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(54) **Axial flow fan**

Axiallüftereinheit

Ventilateur axial

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(73) Proprietor: **LG Electronics, Inc.**
Seoul 150-875 (KR)

(72) Inventors:
• **Sim, Won Chin**
Gwangmyeng-gi
Gyeonggi-do (KR)

• **Moon, Kee Chung**
Yongdungpo-gu
Seoul (KR)

(74) Representative: **Bergquist, Kjell Gunnar et al**
Albihns.Zacco
Torggatan 8
Box 142
401 22 Göteborg (SE)

(56) References cited:
EP-A- 1 087 146 EP-A- 1 298 326
EP-A- 1 422 425 KR-B1- 100 314 960
US-A- 4 930 990 US-A- 5 730 583
US-B1- 6 447 251

EP 1 512 918 B1

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Description

BACKGROUND OF THE INVENTION

1. Field of the Invention

[0001] The present invention relates to an axial flow fan of an air conditioner. An axial flow fan according to the preamble of claim 1 is indicated in US 6,447,251 B1.

2. Background of the Related Art

[0002] In general, an air conditioner is mounted therein with a refrigerating cycle system composed of a compressor, a condenser, a capillary tube, an evaporator and a heat exchanger. The air conditioner is an apparatus for properly sending cold air formed at the evaporator or warm air generated at the condenser according to an indoor condition, and thus genially maintaining indoor atmosphere.

[0003] The air conditioner may be divided into a window type air conditioner where the refrigerating cycle system is mounted in a single body, a split type air conditioner where an indoor unit and an outdoor unit are separated and installed indoors and outdoors respectively, and so forth. Particularly, the split type air conditioner is again divided, according to an installation method, into a wall-mounted type, a permanent-mounted type (including a package air conditioner), a ceiling-mounted type, a ceiling-embedded type and so on. Especially, the indoor unit of the split type air conditioner may have a structure capable of alternatively using the wall-mounted type and the permanent-mounted type and being simultaneously applied as the ceiling-mounted type according to need of a user, which is referred to as a convertible type indoor unit.

[0004] Fig. 1 schematically shows a general air conditioner.

[0005] Referring to Fig. 1, the conventional air conditioner is composed of an outdoor unit 20 which is disposed outdoors and exchanges heat with outdoor air, an indoor unit 10 which is disposed indoors and conditions indoor air, and a connecting line 30 which connects the outdoor unit and the indoor unit with each other.

[0006] To be more specific, the outdoor unit 20 is a means for converting a gas refrigerant of low temperature and low pressure, which is inputted from the indoor unit 10 by exchanging heat with the outdoor air, into a liquid refrigerant of low temperature and low pressure, and is composed of a compressor 11, a condenser 12 and an expansion valve 13.

[0007] Further, the compressor 11 is a component for converting the gas refrigerant of low temperature and low pressure, which is inputted from the indoor unit 10, into the gas refrigerant of high temperature and high pressure, and the condenser 12 is a component for converting the gas refrigerant of high temperature and high pressure into a liquid refrigerant of middle temperature and

high pressure, and the expansion valve 13 is a component for converting the liquid refrigerant of middle temperature and high pressure into the liquid refrigerant of low temperature and low pressure.

5 [0008] Here, the condenser 12 is a component for directly exchanging the heat with the outdoor air, and has a separate fan for introducing the outdoor air.

[0009] Meanwhile, the indoor unit 10 lowers an indoor temperature by means of evaporation, which occurs when the liquid refrigerant of low temperature and low pressure introduced from the outdoor unit 20 is converted into the gas refrigerant of low temperature and low pressure.

10 [0010] The indoor unit 10 is composed of an evaporator 21 and a fan 21a, wherein the evaporator 21 converts the liquid refrigerant of low temperature and low pressure into the gas refrigerant of low temperature and low pressure. The connecting line 30 is a component for connecting the indoor unit 10 and the outdoor unit 20 to circulate the refrigerant, and is properly disposed according to a distance between the outdoor unit 10 and the indoor unit 10.

[0011] As set forth above, the outdoor unit 20 of the split-type air conditioner includes the compressor, the condenser, a cooling fan (hereinafter, referred to as "axial flow fan") which usually generate many noises, and a driving motor for rotating the axial flow fan. The indoor unit 10 includes the evaporator 21 and the blow fan 21a, and performs refrigeration and circulation of the indoor air.

20 [0012] Fig. 2 is a perspective view illustrating a general split type air conditioner.

[0013] As shown in Fig. 2, the indoor unit 10 and the outdoor unit 20 are connected to each other by the connecting line 30.

25 [0014] Meanwhile, the axial flow fan 40, as shown in Fig. 3A, has a hub 42 coupled to a rotational shaft of the driving motor (not shown), and a plurality of blades 44 formed on an outer circumferential surface of the hub 42, wherein the hub 42 is integrally formed with the blades 44.

[0015] When the axial flow fan 40 is rotated by the driving motor, a pressure difference is generated between front and rear sides of the plurality of blades 44 formed on the outer circumferential surface of the hub 42.

30 [0016] This pressure difference generates a suction force capable of sucking up the air, thus sucking up the outdoor air toward the outdoor unit 20 through the suction. Thus, the outdoor air passes through the condenser 12 provided on an intake side of the outdoor unit. At this point, the outdoor air exchanges the heat with the gas refrigerant flowing through the condenser to condense the gas refrigerant into a liquid state, and then flows out outside the outdoor unit 20 through ventilation of the axial flow fan 40.

35 [0017] As for characteristic factors determining a ventilation characteristic of the axial flow fan 40, they are divided into two types: general factors such as the

number of the blades 44, a (outer) diameter D of the axial flow fan, a (outer) diameter d of the hub and so forth, and so-called blade factors such as a pitch angle β , a peak point of the camber P, a maximum quantity of the camber MC, a length of a chord, a sweep angle α and so forth at the blade, which will be described below with reference to Figs. 3A and 3B.

[0018] The pitch angle β of the blade, as in Fig. 3B, is an angle between a flow direction of the fluid or the air (x-axis in the figure) and a straight line, namely a chord, running from a leading edge (L.E) of the blade 44 and its trailing edge (T.E).

[0019] Here, the quantity of the camber refers to a length joining the camber (a central line across a cross section of the blade) and the chord. The maximum point of the camber quantity, i.e., the maximum quantity of the camber MC, as in Fig. 3B, refers to the camber quantity between the L.E. of the blade 44 and the camber peak point P on the chord C running from the L.E to the T.E.

[0020] The sweep angle α refers to an angle between two lines that intersect, one of which is one which connects the center of an inner end of the blade 44 or the center of a portion where the blade 44 comes into contact with the hub 42 but goes with a curvature of the blade 44, and the other is one (Y axis in the figure) which passes through the center (point) of the inner end of the blade 44 and the center (point) of the hub 42.

[0021] Especially, the sweep angle α is a factor determining a noise of an airflow of the axial flow fan 40. When the sweep angle α is great, a phase difference of the airflow between the hub 42 and a tip of the blade 44 becomes great. In contrast, when the sweep angle α is small, the phase difference of the airflow becomes small.

[0022] The phase difference of the airflow causes a phase difference between a noise generated at the outer end of the blade 44 and a noise generated at the inner end of the blade 44. The greater this noise phase difference is, the lower a frequency of the airflow passing through the blade 44 becomes. Hence, the noise becomes lower.

[0023] And, the number of the blades 44 is an important factor determining the airflow noise generated when the axial flow fan 40 is operated.

[0024] One example of this conventional axial flow fan 40 is disclosed in Korean Patent Publication No. 2003-14960, titled *AXIAL FLOW FAN OF OUTDOOR UNIT OF AIR CONDITIONER*, previously filed by the present applicant and published as of February 20, 2003. As for the disclosed axial flow fan of the outdoor unit of the air conditioner, it includes a hub 42 connected with a rotational shaft of a motor and a plurality of blades 44 integrally formed on an outer circumferential surface of the hub, wherein the number of the blades 44 is set to three, a whole outer diameter of the fan is set to 340 ± 2 mm, and a diameter of the hub 42 is set to 100 ± 2 mm.

[0025] Further, each blade 44 is configured so that the pitch angle β is linearly changed from the hub 42 to the

end thereof in a range between 20 degrees and 37 degrees.

[0026] Each blade 44 is also configured so that the peak point of the camber P is formed at a point corresponding to 70% of the chord length in a direction from the L.E thereof to the T.E thereof, and that the maximum quantity of the camber MC is set to 0.5% within each radius from the hub 42 to the end of the blade 44.

[0027] Further, the sweep angle α of each blade 44 has a range between 47 degrees and 49 degrees when a dimensionless radius coordinate is less than 0.3 and is linearly increased when the dimensionless radius coordinate exceeds 0.3 to have a range between 55 degrees and 57 degrees at the end of the blade.

[0028] For reference, the dimensionless radius coordinate is a factor for taking into consideration of performance of the axial flow fan only by the blades 44 except for the hub 42, and is determined between 0 and 1 when a position where the blades and the hub come into contact with each other is set to 0, and the end of each blade 44 is set to 1.

[0029] The dimensionless radius coordinate is obtained by the follow formula. $r = (R - R_h) / (R_t - R_h)$, where R is the length from the center of the axial flow fan (i.e. the center of the hub) to a certain position, R_h is the radius of the hub 42, R_t is the length from the center of the axial flow fan (i.e. the center of the hub) to the end of each blade 44, namely, the radius of the axial flow fan.

[0030] According to the axial flow fan 40 having three blades 44 in the outdoor unit of the foregoing air conditioner, as shown in Figs. 4 and 5, a pressure coefficient and constant pressure efficiency are enhanced as compared to another conventional axial flow fan having four blades. As a result, the motor for the axial flow fan having three blades can be also enhanced in operation efficiency at an operation point, and can be driven with a size smaller than that for another conventional axial flow fan having four blades. In addition, the motor for the axial flow fan having three blades is reduced by about 22% in consumption electrical power required for operation.

[0031] However, when the axial flow fan 40 is driven, a slip stream or wake component is generated at the L.E and T.E of the leading blade 44, and a turbulent flow component is generated by separation on a negative pressure surface. These two components have influence on the trailing blade 44, thus deteriorating the performance of the axial flow fan 40, and simultaneously generating the noise by a turbulent flow.

SUMMARY OF THE INVENTION

[0032] Therefore, an objective of the present invention is to design an axial flow fan within an optimal design range capable of suppressing increase in intensity of a turbulent flow generated from a surface of each blade, increase in thickness of a boundary layer on the surface of each blade and disturbance of an airflow within a region of the hub.

[0033] It is another objective to provide an axial flow fan capable of remarkably reducing a noise within the predetermined frequency range (between about 300 Hz and about 1000 Hz) with respect to the same air volume as the conventional axial flow fan.

[0034] To achieve the above objective, the present invention provides an axial flow fan according to claim 1.

[0035] Therefore, according to the present invention, the axial flow fan can reduce the noise as low as possible and increase the pressure coefficient and the constant pressure efficiency compared to the conventional axial flow fan.

BRIEF DESCRIPTION OF THE DRAWINGS

[0036] The above objects, features and advantages of the present invention will become more apparent from the following detailed description when taken in conjunction with the accompanying drawings, in which:

[0037] Fig. 1 schematically shows a general air conditioner;

[0038] Fig. 2 is a perspective view illustrating a general split type air conditioner;

[0039] Figs. 3A and 3B are front and side views of a conventional axial flow fan, respectively;

[0040] Fig. 4 is a graph showing comparison of relation between a pressure coefficient and a flow rate coefficient in a conventional axial flow fan with that of another conventional axial flow fan;

[0041] Fig. 5 is a graph showing comparison of relation between constant pressure efficiency and a flow rate coefficient in a conventional axial flow fan with that of another conventional axial flow fan;

[0042] Figs. 6A and 6B are front and side views of an axial flow fan according to the present invention, respectively;

[0043] Figs. 7A and 7B show a state where blades are tilted on an outer circumferential surface of a hub at a certain rake angle in axial flow fans according to the prior art and the present invention;

[0044] Fig. 8 is a graph showing a state where a noise is changed according to a change of a solidity with respect to axial flow fans of the prior art and the present invention;

[0045] Fig. 9 is a graph showing a state where a noise is changed according to a change of a quantity of a camber with respect to axial flow fans of the prior art and the present invention;

[0046] Fig. 10 is a graph showing relation between a (constant) pressure coefficient, a constant pressure efficiency and a flow rate coefficient with respect to axial flow fans of the prior art and the present invention; and

[0047] Fig. 11 is a graph showing comparison of a state where a noise is changed according to a change of a frequency of an axial flow fan of the present invention with that of an axial flow fan of the prior art.

DETAILED DESCRIPTION OF PREFERRED EMBODIMENTS

[0048] An exemplary embodiment of the present invention will now be described with reference to the accompanying drawings.

[0049] Figs. 6A and 6B are front and side views of an axial flow fan according to the present invention, respectively. Figs. 7A and 7B show a state where blades are tilted on an outer circumferential surface of a hub at a certain rake angle in axial flow fans according to the prior art and the present invention.

[0050] An axial flow fan 140 of an outdoor unit of an air conditioner according to the present invention is composed of a hub 142 connected with a rotational shaft 141 of a motor, and a plurality of blades 144 integrally formed on an outer circumferential surface of the hub 142.

[0051] The axial flow fan 140 is configured so that the number of the blades is two, that a ratio of an inner diameter to an outer diameter (*i.e.* a ratio of the outer diameter of the hub and the outer diameter of the fan) is between about 0.35 and about 0.4, that a solidity, a ratio of the whole area of the fan 140 and an area of the blades, has a range of 0.85 ± 0.05 , and that a quantity of a camber of the hub 142 has a range of $5.0\% \pm 1.0\%$.

[0052] Hereinafter, a detailed description will be made on the axial flow fan of the outdoor unit of the air conditioner according to the present invention.

[0053] Meanwhile, when the axial flow fan 140 is driven, a slip stream or wake component may be generated at a leading edge (L.E) and a trailing edge (T.E) of the leading blade 144, and a turbulent flow component may be generated by separation on a negative pressure surface. These two components may have influence on the trailing blade 144, thus deteriorating performance of the axial flow fan 140, and simultaneously generating a noise by a turbulent flow. Thus, the present invention aims at preventing the drawbacks of the axial flow fan 140.

[0054] Further, the present invention is to suppress increase in intensity of the turbulent flow generated from a surface of each blade 144, increase in thickness of a boundary layer on the surface of each blade 144, and disturbance of an airflow within a region of the hub 142.

[0055] In order to accomplish the objectives, the axial flow fan 140 is formed so that the number of the blades 144 is two, that the ratio of the inner diameter to the outer diameter (*i.e.* the ratio of the outer diameter of the hub and the outer diameter of the axial flow fan) is between about 0.35 and about 0.4, that the solidity, the ratio of the whole area of the fan 140 and the area of the blades, has the range of 0.85 ± 0.05 , and that the camber quantity of the hub 142 has the range of $5.0\% \pm 1.0\%$. With regard to this, the detailed configuration of the present invention is as follows.

[0056] The axial flow fan 140 of the outdoor unit of the air conditioner according to the present invention, as shown in Fig. 6A, is composed of the hub 142 connected with the rotational shaft 141 of the motor, and the plurality

of blades 144 integrally formed on the outer circumferential surface of the hub 142.

[0057] Here, the number of the blades 144 is set to two. The inner and outer diameter ratio of the axial flow fan 140, i.e. the ratio of the outer diameter of the hub 142 and the outer diameter of the axial flow fan 140, is set to a range between about 0.35 and about 0.40.

[0058] Further, the ratio of the whole area of the axial flow fan 140 and the area of the blades, i.e. the solidity, has the range of 0.85 ± 0.05 , and the camber quantity of the hub 142 has the range of $5.0\% \pm 1.0\%$. The solidity can be expressed by the following formula.

[0059] $Solidity = (chord \times Z) / 2\pi r$

[0060] where $2\pi r$: circumference length when a radius is r , $chord$: straight line joining the L.E of the blade with the T.E of the blade, Z : the number of blades.

[0061] Thus, a value of the solidity presented in the present invention may become a mean value from the hub and a tip, for example, an integral value.

[0062] For the axial flow fan 140, as shown in Figs. 7A and 7B, a rake base line of each blade 144 formed on the outer circumferential surface of the hub 142 is tilted from that formed horizontal to the outer circumferential surface of the conventional hub 42 by a rake angle between about 20 degrees and about 23 degrees. Here, the rake angle refers to an angle determining how much to tilt and form the blades 144 on the circumferential surface of the hub 142.

[0063] As for a state where the blades 144 are formed on the outer circumferential surface of the hub 142 through the rake angle, as shown in Figs. 7A and 7B, among the whole length from the outer circumferential surface of the hub 142 to the outer end (i.e. tip) of each blade 144, a part from the outer circumferential surface of the hub 142 to a predetermined portion of each blade 144 is tilted at the rake angle, and the other part from the predetermined portion of each blade 144 and the tip of each blade 144 is provided with a bulge 146 protruded toward a pressure surface. The tip of each blade 144 has the same angle as the rake angle from the outer circumferential surface of the hub 142 to the predetermined portion of each blade 144. In this manner, a profile of the axial flow fan 140 is formed as a whole.

[0064] In other words, when the section from the outer circumferential surface of the hub to the tip of each blade is divided into two sections, the first section performs rotational displacement at the identical angle, and the second section forms a non-linear angle raised toward the pressure surface. The tip (i.e. a section except for the two sections) is adapted to apply an identical value of the first section.

[0065] At this point, the outer diameter D of the axial flow fan is $460 \pm 2\text{mm}$, and the outer diameter d of the hub 142 is $170 \pm 2\text{mm}$.

[0066] Here, a pitch angle, a peak point of a camber, and a sweep angle of each blade 144 are the same as the pitch angle β , the peak point of the camber P , the maximum quantity of the camber MC , and the sweep

angle α of the conventional blade 44 shown in Figs. 3A and 3B. Now, the pitch angle, the peak point of the camber, and the sweep angle of each blade 144 will be described in detail below.

[0067] The pitch angle β of each blade 144 is configured to be linearly changed from the hub 142 to the end of the blade 144 within a range between 37 degrees and 20 degrees.

[0068] Each blade 144 is configured so that the peak point of the camber P is formed at a position corresponding to 70% of a length of a chord in a direction from the front end of the blade to the rear end of the blade, and that the maximum quantity of the camber MC is kept constant at a value of 0.5% within each radius from the hub 142 to the end of the blade 144.

[0069] Furthermore, the sweep angle α of each blade 144 has a range between about 47 degrees and about 49 degrees when a dimensionless radius coordinate is less than 0.3 and is linearly increased when the dimensionless radius coordinate exceeds 0.3 to have a range between about 55 degrees and about 57 degrees at the end of the blade.

[0070] A change of the noise generated from the axial flow fan configured as set forth above will be described below.

[0071] Fig. 8 is a graph showing a state where a noise is changed according to a change of a solidity with respect to axial flow fans of the prior art and the present invention. Fig. 9 is a graph showing a state where a noise is changed according to a change of a quantity of a camber with respect to axial flow fans of the prior art and the present invention. Fig. 10 is graph showing relation between a (constant) pressure coefficient, a constant pressure efficiency and a flow rate coefficient with respect to axial flow fans of the prior art and the present invention. Fig. 11 is a graph showing comparison of a state where a noise is changed according to a change of a frequency of an axial flow fan of the present invention with that of an axial flow fan of the prior art.

[0072] As seen from the foregoing description and the drawings, the solidity applied to the present invention has a range of 0.85 ± 0.05 and the camber quantity of the hub has a range of $5.0\% \pm 1.0\%$.

[0073] In contrast, the solidity applied to the prior art ($Z=3$) has a relatively great value compared to that of the present invention, and the camber quantity of the hub has a relatively small value.

[0074] The following description will be made with reference to Figs. 10 and 11.

[0075] In the graph of Fig. 10, an upper line shows a comparison of relation of the (constant) pressure coefficient and the flow rate coefficient in the axial flow fan 140 with that of the conventional axial flow fan 40, while a lower line shows a comparison of relation of the constant pressure efficiency and the flow rate coefficient in the axial flow fan 140 with that of the conventional axial flow fan 40.

[0076] For the axial flow fan 140 according to the

present invention, the noise change was measured depending on the change of the solidity as the ratio of the whole area of the fan 140 to the area of the blades. It was found that as a result of the measurement, as shown in Fig. 8, when the ratio of the whole area of the fan 140 to the area of the blades, i.e. the solidity, was about 0.87, the noise was the lowest. Further, the noise change was measured depending on the change of the camber quantity of each blade of the axial flow fan 140. It was found that as a result of the measurement, as shown in Fig. 9, when the camber quantity of the blade 144 was about 0.5%, the noise was lowest.

[0077] For the axial flow fan 140 according to the present invention, it can be seen that as shown in Fig. 10, the pressure coefficient and the constant pressure efficiency were enhanced over the conventional axial flow fan 40, and that the operation efficiency was also enhanced at the operation point according to the enhancement of the pressure coefficient and the constant pressure efficiency of the axial flow fan 140 as set forth above.

[0078] Further, Fig. 11 is a graph showing comparison of a state where a noise is changed according to a change of a frequency of an axial flow fan of the present invention with that of an axial flow fan of the prior art. As shown in Fig. 11, it can be seen that when having an air volume equal to that of the conventional axial flow fan 40, the axial flow fan 140 was subjected to great reduction of the noise in a range between about 300 Hz and about 1000Hz.

[0079] As set forth above, the present invention relates to the axial flow fan configured so that the number of the blades is two, that a predetermined rake angle is kept constant in the part from the hub to the predetermined portion of the blade among the whole part from the hub to the outer end of the blade and is increased in the pressure surface direction in the other part from the predetermined portion of the blade to the outer end of the blade, and the ratio of the inner diameter to the outer diameter is between about 0.35 and about 0.4.

[0080] Therefore, the axial flow fan of the present invention is designed within an optimal design range (that the solidity, the ratio of the whole area of the axial flow fan and the area of the blades, is about 0.87 and that the camber quantity of the hub is about 5.0 %), for example, capable of suppressing increase in intensity of the turbulent flow generated from the surface of each blade, increase in thickness of the boundary layer on the surface of each blade and disturbance of the airflow within the region of the hub. As a result, the axial flow fan of the present invention can reduce the noise as low as possible and increase the pressure coefficient and the constant pressure efficiency compared to the conventional axial flow fan.

[0081] Further, the axial flow fan of the present invention can remarkably reduce the noise within the predetermined frequency range (e.g. between about 300 Hz and about 1000 Hz) with respect to the same air volume

as the conventional axial flow fan.

[0082] While the invention has been shown and described with reference to certain preferred embodiments thereof, it will be understood by those skilled in the art that various changes in form and details may be made therein.

[0083] For example, the axial flow fan of the present invention may be applied to a refrigerator or other apparatuses for condensing and evaporating a refrigerant.

Claims

1. An axial flow fan (140) comprising:

a hub (142) connected with a rotational shaft (141) of a motor; and
at least one blade (144) contacting the hub (142),

wherein the blade (144) has a part from the hub (142) to a predetermined portion of the blade (144) among a whole part from the hub (142) to an outer end of the blade (144), and the other part from the predetermined portion of the blade (144) to the outer end of the blade (144), the part being equally applied at a predetermined rake angle, and the other part being raised in a direction of a pressure surface of the blade (144),

characterised in that

the outer edge has the same angle as the rake angle;

a rake base line of the blade (144) is formed on an outer circumferential surface of the hub (142) and within the part from the hub (142) to the predetermined portion, is tilted by the rake angle between about 20 degrees and about 23 degrees;

and **in that** a ratio of an inner diameter and an outer diameter of the axial flow fan (140) which is the ratio of the outer diameter of the hub (142) and the outer diameter of the axial flow fan (140), is between 0.35 and about 0.4.

2. An axial flow fan (140) as set forth in claim 1, wherein the number of the blades is two.

3. An axial flow fan (140) as set forth in claim 1, wherein a solidity has a range of 0.85 ± 0.05 .

4. An axial flow fan (140) as set forth in claim 1, wherein the outer diameter of the axial flow fan (140) has a range of $460 \pm 2\text{mm}$.

5. An axial flow fan (140) as set forth in claim 1, wherein the hub (142) has a diameter of a range of $170 \pm 2\text{mm}$.

Patentansprüche

1. Axiallüfter (140), umfassend:

eine Nabe (142), die mit einer Rotationswelle (141) eines Motors verbunden ist; und
 zumindest ein Blatt (144), das die Nabe (142) kontaktiert,
 wobei das Blatt (144) einen Teil von der Nabe (142) zu einem Vorbestimmten Abschnitt des Blatts (144), als Teil eines Gesamtteils von der Nabe (142) zu einem äußeren Ende des Blatts (144), und den anderen Teil von dem Vorbestimmten Abschnitt des Blatts (144) zu dem äußeren Ende des Blatts (144) aufweist, wobei der Teil gleichmäßig in einem Vorbestimmten Einsteckwinkel aufgebracht ist und der andere Teil in einer Richtung einer Druckoberfläche des Blatts (144) angehoben ist, **dadurch gekennzeichnet, dass**
 die äußere Kante denselben Winkel wie den Einsteckwinkel aufweist;
 eine Einsteckgrundlinie des Blatts (144) an einer äußeren Umfangsoberfläche der Nabe (142) und innerhalb des Teils von der Nabe (142) zu dem Vorbestimmten Abschnitt ausgebildet ist, um den Einsteckwinkel zwischen etwa 20 Grad und etwa 23 Grad gekippt ist;
 und dass ein Verhältnis eines Innendurchmessers und eines Außendurchmessers des Axiallüfters (140), welches das Verhältnis des Außendurchmessers der Nabe (142) und des Außendurchmessers des Axiallüfters (140) ist, zwischen 0,35 und etwa 0,4 liegt.

2. Axiallüfter (140) nach Anspruch 1, wobei die Anzahl der Blätter zwei ist.
3. Axiallüfter (140) nach Anspruch 1, wobei ein Bedeckungsgrad einen Bereich von $0,85 \pm 0,05$ aufweist.
4. Axiallüfter (140) nach Anspruch 1, wobei der Außendurchmesser des Axiallüfters (140) in einem Bereich von 460 ± 2 mm liegt.
5. Axiallüfter (140) nach Anspruch 1, wobei die Nabe (142) einen Durchmesser in einem Bereich von 170 ± 2 mm hat.

Revendications

1. Ventilateur axial (140) comprenant :

un moyeu (142) raccordé avec un arbre de rotation (141) d'un moteur ; et
 au moins une pale (144) en contact avec le moyeu (142),

dans lequel la pale (144) a une partie allant du moyeu (142) jusqu'à une partie prédéterminée de la pale (144) parmi la partie entière allant du moyeu (142) jusqu'à une extrémité externe de la pale (144) et l'autre partie allant de la partie prédéterminée de la pale (144) jusqu'à l'extrémité externe de la pale (144), la partie étant appliquée de manière égale selon un angle d'inclinaison prédéterminé, et l'autre partie étant arrondie dans une direction d'une surface de pression de la pale (144),

caractérisé en ce que :

le bord externe a le même angle que l'angle d'inclinaison ;
 une ligne de base d'inclinaison de la pale (144) est formée sur une surface circonférentielle externe du moyeu (142) et dont la partie allant du moyeu (142) jusqu'à la partie prédéterminée, est inclinée selon l'angle d'inclinaison compris entre environ 20 degrés et environ 23 degrés ;
 et **en ce qu'un** rapport d'un diamètre interne et d'un diamètre externe du ventilateur axial (140) qui est le rapport du diamètre externe du moyeu (142) et du diamètre externe du ventilateur axial (140), est compris entre 0,35 et environ 0,4.

2. Ventilateur axial (140) selon la revendication 1, dans lequel le nombre de pales est de deux.
3. Ventilateur axial (140) selon la revendication 1, dans lequel une solidité est de l'ordre de $0,85 \pm 0,05$.
4. Ventilateur axial (140) selon la revendication 1, dans lequel le diamètre externe du ventilateur axial (140) est de l'ordre de 460 ± 2 mm.
5. Ventilateur axial (140) selon la revendication 1, dans lequel le moyeu (142) a un diamètre de l'ordre de 170 ± 2 mm.

Fig.1
Related Art

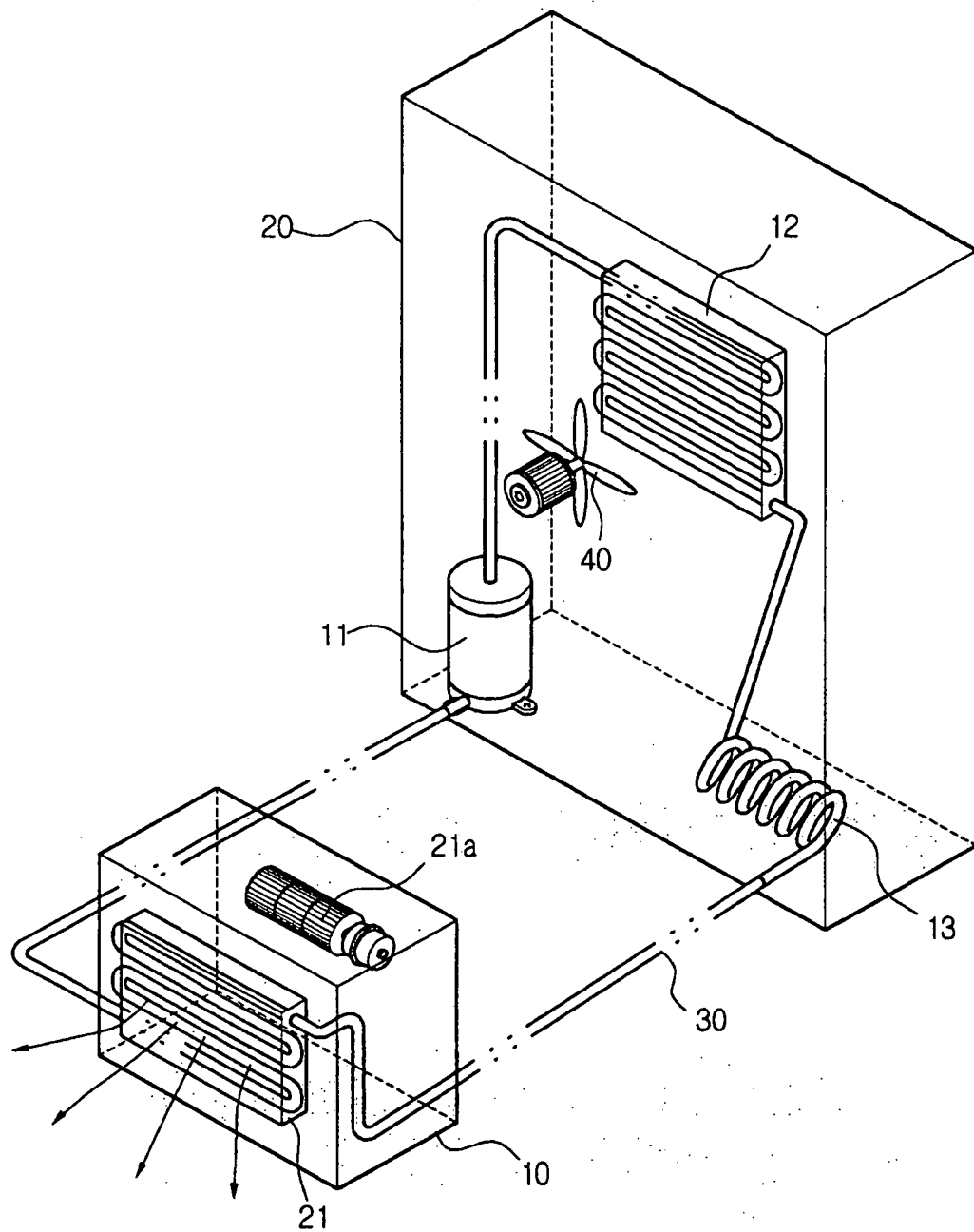


Fig.2
Related Art

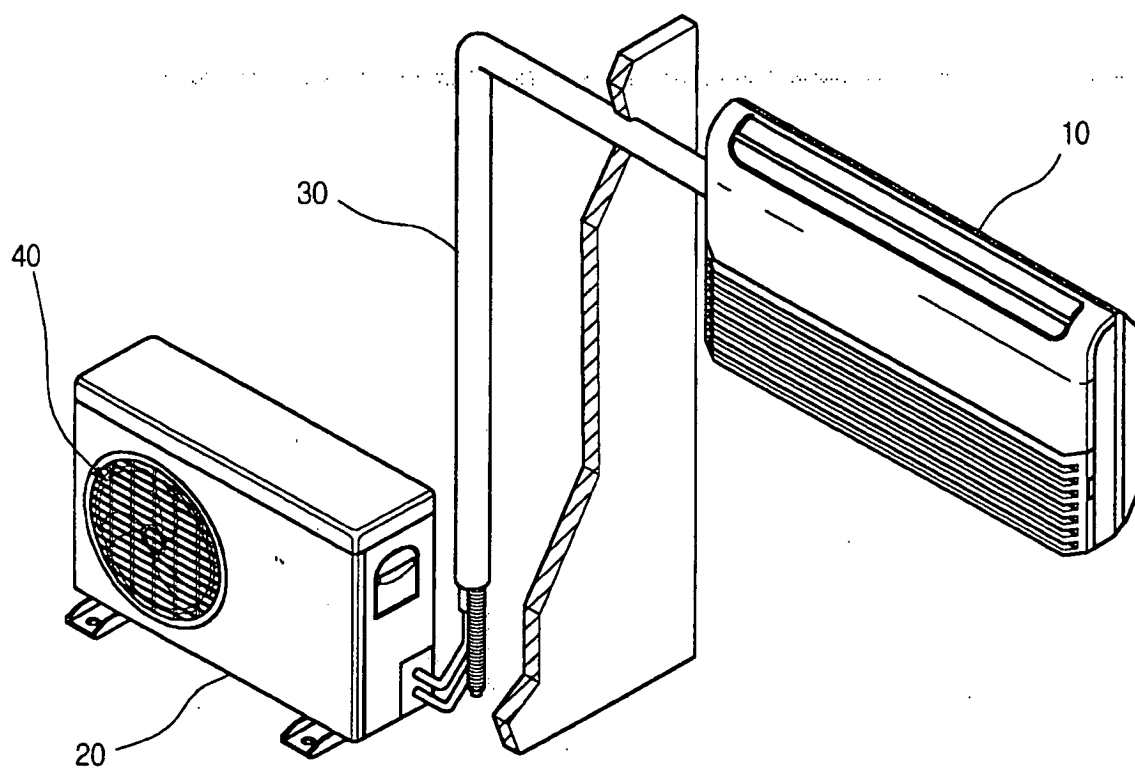


Fig.3a
Related Art

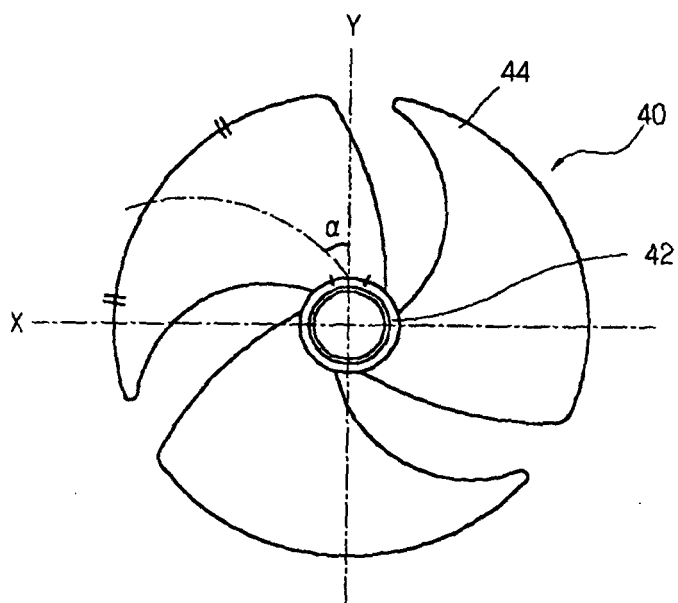


Fig.3b
Related Art

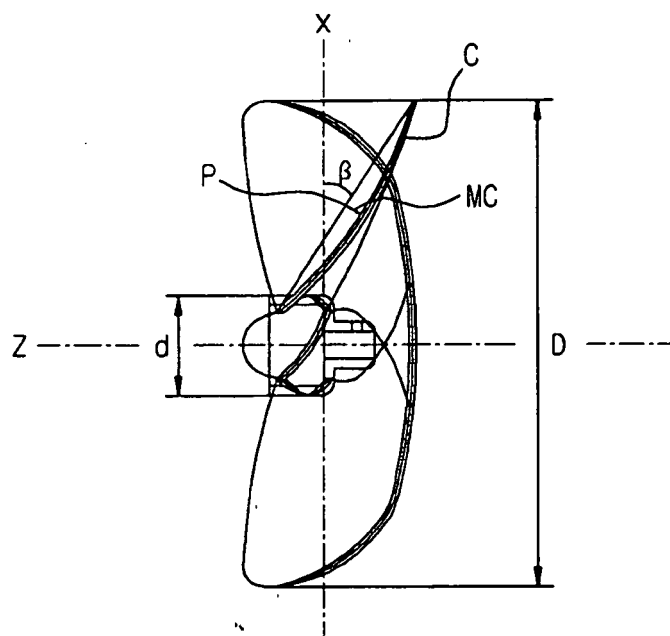


Fig.4
Related Art

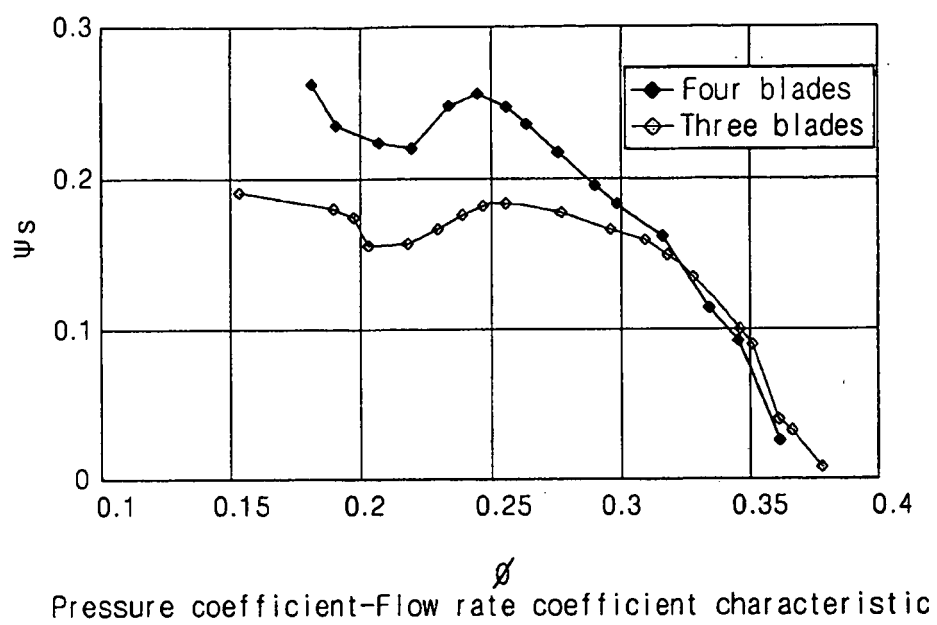
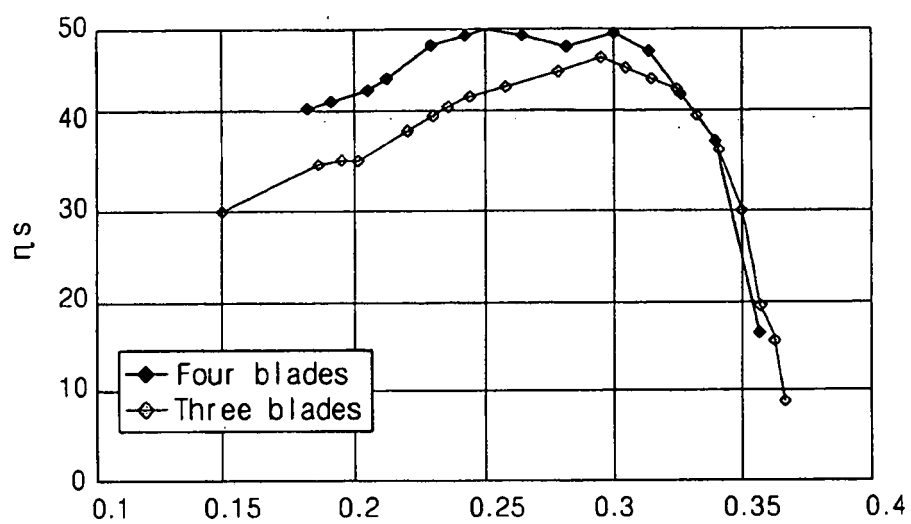


Fig.5
Related Art



Constant pressure efficiency-Flow rate coefficient characteristic

Fig.6a

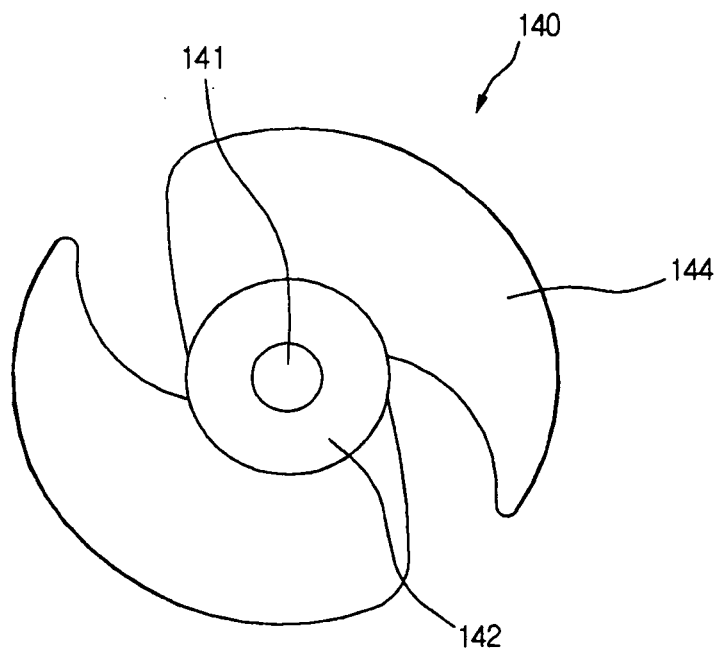


Fig.6b

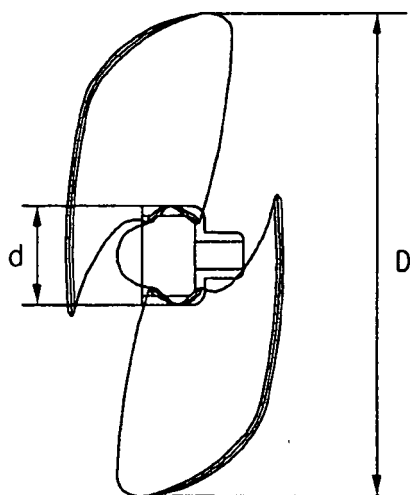


Fig.7a

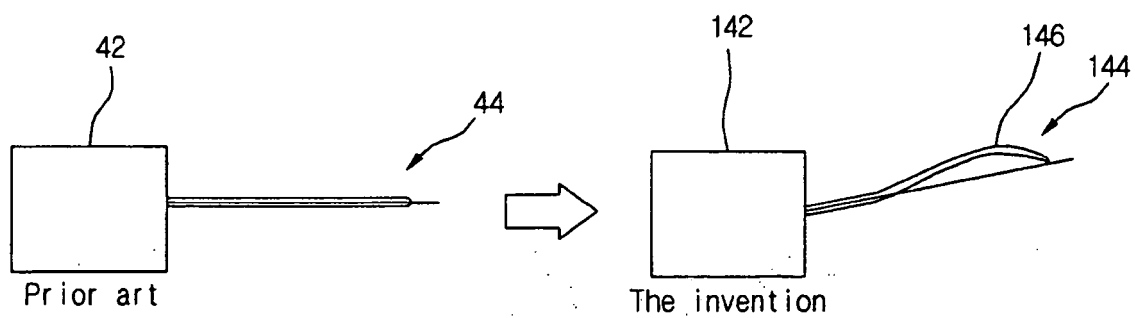
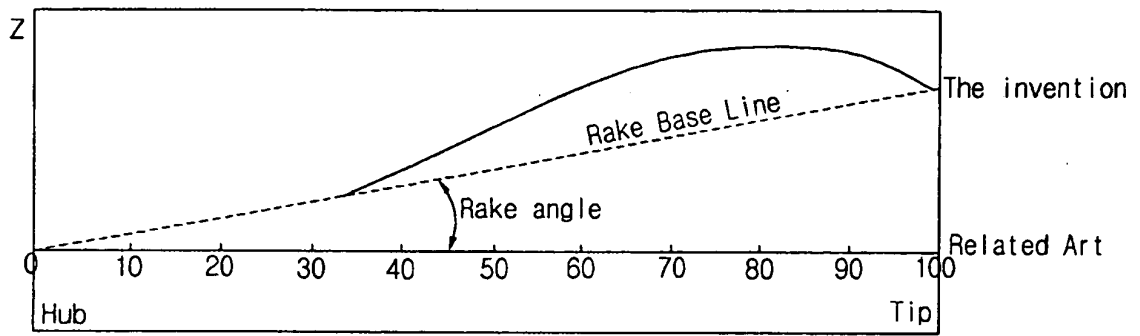


Fig.7b



Radial rake distribution

Fig.8

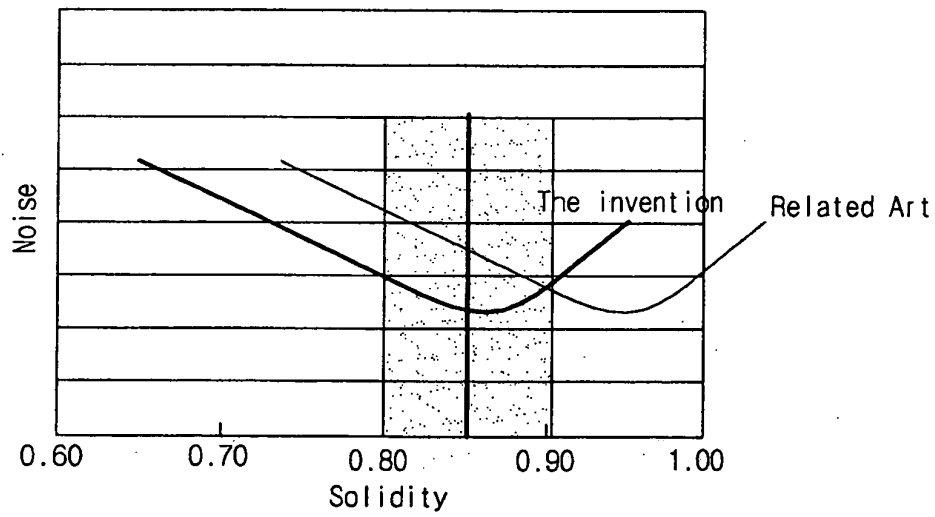


Fig.9

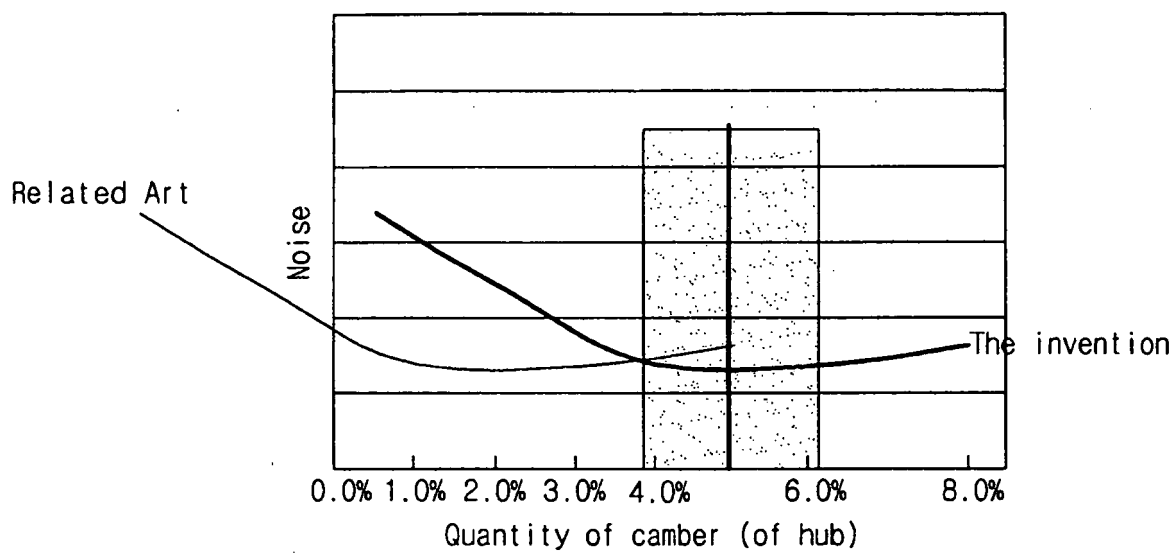


Fig. 10

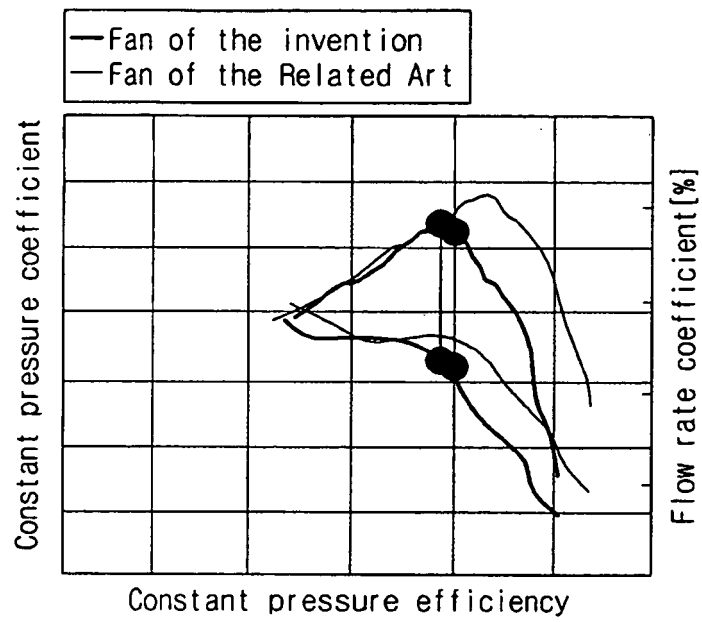
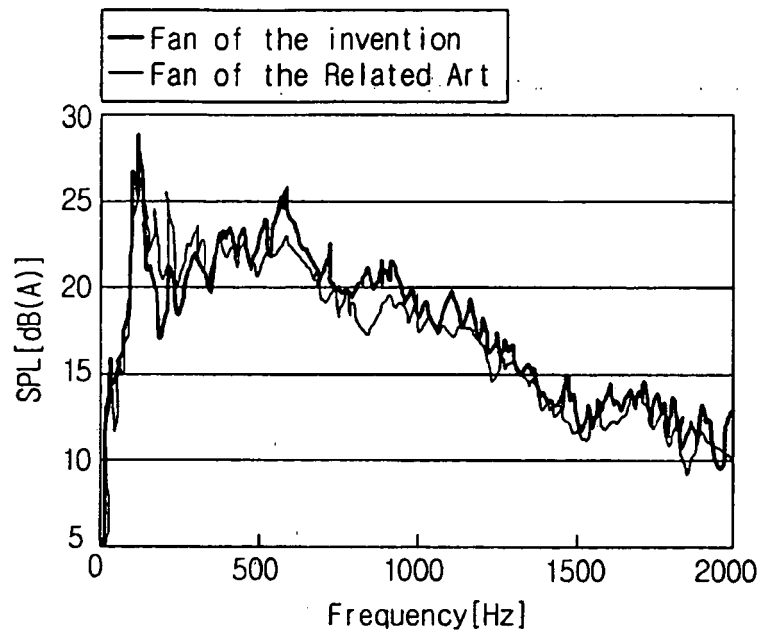


Fig.11



REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- US 6447251 B1 [0001]
- KR 200314960 [0024]