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(11) **EP 1 070 849 A2**

(12) **EUROPEAN PATENT APPLICATION**

(43) Date of publication:
24.01.2001 Bulletin 2001/04

(51) Int. Cl.⁷: **F04D 29/32**, F04D 29/38,
F04D 29/66

(21) Application number: **00114849.3**

(22) Date of filing: **11.07.2000**

(84) Designated Contracting States:
**AT BE CH CY DE DK ES FI FR GB GR IE IT LI LU
MC NL PT SE**
Designated Extension States:
AL LT LV MK RO SI

(30) Priority: **22.07.1999 KR 9929802**
22.07.1999 KR 9929803

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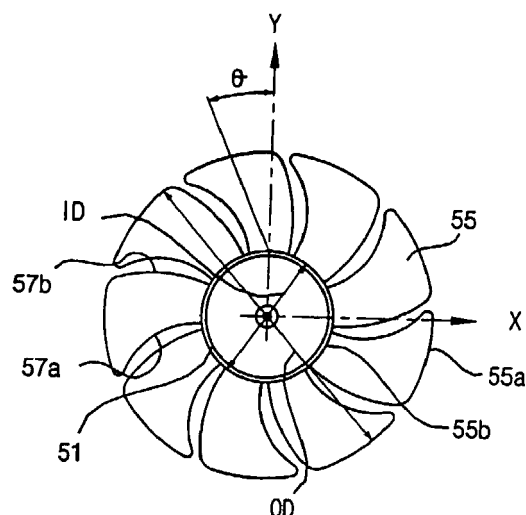
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(54) **Axial flow fan**

(57) An axial flow fan for refrigerators is disclosed. In the axial flow fan of this invention, a hub(51) is mounted to the rotating shaft of a drive motor, with a plurality of blades(55) regularly fixed around the hub(51). A variety of designing factors of the blades, such as the number of blades, hub ratio, sweep angle, pitch angle and maximum camber ratio, are optimally designed to allow a smooth flow of cool air agreeable with both the large pressure loss and complex flow passage of refrigerators while accomplishing a reduction in air flow noise of the refrigerators.

A vortex prevention means is provided on the concave back surface of the annular rim formed of the shroud for preventing discharged air from forming a vortex stream in back of the fan.

FIG.7a



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Description

BACKGROUND OF THE INVENTION

Field of the Invention

[0001] The present invention relates, in general, to an axial flow fan for refrigerators, used for feeding cool air from an evaporator into both a freezer compartment and a fresh compartment in refrigerators, and, more particularly, to an axial flow fan for refrigerators optimally designed in a variety of designing factors, such as the number of blades, hub ratio, sweep angle, pitch angle and maximum camber ratio, thus accomplishing a reduction in operational noise of the refrigerators and in vortex formed around the fan and thereby finally reducing its flow resistance.

Description of the Prior Art

[0002] Fig. 1 is a perspective view of a conventional axial flow fan for refrigerators. Fig. 2 is a sectional view of the blade tip of the conventional axial flow fan.

[0003] The construction and operation of such a conventional axial flow fan will be described herein below in conjunction with Figs. 1 and 2.

[0004] As shown in the drawings, the conventional axial flow fan comprises a hub 1, which is firmly mounted to the rotating shaft of the drive motor, with a plurality of blades 5 regularly fixed around the hub 1.

[0005] In the conventional axial flow fan, the number of the blades 5 is typically set to three to five, with a hub ratio of the hub diameter to the outer diameter of the fan being set to 0.25 ~ 0.3 and a pitch angle of each blade 5 ranging from 25° to 35°.

[0006] In such an axial flow fan, the pitch angle is formed between the radial straight line of each blade 5 and another straight line extending from the blade leading edge to the blade trailing edge. This pitch angle is determined by an inclination of each blade 5 relative to a plane perpendicular to the rotating axis of the fan.

[0007] However, such a conventional axial flow fan for refrigerators has been designed while considering only some designing factors, such as the number of blades 5, hub ratio and pitch angle, and so it is almost impossible for the axial flow fan to form appropriate flow agreeable with the large pressure loss and the complex flow passage required in large-sized modern refrigerators. This finally increases operational noise of such large-sized refrigerators.

[0008] In addition, during operation of such a conventional axial flow fan, each trailing blade 5 undesirably confronts the vortex stream formed by a leading blade 5, thus generating fluid noise. In the art of this invention, such a phenomenon is so-called "blade vortex interaction (BVI)".

[0009] In the recent designing process for axial flow fans, a technique for accomplishing a desired reduction

in BVI has been actively studied.

[0010] As shown in Fig. 2, the blade tip 5a of an axial flow fan forms a smoothly curved cross-section consisting of a pressure surface 5b and a negative pressure surface 5c. In operation of the fan, air pressure caused by an air current acts on the pressure surface 5b, while negative pressure acts on the negative pressure surface 5c opposite to the surface 5b. Due to such a smoothly curved cross-section of the blade tip 5a, static pressure of the air current flowing from the pressure surface 5b to the negative pressure surface 5c is restored abruptly and quickly.

[0011] Therefore, the blade passing frequency (BPF), which is the main frequency of fluid noise caused by a collision of the air current against the blades 5 during operation of the fan and is calculated by a plus integral times of the result of multiplication of the number of blades 5 by rpm of the fan, is reduced to a low level.

[0012] However, the conventional doors of a refrigerator, used for intercepting noise leaking from the compressor and fan of the machine room and from the axial flow fan used for accomplishing a circulation of cool air within the refrigerator into the outside of the cabinet of the refrigerator through a variety of passages, have been typically designed to intercept high frequency noise of not lower than 700Hz. Therefore, it is almost impossible for such conventional doors to intercept such a low blade passing frequency (BPF) generated from the conventional axial flow fan.

[0013] In other words, the conventional axial flow fan typically generates operational noise having a large low frequency band and a low BPF, and so the conventional doors of refrigerators fail to accomplish a desired noise intercepting effect in the case of operational noise of the conventional axial flow fan, but regrettably allow the noise to leak from the axial flow fan to the outside of the cabinet of the refrigerator. Such operational noise disturbs those around the refrigerator.

[0014] Fig. 3 is a front view, showing a conventional shroud installed around the axial flow fan for refrigerators. Fig. 4 is a sectional view, showing the construction of the conventional shroud for axial flow fans.

[0015] As shown in the drawings, the conventional shroud 7 is installed around the blades 5 of the fan, with an annular rim 9 being formed closely around the blades 5 while being bulged to the front of the fan in an air inlet direction.

[0016] Such a conventional shroud 7 is installed around the blade tips 5a of the fan while leaving a pre-determined annular gap between the tips 5a and the inside edge of the shroud 7. The above shroud 7 guides the cool air current when the air current flows in an axial direction of the fan during operation.

[0017] On the other hand, the annular rim 9, formed closely around the blades 5 while being bulged to the front of the fan in the air inlet direction, induces a smooth airflow during operation of the fan. That is, the cool air current flows over the bulged annular rim 9, thus

smoothly flowing on the shroud 7 without forming an undesirable flow resistance.

[0018] However, the conventional shroud 7 has the following problem. That is, since the annular rim 9 is formed closely around the blades 5 while being bulged to the front of the fan in the air inlet direction as best seen in Fig. 5, the rear surface of the rim 9 in the air outlet side of the fan is concaved, and so cool air discharged from the fan comes into undesirable collision against the concave surface of the rim 9 and forms an intensive and large-scaled vortex, thus finally increasing the flow loss.

[0019] That is, the annular rim 9, positioned around the blades of the axial flow fan, has so large a diameter that the inlet air is partially brought into collision against the concave surface of the rim 9 while undesirably forming an intensive and large-scaled vortex around the concave surface, thus resulting in a substantial flow loss during operation of the fan.

SUMMARY OF THE INVENTION

[0020] Accordingly, the present invention has been made keeping in mind the above problems occurring in the prior art, and an object of the present invention is to provide an axial flow fan for refrigerators, which is optimally designed in a variety of designing factors, such as the number of blades, hub ratio, sweep angle, pitch angle and maximum camber ratio, thus accomplishing a desired reduction in operational noise of the refrigerators and in vortex formed around the fan and thereby finally reducing its flow resistance.

[0021] Another object of the present invention is to provide an axial flow fan for refrigerators, which is improved in the structure of its shroud so as to be free from a formation of a vortex in the discharged cool air current in the air outlet side of the fan, thus minimizing its flow loss.

[0022] In order to accomplish the above object, the primary embodiment of the present invention provides an axial flow fan for refrigerators, comprising a hub mounted to the rotating shaft of a drive motor, with a plurality of blades regularly fixed around the hub, wherein a variety of designing factors of the blades, such as the number of blades, hub ratio, sweep angle, pitch angle and maximum camber ratio, are optimally designed to allow a smooth flow of cool air agreeable with both the large pressure loss and complex flow passage of refrigerators while accomplishing a reduction in air flow noise of the refrigerators.

[0023] Another embodiment of this invention provides an axial flow fan for refrigerators, comprising a hub mounted to a rotating shaft of a motor, with a plurality of blades regularly fixed around the hub and a shroud installed around the blades to guide an air current, wherein the shroud consists of an annular rim formed closely around the blades while being bulged to the front of the fan in an air inlet direction, and a vortex pre-

vention means provided on a concave back surface of the annular rim for preventing discharged air from forming a vortex stream in back of the fan.

BRIEF DESCRIPTION OF THE DRAWINGS

[0024] The above and other objects, features and other advantages of the present invention will be more clearly understood from the following detailed description taken in conjunction with the accompanying drawings, in which:

Fig. 1 is a perspective view of a conventional axial flow fan for refrigerators;

Fig. 2 is a sectional view of the blade tip of the conventional axial flow fan;

Fig. 3 is a front view, showing a conventional shroud installed around the axial flow fan for refrigerators;

Fig. 4 is a sectional view taken along the line A-A of Fig. 3, showing the construction of the conventional shroud for axial flow fans;

Fig. 5 is a view, showing an air current formed around the conventional shroud;

Fig. 6 is a perspective view of an axial flow fan for refrigerators in accordance with the preferred embodiment of the present invention;

Figs. 7a and 7b are front and side views of the axial flow fan according to the preferred embodiment of this invention;

Figs. 8a and 8b are sectional views, showing the shape of a blade included in the axial flow fan according to the preferred embodiment of this invention;

Fig. 9 is a sectional view of the blade tip of the axial flow fan according to this invention;

Fig. 10 is a graph showing operational noise of the axial flow fan according to the invention as a function of the hub ratio of the axial flow fan;

Fig. 11 is a graph showing operational noise of the axial flow fan according to the invention as a function of the sweep angle of the axial flow fan;

Fig. 12 is a graph showing operational noise of the axial flow fan according to the invention as a function of the pitch angle of the axial flow fan;

Fig. 13 is a graph showing operational noise of the axial flow fan according to the invention as a function of the maximum camber position of the axial flow fan;

Fig. 14 is a graph showing operational noise of the axial flow fan according to the invention as a function of the rake angle of the axial flow fan;

Fig. 15 is a graph, comparatively showing the operational noise characteristics of the front sections of refrigerators using a conventional axial flow fan and the axial flow fan of this invention;

Fig. 16 is a front view, showing a shroud installed around the axial flow fan for refrigerators in accord-

ance with an embodiment of the present invention;

Fig. 17 is a sectional view taken along the line B-B of Fig. 16, showing the construction of the shroud for axial flow fans according to the embodiment of this invention;

Fig. 18 is a view, showing an air current formed around the shroud of Fig. 16;

Fig. 19 is a sectional view, showing the construction of a shroud in accordance with another embodiment of this invention; and

Fig. 20 is a graph showing operational noise of the axial flow fan with the shroud according to the invention as a function of air volume.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

[0025] Fig. 6 is a perspective view of an axial flow fan for refrigerators in accordance with the preferred embodiment of the present invention. Figs. 7a and 7b are front and side views of the axial flow fan according to the preferred embodiment of this invention. Figs. 8a and 8b are sectional views, showing the shape of a blade included in the axial flow fan according to the preferred embodiment of this invention. Fig. 9 is a sectional view of the blade tip of the axial flow fan according to this invention.

[0026] The construction and operation of the axial flow fan for refrigerators according to the preferred embodiment of this invention will be described herein below in conjunction with Figs. 6 to 9.

[0027] As shown in the drawings, the axial flow fan of this invention comprises a hub 51, which is firmly mounted to the rotating shaft of the drive motor, with a plurality of blades 55 regularly fixed around the hub 51.

[0028] In the axial flow fan of this invention, the number of the blades 55 is preferably set to at least seven. When considering a variety of factors, such as desired operational efficiency, desired air volume and desired air pressure of the fan, it is most preferable to set the number of the blades 55 to nine.

[0029] In the axial flow fan of this invention, the hub ratio of the hub diameter ID to the outer diameter OD of the fan is set to $0.45 \sim 0.55$, with the diameter ID of the hub 51 being set to $55 \text{ mm} \pm 5 \text{ mm}$ and the outer diameter OD of the fan being set to $110 \text{ mm} \pm 10 \text{ mm}$.

[0030] In addition, the sweep angle θ of each blade 55 ranges from 30° to 34° .

[0031] The above sweep angle θ of each blade 55 is an angle formed between a straight line extending between the center of the blade hub 55b and the center of the blade tip 55a and another straight line extending between the center of the blade hub 55b and the center of the hub 51. This sweep angle θ of each blade 55 expresses the tilt of the blade 55 in the rotating direction of the blades 55.

[0032] In the axial flow fan of this invention, the pitch angle Ψ of each blade 55 is $32^\circ \pm 2^\circ$ at the blade

tip 55a and $45^\circ \pm 2^\circ$ at the blade hub 55b.

[0033] The above pitch angle Ψ of each blade 55 is an angle formed between a straight line extending between the blade leading edge 57a to the blade trailing edge 57b and an X-axis perpendicular to a Z-axis that is the rotating axis of the fan. This pitch angle Ψ of each blade 55 expresses the slope of the blade 55 relative to a plane perpendicular to the Z-axis.

[0034] On the other hand, the maximum camber position of each blade 55 is set to 0.65, with the camber positions being uniformly distributed on each blade 55 from the blade hub 55b to the blade tip 55a. In addition, the maximum camber ratio of each blade 55 is 11.5% at the blade tip 55a and 8% at the blade hub 55b.

[0035] In such a case, the maximum camber position of each blade 55 is indicated as a ratio(CP/CX) of the distance CP from the blade leading edge 57a to a point being spaced furthest from the blade 55 on a cord CL that is a straight line extending from the blade leading edge 57a to the blade trailing edge 57b to the length CX of the cord CL. The distance between said straight line and said position on the blade 55 is the maximum camber C. The maximum camber ratio is a ratio of the maximum camber C to the cord length CX.

[0036] In addition, the rake angle of each blade 55 is zero. This rake angle expresses the slope of the blade 55 relative to a positive axial direction.

[0037] As described above, when designing each blade of the axial flow fan to have a large sweep angle θ , a large pitch angle Ψ , and a large maximum camber ratio, it is possible to desirably reduce fluid noise generated by the fan during operation.

[0038] In addition, the blade passing frequency (BPF), which is the main frequency of fluid noise caused by a collision of the air current against the blades 55 during operation of the fan and is calculated by a plus integral times of the result of multiplication of the number of blades 55 by rpm of the fan, is increased to a high level. Therefore, the doors of a refrigerator typically designed to intercept high frequency noise effectively intercept such a BPF. It is thus possible to desirably reduce operational noise of refrigerators.

[0039] As shown in Fig. 9, the blade tip 55a of each blade 55 of the axial flow fan forms a curved cross-section consisting of a pressure surface 56b and a negative pressure surface 56a. In operation of the fan, air pressure caused by an air current acts on the pressure surface 56b, while negative pressure acts on the negative pressure surface 56a opposite to the surface 56b. In the present invention, the blade tip 55a is curved from the pressure surface 56b to the negative pressure surface 56a while forming a predetermined radius of curvature. In such a case, it is preferable to set the radius of curvature of the blade tip 55a to the same as the radius of not larger than 0.1 times of the diameter of the fan.

[0040] When designing the blade tip 55a as described above, static pressure of the air current flowing from the pressure surface 56b to the negative pres-

sure surface 56a is restored slowly and gently. The blades 55 during rotation are thus less likely to form a vortex stream in their trailing positions and desirably reduce the blade vortex interaction (BVI).

[0041] Figs. 10 to 14 are graphs showing operational noise of the axial flow fan of the invention as a function of a variety of designing factors of the axial flow fan.

[0042] That is, Fig. 10 is a graph showing operational noise of the axial flow fan as a function of the hub ratio of the fan. This graph shows that it is possible to accomplish a desired low operational noise of 22.3 ± 0.2 dB when the hub ratio of the blades 55 is set to $0.45 \sim 0.55$. Particularly when setting the hub ratio of the blades to 0.5, it is possible to accomplish a minimum operational noise of the fan.

[0043] Fig. 11 is a graph showing operational noise of the axial flow fan as a function of the sweep angle θ of the blades 55. This graph shows that it is possible to accomplish a desired minimum operational noise of 22.4 ± 0.2 dB when the sweep angle θ of each blade 55 is set to $32^\circ \sim 34^\circ$.

[0044] Fig. 12 is a graph showing the operational noise of the axial flow fan as a function of the pitch angle Ψ of the blades 55. This graph shows that it is possible to accomplish a desired minimum operational noise of 22.3 ± 0.2 dB when the pitch angle Ψ of each blade 55 is set to $32^\circ \pm 2^\circ$ at the blade tip 55a and to $45^\circ \pm 2^\circ$ at the blade hub (55b).

[0045] Fig. 13 is a graph showing the operational noise of the axial flow fan as a function of the maximum camber position of the axial flow fan. This graph shows that it is possible to accomplish a desired minimum operational noise of 22.5 dB when the maximum camber position is set to 0.65, with the maximum camber ratio of each blade 55 being set to 11.5% at the blade tip 55a and to 8% at the blade hub 55b.

[0046] Fig. 14 is a graph showing operational noise of the axial flow fan according to the invention as a function of the rake angle of the axial flow fan. This graph shows that it is possible to accomplish a desired minimum operational noise of 23 dB when the rake angle is set to zero.

[0047] When such an axial flow fan of this invention and a conventional axial flow fan are used in refrigerators, it is possible to obtain a graph of Fig. 15, comparatively showing the operational noise characteristics of the front sections of the refrigerators. In the graph of Fig. 15, the curve "a" is an operational characteristic curve of a refrigerator having the axial flow fan of this invention, while the curve "b" is an operational characteristic curve of a refrigerator having the conventional axial flow fan.

[0048] As shown in the graph of Fig. 15, it is possible to desirably reduce operational noise of the front section of the refrigerator using the axial flow fan of this invention by about 4.3 dB(A) in comparison with the refrigerator using the conventional axial flow fan.

[0049] Fig. 16 is a front view, showing a shroud installed around the axial flow fan in accordance with an embodiment of the present invention. Fig. 17 is a sectional view taken along the line B-B of Fig. 16, showing the construction of the shroud according to the embodiment of this invention.

[0050] As shown in the drawings, the shroud 60 of this invention is installed around the blades 55 of the fan, with an annular rim 62 being formed closely around the blades 55 while being bulged to the front of the fan in an air inlet direction. In addition, a vortex prevention means is provided on the back surface of the annular rim 62 for preventing an undesirable collision of a discharged air current against the back surface of the rim 62 in back of the blades 55, thus preventing a formation of a vortex stream.

[0051] The vortex prevention means of this invention comprises an annular skirt 64, which is perpendicularly mounted along the central circular line on the concave back surface of the rim 62 so as to project to a length L from the flat surface of the shroud 60 in the air discharging direction. In the present invention, it is possible to install two or more annular skirts 64 as desired in accordance with the size of the rim 62. The projection length L of the annular skirt 64 is preferably set to about $1 \sim 2$ mm. That is, the width of the annular skirt 64 is larger than the radius of the annular rim 62 by about $1 \sim 2$ mm.

[0052] When the shroud 60 has a large diameter rim 62, the rim 62 undesirably generates a large-sized vortex, and so it may be necessary to install a plurality of annular skirts 64 on the concave back surface of the rim 62 so as to appropriately divide the space on the concave back surface into a plurality of small sections. This finally reduces the size of a vortex formed on the back surface of the annular rim 62.

[0053] In addition, since the width of the annular skirt 64 is designed to be larger than the radius of the annular rim 62 by the projection length L, it is possible to prevent the discharged air current from an undesired collision against the concave back surface of the rim 62 and to more effectively prevent a formation of an undesired vortex on the back surface of the shroud 60.

[0054] Fig. 18 is a view, showing an air current formed around the shroud 60.

[0055] During operation of the axial flow fan of this invention, cool air flows in an axial direction of the fan under the guide of the shroud 60. In such a case, the bulged annular rim 62, formed closely around the blades 55 of the fan, guides the air to flow on its rounded front surface, thus minimizing an airflow resistance on the shroud 60.

[0056] In a conventional axial flow fan, discharged air, flowing around the edge of the fan, regrettably comes into collision on the concave back surface of the rim 62, thus forming an undesired vortex on said back surface. Such a vortex disturbs a smooth air circulation around the fan. However, in the axial flow fan of this

invention, the annular skirt 64 effectively prevents such discharged air from flowing into the concave back surface of the rim 62, but guides the discharged air so as to allow the air to flow in the axial direction of the fan. Therefore, the vortex prevention means of this invention effectively prevents a formation of a vortex in back of the fan while allowing a smooth circulation of discharged cool air around the fan.

[0057] That is, the discharged air, flowing around the edge of the blade tips 55a, regrettably moves outwardly in a radial direction of the fan and flows into the concave back surface of the rim 62 while forming a vortex. However, in the axial flow fan of this invention, the annular skirt 64, formed on the concave back surface of the rim 64 while projecting in the air discharging direction of the fan, effectively prevents such discharged air from flowing into the concave back surface of the rim 62 or from a collision against the concave back surface, thus finally preventing a formation of a vortex on the back surface. Therefore, it is possible for the discharged cool air to smoothly flow in back of the fan without forming any vortex.

[0058] Fig. 19 is a sectional view, showing the construction of a shroud having a vortex prevention means in accordance with another embodiment of this invention.

[0059] In the embodiment of Fig. 19, the vortex prevention means comprises a vortex prevention bracket 68, which has an annular shape and is mounted to the back surface of the shroud 60 to cover the concave back surface of the circular rim 62. This bracket 68 thus prevents the discharged cool air from flowing into the concave back surface of the rim 62.

[0060] The vortex prevention bracket 68 consists of an angled portion 70 mounted to the back surface of the flat portion of the shroud 60. A slope portion 72 integrally and inclinedly extends from the edge of the angled portion 70 so as to cover the concave back surface of the rim 62 prior to being mounted to the inside edge of the rim 62. The slope portion 72 of the bracket 68 thus almost completely prevents discharged cool air from flowing into the concave back surface of the rim 62.

[0061] Since the vortex prevention bracket 68 of this embodiment almost completely covers the concave back surface of the circular rim 62, it almost completely prevents discharged cool air from flowing into the concave back surface of the rim 62 and prevents a formation of a vortex around said concave back surface. This finally minimizes the airflow resistance of the axial flow fan.

[0062] That is, when discharged air, flowing around the edge of the blade tips 55a, moves outwardly in a radial direction of the fan, the air does not flow into the concave back surface of the rim 62, but is guided by the slope portion 72 of the bracket 68 to the axial direction of the fan. Therefore, the vortex prevention means of this embodiment effectively prevents such discharged air from flowing into the concave back surface of the rim

62 or from a collision against the concave back surface. This finally prevents a formation of a vortex on the back surface of the rim 62 and minimizes airflow resistance of the axial flow fan.

[0063] Fig. 20 is a graph showing operational noise of the axial flow fan with the shroud according to the invention as a function of air volume.

[0064] In the graph of Fig. 20, the curve P is of an axial flow fan with a conventional shroud, and expresses that the fan generates some upsetting operational noise of 23.6dB in the case of an air volume of 0.81 CMM. The curve Q is of an axial flow fan with the shroud according to the embodiment of Fig. 18 of this invention, and expresses that the fan generates some agreeable operational noise of 22.4dB in the case of an air volume of 0.81 CMM. On the other hand, the curve R is of an axial flow fan with the shroud according to the embodiment of Fig 19 of this invention, and expresses that the fan generates more agreeable operational noise of 21.3dB in the case of an air volume of 0.81 CMM.

[0065] In a brief description, the shrouds of this invention effectively reduce the operational noise of the axial flow fan by 1.2dB to 2.3dB in comparison with the conventional shroud.

[0066] As described above, the present invention provides an axial flow fan for refrigerators. This axial flow fan is optimally designed in a variety of designing factors, such as the number of blades, hub ratio, sweep angle, pitch angle and maximum camber ratio, thus allowing a smooth flow of cool air agreeable with both the large pressure loss and complex flow passage of refrigerators while accomplishing a reduction in air flow noise of the refrigerators.

[0067] In addition, the axial flow fan of this invention increases the blade passing frequency (BPF), which is the main frequency of fluid noise caused by a collision of the air current against the blades during operation of the fan, at least two times. The doors of a refrigerator typically designed to intercept high frequency noise effectively intercept such a high level BPF. It is thus possible to desirably reduce operational noise of refrigerators.

[0068] In the axial flow fan of this invention, the blade tip of each blade is curved from its pressure surface to its negative pressure surface while forming a predetermined radius of curvature. The blades during rotation are thus less likely to form a vortex stream in their trailing positions and desirably reduce the blade vortex interaction (BVI), in which each trailing blade undesirably confronts a vortex stream formed by a leading blade during operation of a conventional axial flow fan, thus generating fluid noise.

[0069] In addition, a vortex prevention means is provided on the back surface of the shroud's annular rim for preventing an undesirable collision of a discharged air current against the concave back surface of the rim, thus preventing a formation of a vortex stream in back of the rim and reducing air flow loss of the fan, and reduc-

ing operational noise of the fan.

[0070] Although the preferred embodiments of the present invention have been disclosed for illustrative purposes, those skilled in the art will appreciate that various modifications, additions and substitutions are possible, without departing from the scope and spirit of the invention as disclosed in the accompanying claims.

Claims

1. An axial flow fan for refrigerators, comprising a hub mounted to a rotating shaft of a motor, with a plurality of blades regularly fixed around said hub, wherein

the number of said blades is at least seven, and a hub ratio of a hub diameter to an outer diameter of said fan is set to 0.45 ~ 0.55.

2. The axial flow fan according to claim 1, wherein the number of said blades is nine.

3. The axial flow fan according to claim 1, wherein a pitch angle of each of said blades is $32^\circ \pm 2^\circ$ at a blade tip and $45^\circ \pm 2^\circ$ at a blade hub.

4. The axial flow fan according to claim 1, wherein a maximum camber position of each of said blades is set to 0.65 ± 0.05 , with camber positions being uniformly distributed on each of the blades from a blade hub to a blade tip.

5. The axial flow fan according claim 1, wherein a maximum camber ratio of each of the blades is $11.5 \pm 0.5\%$ at the blade tip and $8 \pm 0.5\%$ at the blade hub.

6. The axial flow fan according to claim 1, wherein a sweep angle of each of said blades ranges from 32° to 34° .

7. The axial flow fan according to claim 1, wherein a blade tip of each of the blades is curved from its pressure surface to its negative pressure surface while forming a predetermined radius of curvature.

8. The axial flow fan according to claim 1, wherein a rake angle of each of the blades is zero.

9. An axial flow fan for refrigerators, comprising a hub mounted to a rotating shaft of a motor, with a plurality of blades regularly fixed around said hub and a shroud installed around the blades to guide an air current, wherein said shroud consists of:

an annular rim formed closely around the blades while being bulged to the front of said fan in an air inlet direction; and

vortex prevention means provided on a concave back surface of said annular rim for preventing discharged air from forming a vortex stream in back of the fan.

10. The axial flow fan according to claim 9, wherein said vortex prevention means consists of an annular skirt mounted along a central circular line on said concave back surface of the annular rim while projecting from a flat surface of the shroud in an air flowing direction.

11. The axial flow fan according to claim 10, wherein a plurality of annular skirts are regularly installed on the concave back surface of the annular rim so as to divide a space on the concave back surface of the rim into a plurality of small sections.

12. The axial flow fan according to claim 10, wherein said annular skirt projects to a predetermined length from the flat surface of the shroud in the air discharging direction.

13. The axial flow fan according to claim 9, wherein said vortex prevention means comprising a vortex prevention bracket preventing the discharged air from flowing into the concave back surface of the rim formed on the shroud.

14. The axial flow fan according to claim 13, wherein said vortex prevention bracket consists of:

an angled portion mounted to a back surface of said flat portion of the shroud; and a slope portion integrally and inclinedly extending from the edge of said angled portion so as to cover the concave back surface of the annular rim prior to being mounted to an inside edge of said rim, thus preventing the discharged air from flowing into the concave back surface of the rim.

FIG.1 (Prior Art)

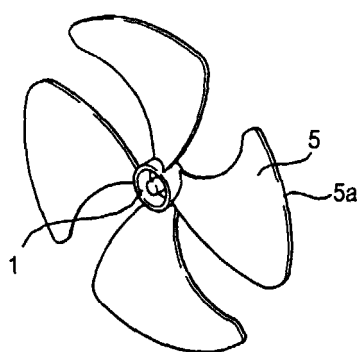


FIG.2 (Prior Art)

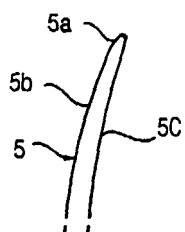


FIG.3 (Prior Art)

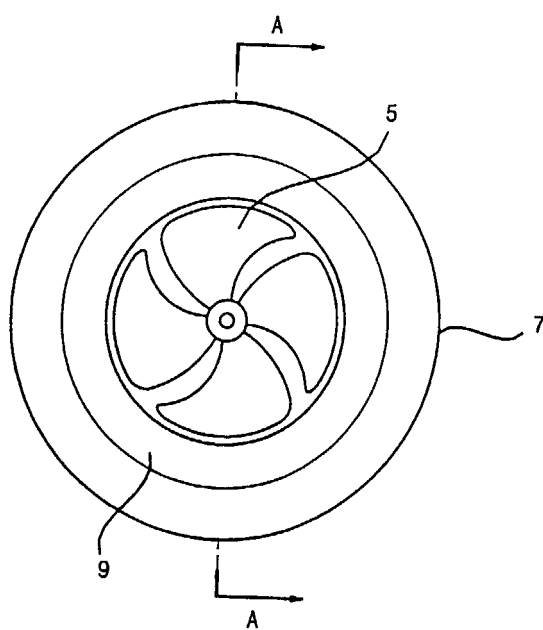


FIG.4 (Prior Art)

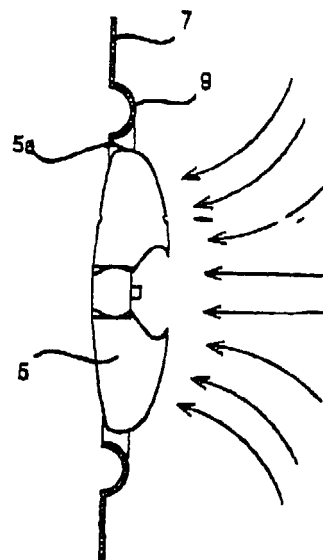


FIG.5 (Prior Art)

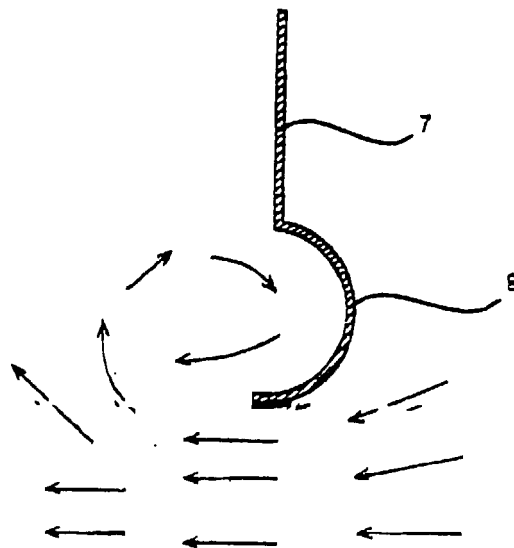


FIG.6

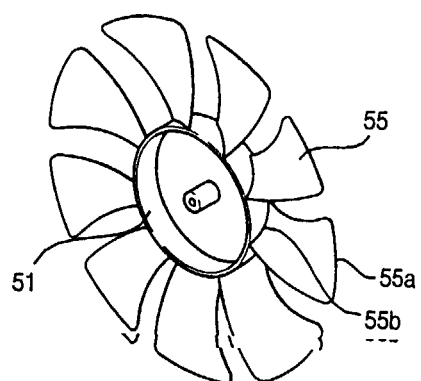


FIG.7a

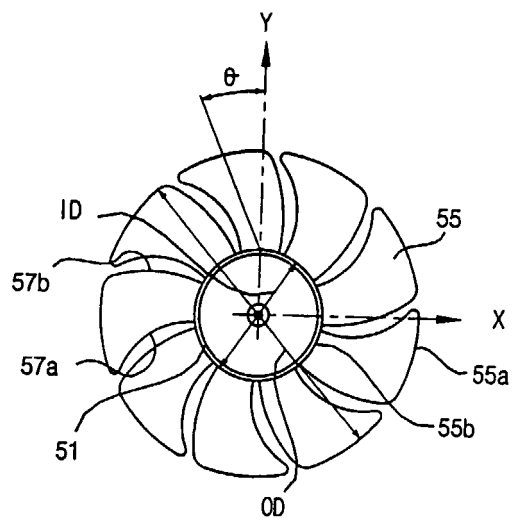


FIG.7b

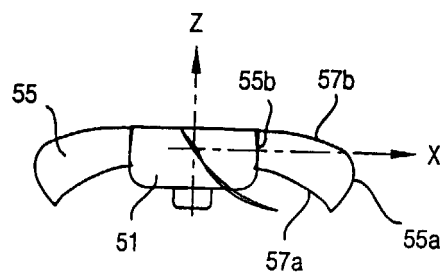


FIG.8a

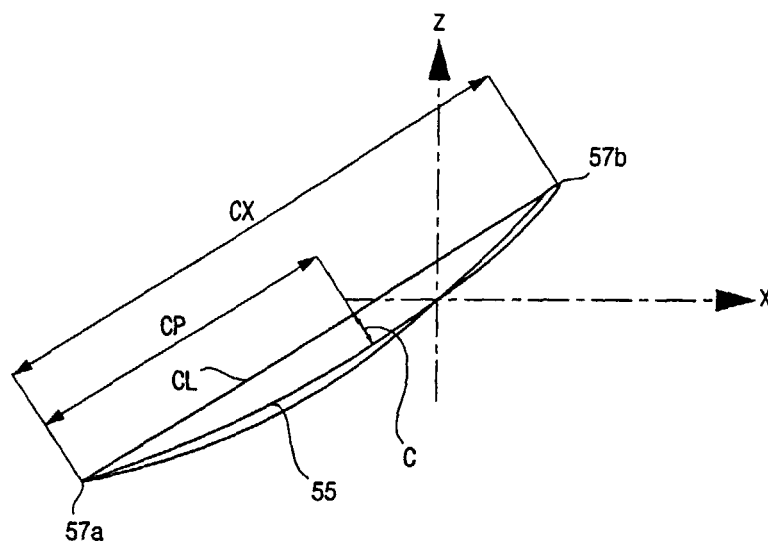


FIG.8b

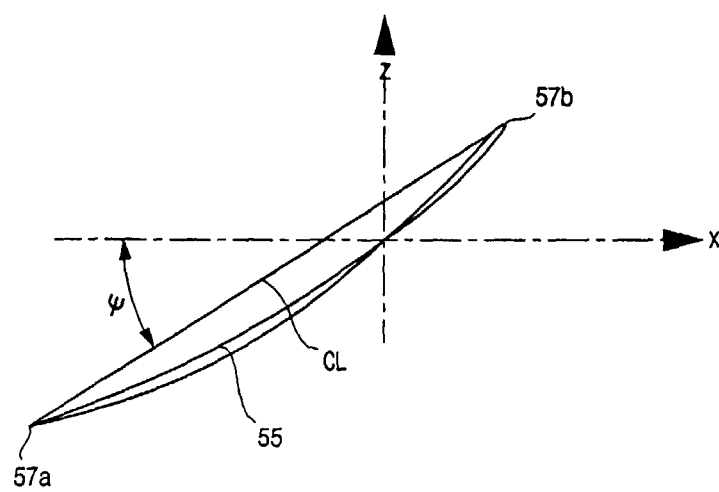


FIG.9

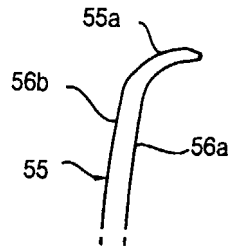


FIG.10

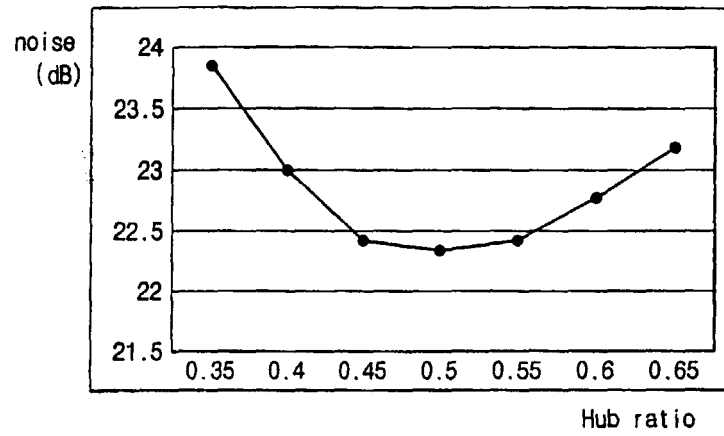


FIG.11

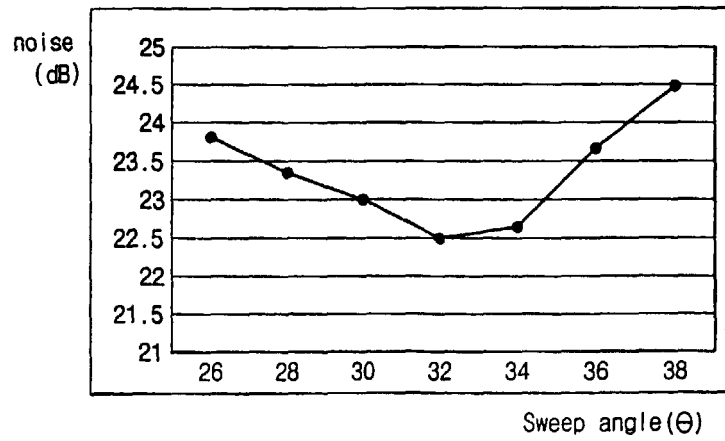


FIG.12

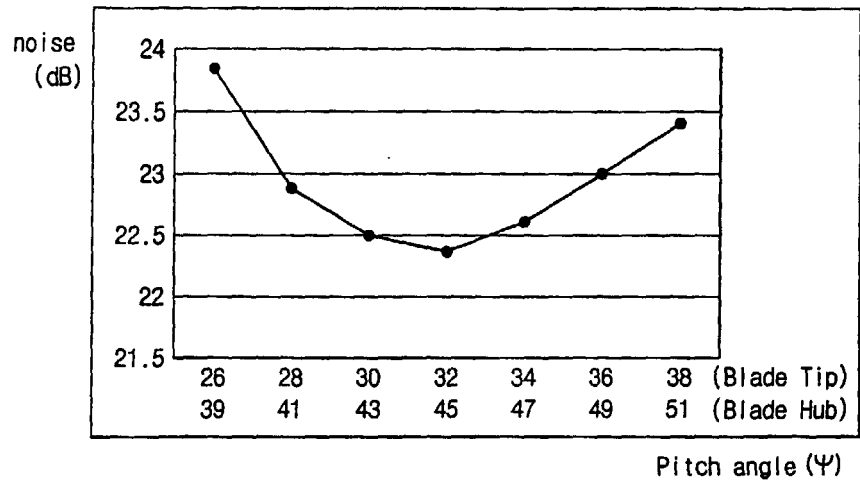


FIG.13

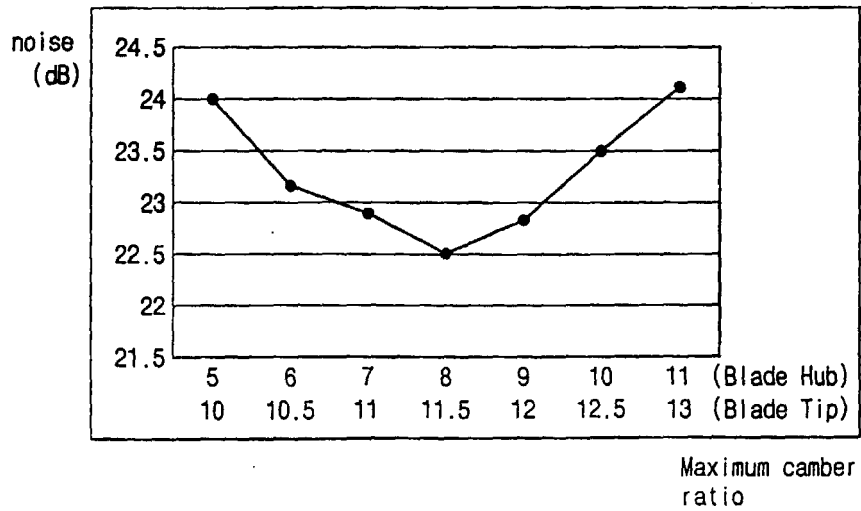


FIG. 14

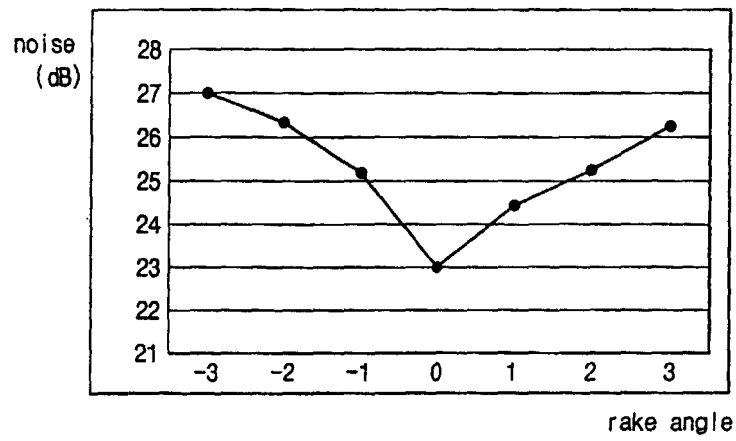


FIG. 15

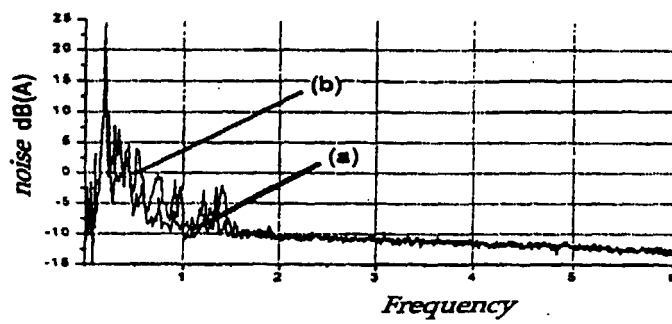


FIG. 16

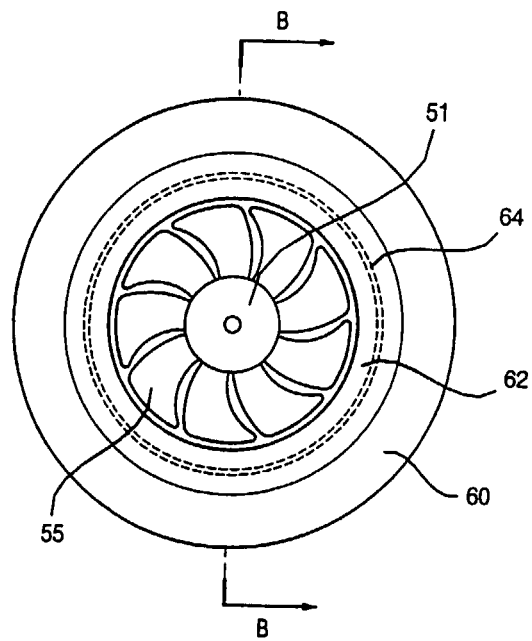


FIG.17

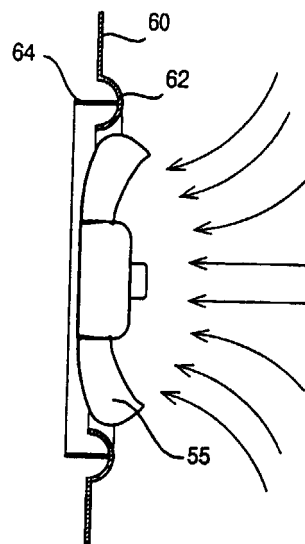


FIG.18

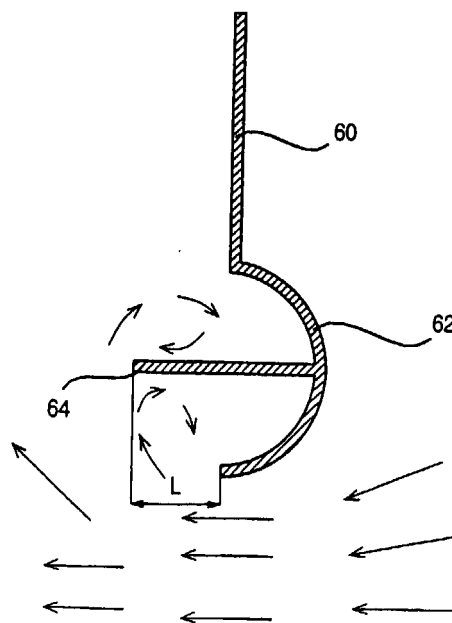


FIG.19

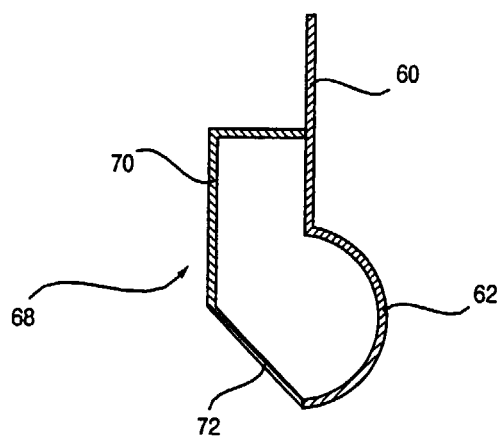


FIG.20

