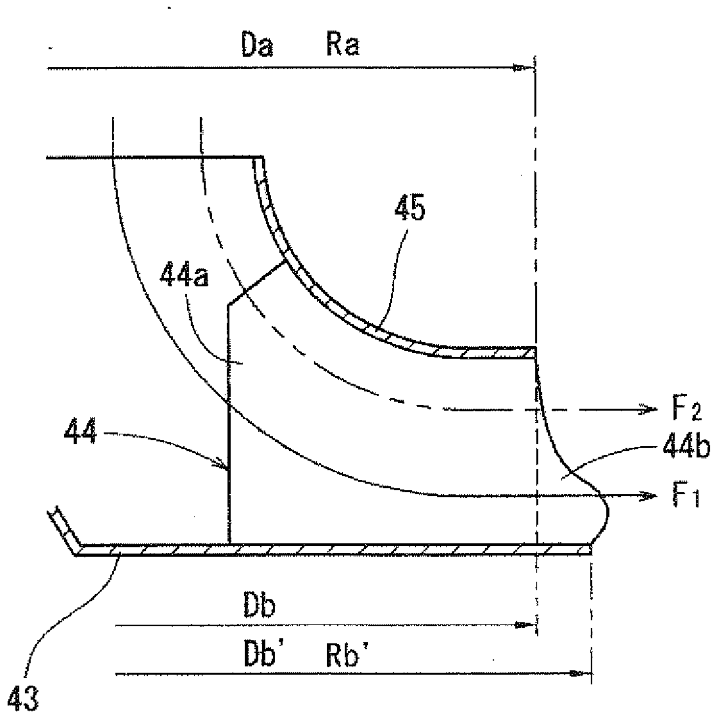
# Fig.15


Specific Noise KsA (dBA)
1 [dBA]
0.05
0.15
Flow Rate Coefficeient
a Prior Art Example
b Comparative Example
c Present Example
d Present Example

# Fig.16


Da
Ra
45
44a
F2
44
44b
F1
Db
Db'
Rb'
43

# Fig.17



# Fig.18

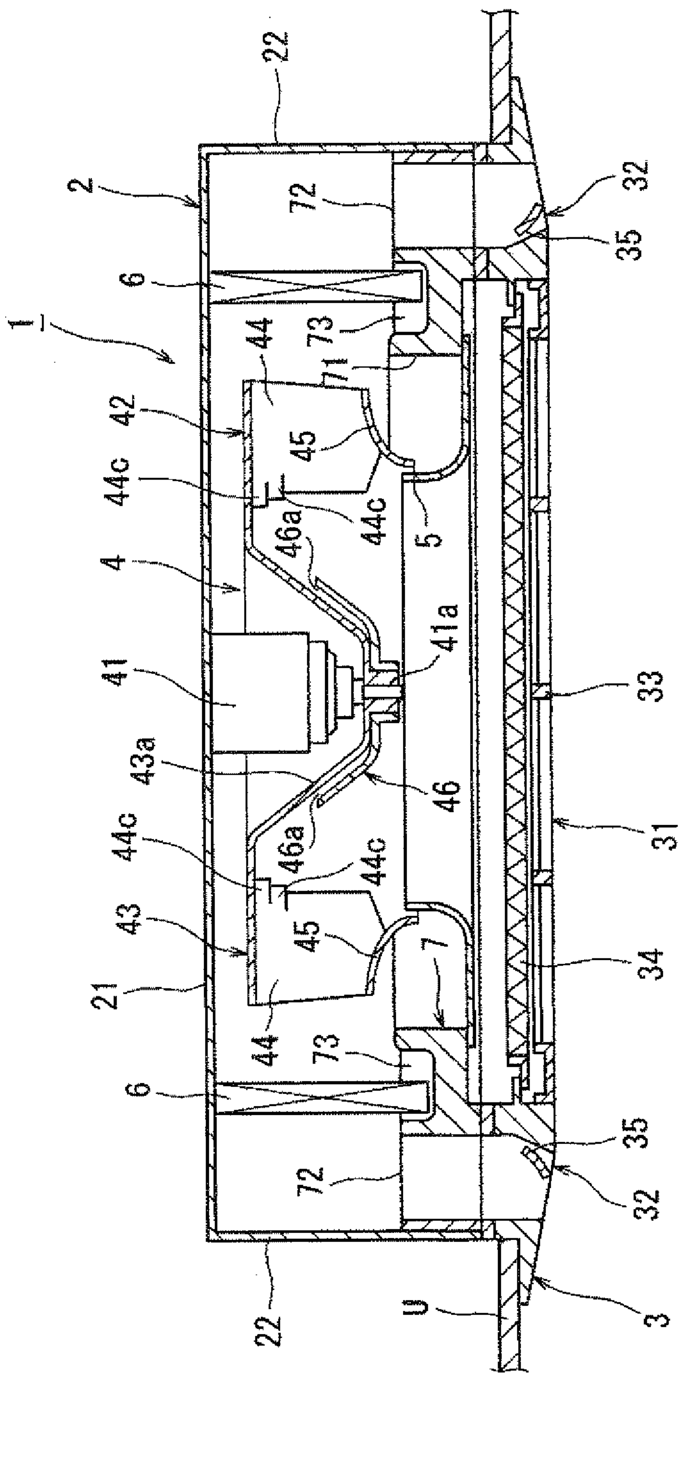

22
2
72
32
35
6
73
1
44
71
42
45
44c
44c
5
4
46a
41a
41
33
43a
46
46a
44c
44c
31
43
45
7
34
21
44
73
6
35
72
32
22
U
3

Fig.19

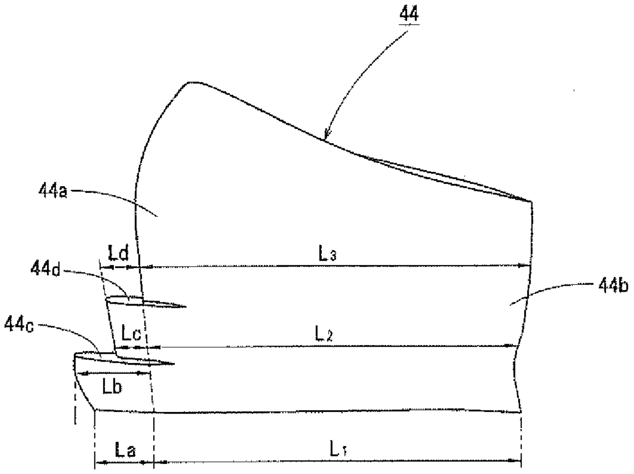
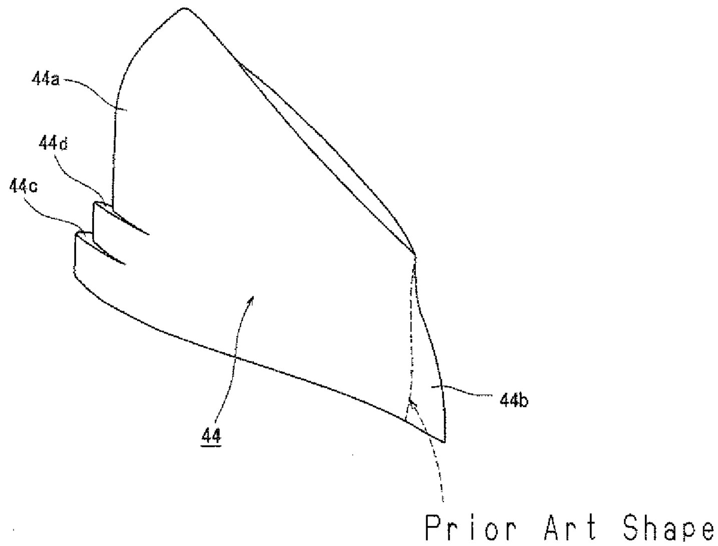
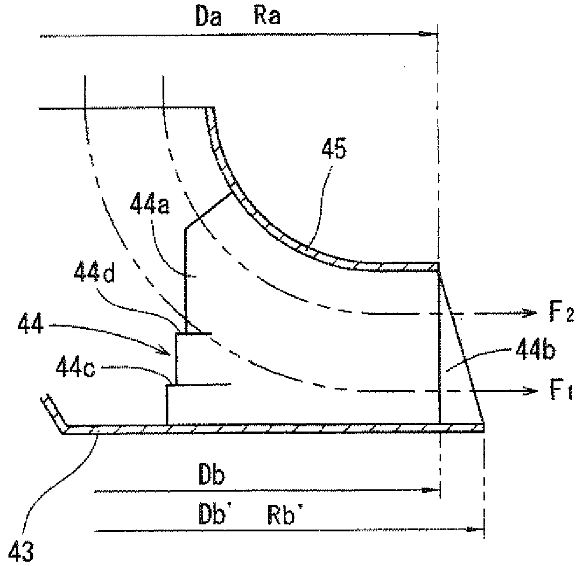

44
44a
44d
44c
44b
Ld
Lc
Lb
La
L3
L2
L1

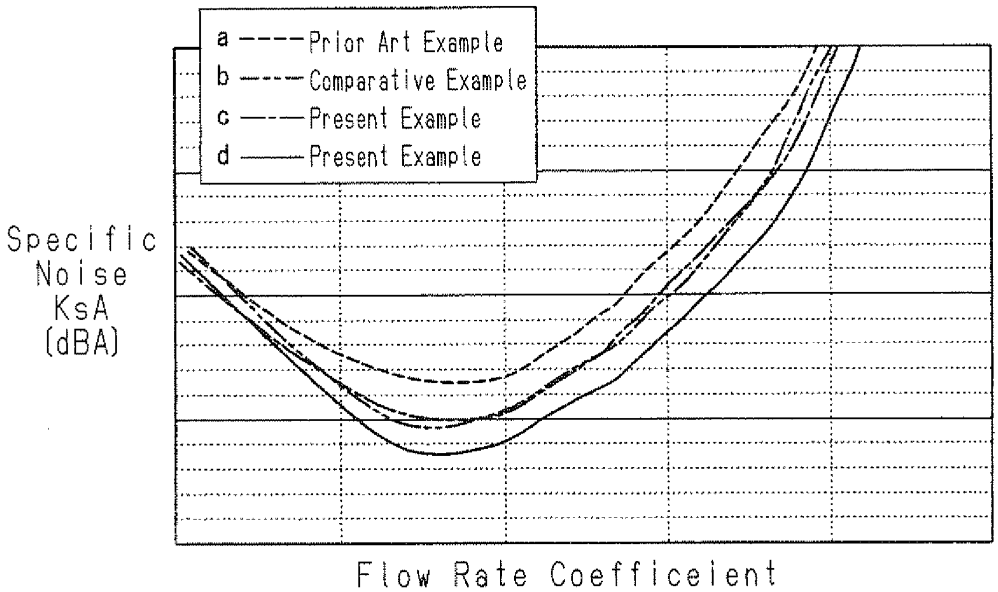
Fig.20


44a
44d
44c
44
44b
Prior Art Shape

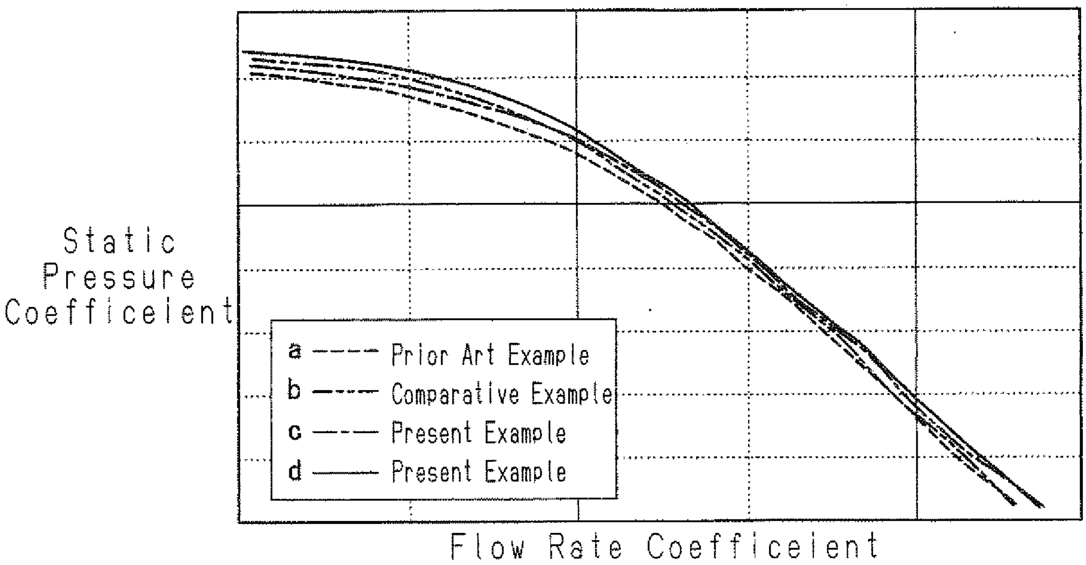
# Fig.21


Da Ra
45
44a
44d
44
44c
F2
44b
F1
Db
Db' Rb'
43

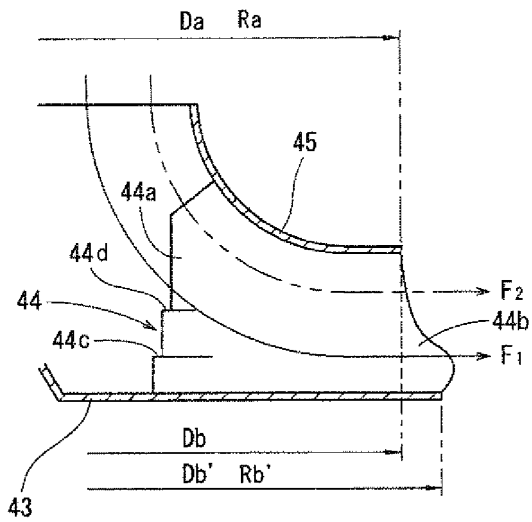
# Fig.22


a Prior Art Example
b Comparative Example
c Present Example
d Present Example
Specific Noise KsA (dBA)
Flow Rate Coefficeient

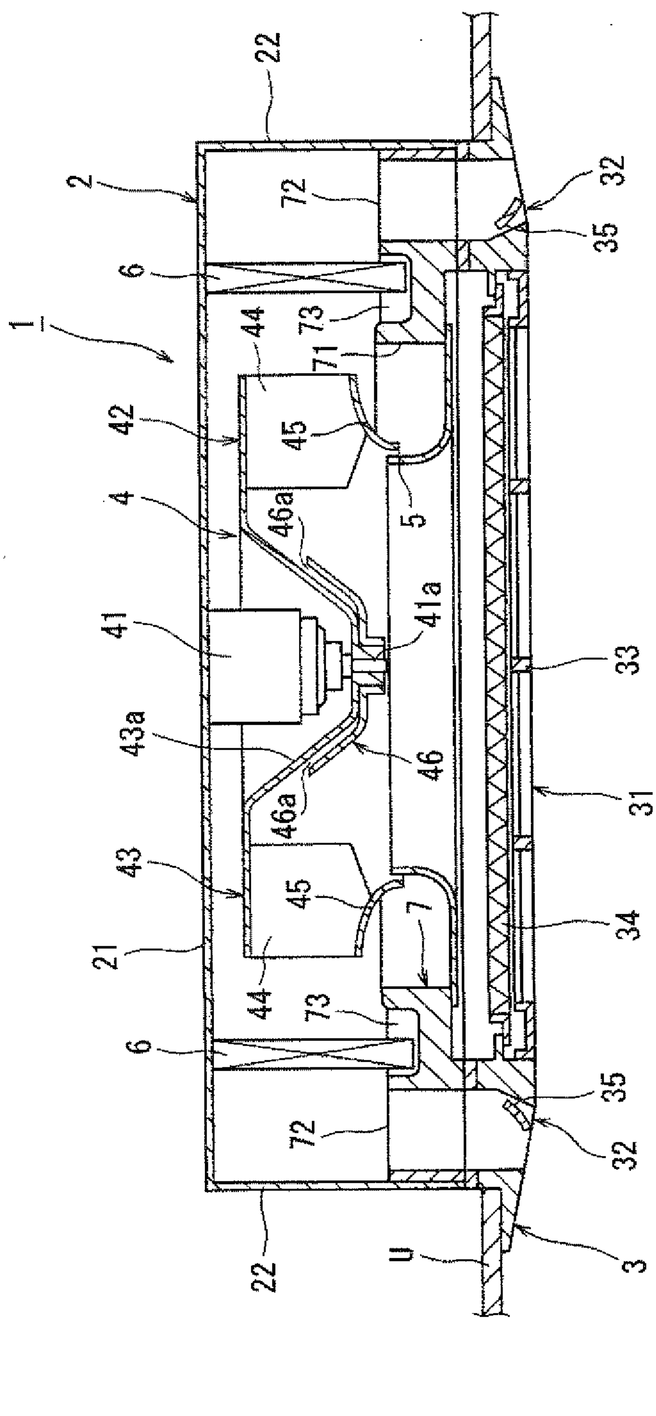
# Fig.23


Static Pressure Coefficeient
a Prior Art Example
b Comparative Example
c Present Example
d Present Example
Flow Rate Coefficeient

# Fig.24


Da Ra
45
44a
44d
44
44c
F2
44b
F1
Db
Db' Rb'
43

# Fig.25


22
2
72
32
35
6
1
44
73
71
42
45
5
4
46a
41
41a
33
43a
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31
43
34
21
7
44
73
6
35
72
32
22
U
3

Fig.26

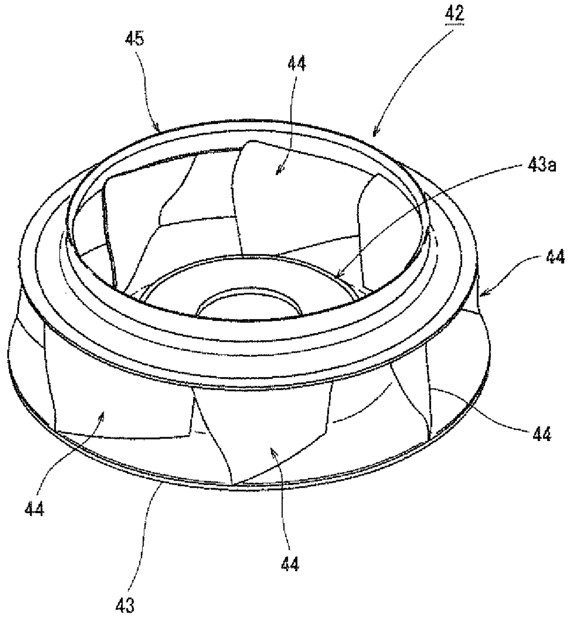
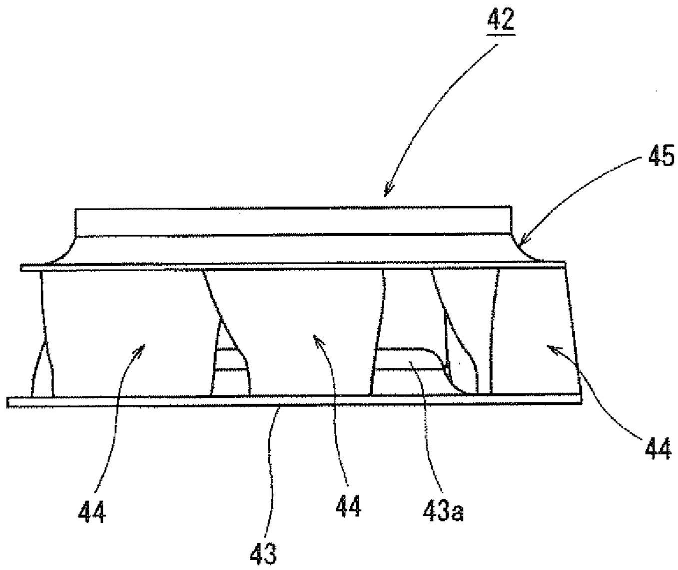
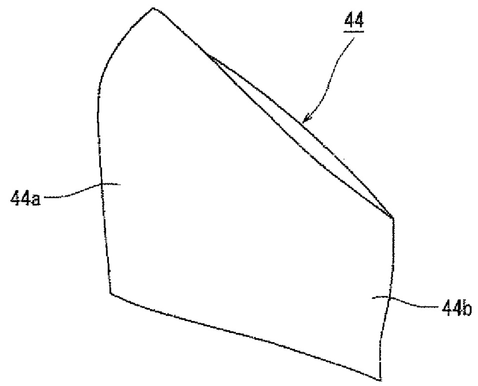


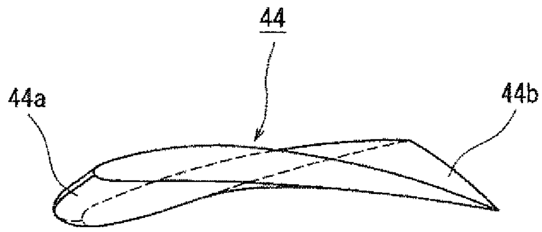

Fig.27



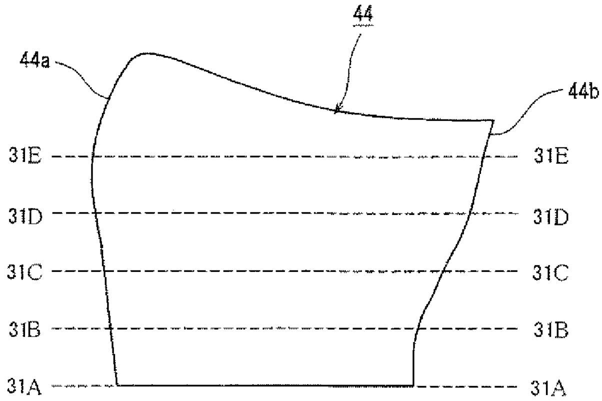
# Fig.28



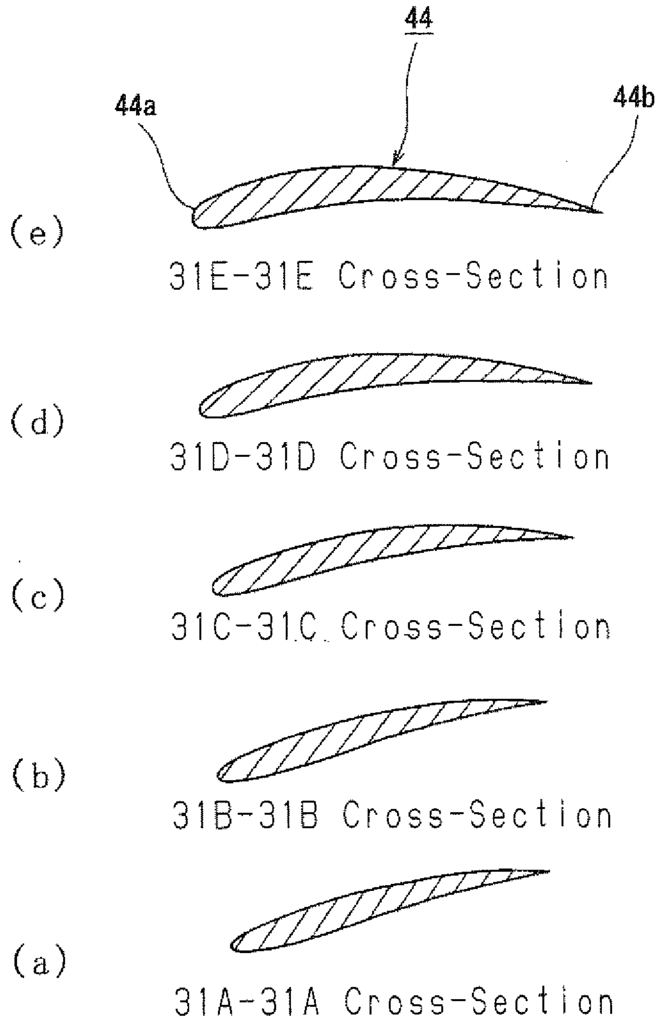

# Fig.29



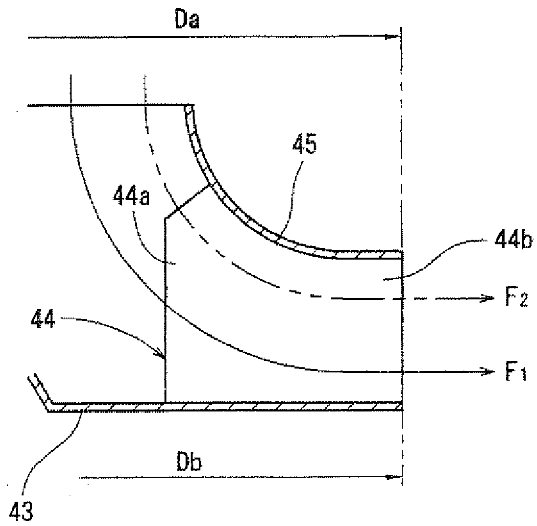

# Fig.30



# Fig.31


44
44a
44b
(e)
31E-31E Cross-Section
(d)
31D-31D Cross-Section
(c)
31C-31C Cross-Section
(b)
31B-31B Cross-Section
(a)
31A-31A Cross-Section

# Fig. 32



## INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2009/058938

A. CLASSIFICATION OF SUBJECT MATTER
*F04D29/30*(2006.01)i

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)
F04D29/30

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

| | | | |
|---|---|---|---|
| Jitsuyo Shinan Koho | 1922-1996 | Jitsuyo Shinan Toroku Koho | 1996-2009 |
| Kokai Jitsuyo Shinan Koho | 1971-2009 | Toroku Jitsuyo Shinan Koho | 1994-2009 |

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. |
|---|---|---|
| Y | JP 3544325 B2 (Mitsubishi Electric Corp.), 21 July, 2004 (21.07.04), Par. No. [0057]; Fig. 17 (Family: none) | 1-10 |
| Y | JP 3055238 B2 (Matsushita Electric Industrial Co., Ltd.), 26 June, 2000 (26.06.00), Claim 1; Fig. 2 (Family: none) | 1-10 |
| Y | JP 09-79184 A (Matsushita Electric Industrial Co., Ltd.), 25 March, 1997 (25.03.97), Claim 1; Fig. 1 (Family: none) | 1-10 |

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

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| Date of the actual completion of the international search | Date of mailing of the international search report |
|---|---|
| 06 July, 2009 (06.07.09) | 14 July, 2009 (14.07.09) |

| Name and mailing address of the ISA/ | Authorized officer |
|---|---|
| Japanese Patent Office | |
| Facsimile No. | Telephone No. |

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INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2009/058938

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

| Category* | Citation of document, with indication, where appropriate, of the relevant passages | Relevant to claim No. |
|---|---|---|
| Y | JP 2005-155510 A (Daikin Industries, Ltd.), 16 June, 2005 (16.06.05), Par. No. [0037]; Fig. 4 & US 2007/0098556 A1 Par. No. [0103]; Fig. 4 | 1-10 |
| Y | JP 2007-239567 A (Daikin Industries, Ltd.), 20 September, 2007 (20.09.07), Par. No. [0053]; Fig. 4 & WO 2007/102476 A1 Par. No. [0029]; Fig. 3 | 1-10 |
| Y | JP 02-70997 A (Matsushita Refrigeration Co.), 09 March, 1990 (09.03.90), Claim 1; Fig. 2 (Family: none) | 6-10 |
| Y | JP 60-156997 A (Nippondenso Co., Ltd.), 17 August, 1985 (17.08.85), Full text; Fig. 4 (Family: none) | 6-10 |
| Y | JP 2002-202095 A (Kioritz Corp.), 19 July, 2002 (19.07.02), Par. No. [0009]; Fig. 6 & US 2002/0000000 A1 Par. No. [0024]; Fig. 6 | 6-10 |

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- JP 2002339897 A **[0037]**
- JP 2005155510 A **[0037]**
- JP 5060096 A **[0037]**
- JP 2001132687 A **[0037]**