

EUROPEAN PATENT APPLICATION

Application number: 90102729.2

Int. Cl.⁵: **F04D 23/00**

Date of filing: 12.02.90

Priority: 13.02.89 JP 31033/89
03.03.89 JP 51390/89

Date of publication of application:
22.08.90 Bulletin 90/34

Designated Contracting States:
DE FR GB IT

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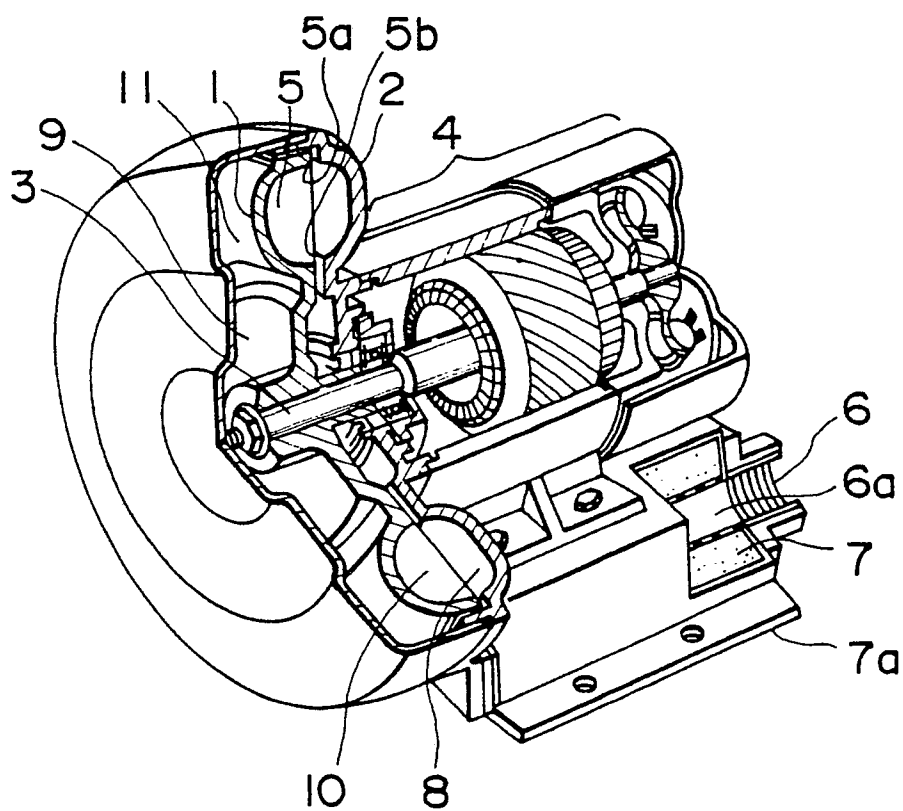
EP 0 383 238 A2

Vortex flow blower and method of manufacturing the same.

The present invention relates to a vortex flow blower used as an air source to be incorporated into general industrial machines. The vortex flow blower according to the present invention is characterized in that the shape of the blade in its impeller is three dimensionally formed such that at least the inner portion of the blade can be adapted to the three dimensional internal flow. According to the present invention, the aerodynamic performance can be sig-

nificantly improved and the size of a vortex flow blower can be reduced. Furthermore, an impeller having three dimensionally shaped blades is manufactured by independently manufacturing the shroud and the blades and coupling them, so that the impeller of a complicated shape can be readily manufactured. Furthermore, since the blade can be made of a thin and light material, the secondary moment of inertia of the impeller can be reduced.

FIG. 1



VORTEX FLOW BLOWER AND METHOD OF MANUFACTURING THE SAME

BACKGROUND OF THE INVENTION

The present invention relates to a vortex flow blower used as an air source to be incorporated into a general industrial machine such as an apparatus for transporting pulverized materials, an absorber for paper or an aeration apparatus, and, more particularly, to the shape of impeller blades capable of significantly improving the aerodynamic performance of a vortex flow blower, the shape of a casing suitable for the shape of the blades and a manufacturing method therefor.

Hitherto, the vortex flow blower has usually been provided with blades formed radially in the impeller. Since the vortex flow blower exhibits an advantage in that high wind pressure can be obtained with reduced size, a variety of disclosures and studies have been made for the purpose of improving the above-described advantage.

For example, a study is disclosed in Transaction of Japan Machinery Society, Vol. 45 (published in August 1979), P. 1108-1116. According to the study, the characteristic (i.e., characteristic about the relationship between discharge flow rate and discharge pressure) of the vortex flow blower is changed by changing the ratio R_1/R_2 , where R_1 represents the radius of a circle connecting the inner end of a blade and the axial center and R_2 represents a radius connecting the outer end of the blade and the axial center. According to this it is disclosed that both the flow rate coefficient and the pressure coefficient are higher when the value of R_1/R_2 is 0.68 than when it is 0.82, and they become further high when it is 0.75. In the vortex flow blowers which have been put into practical use, the smallest value of R_1/R_2 is about 0.68.

Although R_2 must be a small value for the purpose of reducing the size of the vortex flow blower, the following problems arise: namely, the value of R_1/R_2 must be decreased when the desired flow rate is satisfied with a reduced size of the vortex flow blower since the flow rate significantly depends upon the value of $R_2^2 (1-R_1/R_2)$. However, if the value of R_1/R_2 is reduced to 0.75 or less, the pressure coefficient becomes smaller as described above. Furthermore, since the outer radius R_2 has been reduced, peripheral speed u_2 at the outer radius R_2 is also lowered, thereby causing the discharge pressure to be excessively lowered since the pressure characteristic is determined by the product of the pressure coefficient and the square of u_2 . Therefore, R_2 must be a small value, and R_1/R_2 must be a small value and the pressure coefficient must be significantly increased in order to reduce the size of the vortex flow blower.

When improved characteristics are desired without any change in the size of the vortex flow blower, the following problem arises: namely, if the value of R_1/R_2 is increased to about 0.75 for the purpose of improving the pressure performance in the case where the value of R_1/R_2 is constant, the flow rate is inevitably reduced and, on the contrary, if the value of R_1/R_2 is reduced for the purpose of increasing the flow rate, the pressure coefficient is lowered. Therefore, when an improved characteristic is desired without changing the size of the vortex flow blower, R_1/R_2 must be reduced and the pressure coefficient must be increased.

Vortex flow blowers designed to improve their aerodynamic performance are disclosed, for example, in Japanese Patent Unexamined Publication No. 50-5914 and Japanese Patent Unexamined Publication No. 61-155696, each of which is provided with an impeller formed in such a manner that only the axial inlet angle and the exit angle of its blade are inclined at a certain angle which is respectively smaller or larger than 90 degrees. Furthermore, vortex flow blowers, although their objects are unclear, are disclosed in Japanese Utility Model Examined Publication No. 55-48158 and Japanese Utility Model Unexamined Publication No. 56-85091, each of which is provided with an impeller formed in such a manner that both or one of the inlet angle and the exit angle in the circumferential direction of its blade are or is inclined at a certain angle which is different from 90 degrees.

Further, a method of manufacturing an impeller is disclosed in Japanese Patent Unexamined Publication No. 51-57011, and according to this method the impeller is composed of two pieces divided in its axial direction in order to make a core unnecessary when forming the impeller from a casting, and the thus divided two pieces are coupled to each other afterwards.

Since the vortex flow blower exhibits an advantage in that it can serve as a clean air source with a reduced size, it has been widely used recently. Therefore, there arises a desire for the vortex flow blower which is capable of generating higher wind pressure and whose size is reduced with the discharge pressure maintained as it is. However, in the conventional technologies including the above-described technologies, only one of the exit angle in the circumferential direction, the inlet angle and the axial angle of the blade is taken into consideration and the shape of the blade is not formed so as to be adapted to the three dimensional internal flow which takes place inherently in the vortex flow blower, so that turbulence of internal flow such as swirls and stagnations cannot be prevented. There-

fore, the following problems take place: namely,

(1) it is difficult to further reduce the size of the vortex flow blower with a predetermined pressure maintained, and

(2) it is difficult to obtain higher discharge pressure with the flow rate maintained without enlarging the size of the vortex flow blower.

Furthermore, since the conventional vortex flow blowers have been insufficient in terms of noise reduction, they cannot be used as medical equipments or the like which are used in quite environments.

In addition, according to the method of manufacturing an impeller disclosed in Japanese Patent Unexamined Publication No. 51-57011, it is difficult to manufacture an impeller blade having a three dimensional shape.

Furthermore, where the impeller is manufactured die-casting or low pressure casting process, since there are problems of run or fluidity it is difficult to reduce thickness of the blade. Therefore, it is difficult to reduce the secondary moment of inertia of the impeller, thereby causing a necessity of starting torque when starting the impeller and, as a result, the size of the motor cannot be reduced.

Furthermore, the metal mold used when the impeller is manufactured by an integral molding process such as die-casting or chill-casting process is expensive, so that it is difficult to cheaply manufacture an impeller having different aerodynamic performance.

SUMMARY OF THE INVENTION

The present invention has been accomplished in view of the foregoing, and a first object of the present invention is to provide a vortex flow blower exhibiting improved aerodynamic performance in comparison with conventional vortex flow blower.

A second object of the present invention is to provide a vortex flow blower whose noise is reduced.

A third object of the present invention is to provide a vortex flow blower whose aerodynamic performance is significantly improved and whose discharge pressure can be controlled to a set value.

A fourth object of the present invention is to provide a vortex blower whose size is reduced.

A fifth object of the present invention is to provide a method of efficiently and easily manufacturing an impeller even if it has a complicated shape.

A sixth object of the present invention is to provide a method of manufacturing an impeller having reduced secondary moment of inertia.

A seventh object of the present invention is to

provide a method of cheaply manufacturing impellers having different aerodynamic characteristics by manufacturing only the blades of different shapes.

In order to achieve the above-described objects, the first aspect of the present invention lies in that the shape of the blade is formed in a proper three dimensional shape such that at least the inner portion of the blade is adapted to the three dimensional internal flow.

That is, when it is assumed that the radius of a circle connecting the inner end of the blade and the axial center is R_1 , the inlet angle of the front edge of the blade in the inner end is γ_1 , the inlet angle at the front edge of the blade in an intermediate portion between the inner end and a central portion is γ_i , the radius at a center between the inner end and the outer end is R_c and the inlet angle at the front edge of the blade in the center is γ_c , the shape of the blade is formed by a surface smoothly curved so as to make at least γ_1 , γ_i and γ_c smaller than 90 degrees and to meet the relationship of $\gamma_1 > \gamma_c$ or $\gamma_i > \gamma_c$. Further, it may be formed so as to make γ_1 smaller than 90 degrees and to meet the relationship of $\gamma_1 > \gamma_c$.

Furthermore, the position of the blade at its front edge on a circle whose radius is R_c is arranged to delay with respect to the direction of rotation of the impeller than that at its inner end.

The second aspect lies in that the shape of the blade of the impeller is three dimensionally formed such that the inner and the outer portions of the blade are adapted to the three dimensional internal flow, thereby projecting the front edge of the outer portion of the blade with respect to the direction of rotation of the impeller.

The third aspect lies in that the front edge of the outer portion of the blade is retracted with respect to the direction of rotation of the impeller and γ_0 is made larger than 90 degrees.

The fourth aspect lies in that as mentioned before the shape of the blade of the impeller is three dimensionally formed and R_1/R_2 is set to 0.75 or less and preferably to a range between 0.75 or less and 0.3 or more.

The fifth aspect lies in that the shape of the casing of the vortex flow blower is formed in such a manner that the shape of a partition wall thereof is formed so as to cause fluid to be introduced and discharged along the shape of the blade.

In order to achieve the fifth object of the present invention, the method of manufacturing an impeller according to the present invention comprises the steps of independently manufacturing the shroud and the blades and coupling the thus independently manufactured shroud and the blades so as to form the impeller. Further, as occasion demands, a filler may be filled into the corners between the base portion of the shroud and the

blades.

Furthermore, in the method of manufacturing an impeller according to the present invention, grooves into which the blades are to be inserted are formed in the annular groove formed in the shroud by the number corresponding to the number of the blades so that the impeller is formed by inserting the blades into these grooves.

Furthermore, in the method of manufacturing an impeller according to the present invention, cores each of which has such a structure that, when the impeller has been formed by casting, neighboring blades partition the annular groove of the shroud are positioned on the circumference at a predetermined interval, fluid (e.g., molten alloy) is poured between the neighboring cores and between the core and the outer mold, and the fluid is solidified so that the impeller is manufactured.

Furthermore, in the method of manufacturing an impeller according to the present invention, impeller component units each of which has neighboring blades and a part of the annular groove of the shroud formed therebetween are manufactured, and a plurality of these units are assembled to each other on the circumference so that the impeller is manufactured.

In order to achieve the above-described sixth object, in the method of manufacturing an impeller according to the present invention, the blades are made of thin and light material.

In order to achieve the above-described seventh object, the method of manufacturing an impeller according to the present invention is characterized in that the impeller is manufactured by manufacturing only the blades so as to have different shapes and coupling the thus manufactured blades and the shroud.

BRIEF DESCRIPTION OF THE DRAWINGS

Fig. 1 is a perspective view which illustrates an embodiment of a vortex flow blower according to the present invention;

Fig. 2 is a perspective view which illustrates an impeller of the vortex flow blower shown in Fig. 1;

Fig. 3 is an enlarged plan view of a part of the impeller shown in Fig. 2;

Figs. 4 to 6 are cross sectional views respectively taken along lines A-A, B-B and C-C of Fig. 3;

Figs. 7 to 14 illustrate the internal flow in an impeller;

Figs. 15 and 16 are tables showing test data of the embodiment according to the present invention in comparison with those of the conventional technology;

Fig. 17 illustrates the relationship between the flow rate coefficient and the pressure coefficient in the embodiment of the present invention in comparison with that in the conventional vortex flow blower;

Fig. 18 illustrates the relationship between the flow rate coefficient and the pressure coefficient when the inlet angle in the circumferential direction is selected to be a specific value;

Fig. 19 illustrates the pressure coefficient ratio when the angle in the circumferential direction is changed;

Fig. 20 illustrates the pressure coefficient ratio when the axial inlet angle is changed;

Figs. 21 to 33 illustrate another embodiment of the present invention, where:

Fig. 21 is a perspective view of an impeller;

Fig. 22 is a perspective view which visually expresses the angle at each of the portions of the impeller shown in Fig. 21 by means of composing with multiple planes;

Fig. 23 is an enlarged plan view which illustrates a part of the impeller shown in Fig. 21;

Figs. 24 to 26 are cross sectional views respectively taken along lines A-A, B-B and C-C of Fig. 23;

Fig. 27 is a graph which illustrates the transitions of the axial inlet angle and exit angle at each of the portions in the impeller shown in Fig. 21;

Fig. 28 illustrates test data of the embodiment according to the present invention in comparison with those of the conventional technology;

Fig. 29 illustrates the relationship between the flow rate coefficient and the pressure coefficient in the conventional technology in which only β_2 is changed;

Fig. 30 illustrates the pressure coefficient ratio with respect to the case where $\beta_2 = 90^\circ$ at the time where the flow rate coefficient, shown in Fig. 29, is 0 (i.e., in closed state);

Fig. 31 illustrates the pressure coefficient ratio obtained from test data and with respect to the case where $\beta_2 = 90^\circ$ and $\gamma_0 = 90^\circ$ at the time where the flow rate coefficient is 0 (i.e., in closed state) when the axial exit angle γ_0 is changed;

Fig. 32 illustrates the pressure coefficient ratio with respect to the case where $\beta_2 = 90^\circ$ and $\gamma_0 = 90^\circ$ at the time where the flow rate coefficient is 0 when the exit angle in the circumferential direction and the axial exit angle are changed;

Fig. 33 illustrates the relationship between the flow rate coefficient and the pressure coefficient in the case where β_1 and γ_1 in the inner portion are modified, in addition to β_2 and γ_0 in the outer portion, as in the embodiment shown in Fig. 1 and in the case where only β_2 and γ_0 in the outer portion are modified;

Fig. 34 is a plan view which illustrates another embodiment in which the angle in the circumferential direction of the front edge of the blade is smoothly changed;

Fig. 35 illustrates another embodiment of the present invention in which the shape of the front edge of the blade is changed;

Figs. 36 to 40 illustrate a further another embodiment of the present invention, where:

Fig. 36 is a perspective view which illustrates an impeller;

Fig. 37 is an enlarged plan view which illustrates a part of the impeller shown in Fig. 36;

Figs. 38 to 40 are cross sectional views respectively taken along lines a-a, b-b and c-c of Fig. 37;

Figs. 41 to 44 illustrate a further another embodiment of the present invention, where:

Fig. 41 is a perspective view which illustrates an impeller;

Figs. 42 to 44 are cross sectional views respectively taken along lines A-A, B-B and C-C of Fig. 23;

Fig. 45 and 46 illustrate a further another embodiment of the present invention, where:

Fig. 45 is a perspective view which illustrates an impeller;

Fig. 46 shows cross sections respectively taken along lines B-B, C-C, D-D, E-E and F-F of Fig. 45;

Fig. 47 is a front elevational view which illustrates the shape of a partition wall formed between an inlet port and an outlet port of a casing;

Fig. 48 illustrates noise spectrum of the vortex flow blower;

Fig. 49 is a perspective view which illustrates another embodiment of the present invention;

Figs. 50 to 54 illustrate a further another embodiment of the present invention; where:

Fig. 50 is a perspective view which illustrates the vortex flow blower in which a double-blade impeller is mounted;

Fig. 51 is a perspective view which illustrates the double-blade impeller;

Figs. 52 to 54 are cross sectional views respectively taken along lines A-A, B-B and C-C of Fig. 51;

Figs. 55 to 66 illustrate a method of manufacturing an impeller according to the present invention, where:

Fig. 55 is a front elevational view which illustrates the shape of a blade;

Fig. 56 is a vertical sectional view which illustrates the shape of a shroud;

Figs. 57 to 60 are vertical cross sectional views each of which illustrates a plastic working process when the blade and the shroud are coupled to one another;

Fig. 61 is a vertical sectional view which illustrates an embodiment in which a filler is filled in a corner portion between the blade and the shroud;

Fig. 62 is a perspective view which illustrates the blade to which a skin material is brazed;

Fig. 63 illustrates a method of manufacturing an impeller by using an ultrasonic oscillator;

Figs. 64 to 66 are perspective views each of which illustrates the shape of the blade;

Figs. 67 to 70 are vertical sectional views which illustrate another method of manufacturing an impeller according to the present invention arranged in such a manner that the blade and the shroud are coupled to one another by screw;

Figs. 71 and 72 illustrate a further another embodiment of the present invention, where:

Fig. 71 is a perspective view which illustrates a core;

Fig. 72 is a vertical sectional view which illustrates a metal mold for forming casing by using the core shown in Fig. 71;

Figs. 73 to 75 are perspective views which illustrate the structure of the core according to a further another embodiment of the manufacturing method of the present invention; and

Fig. 76 is a perspective view which illustrates the structure of the impeller manufactured by a further another method of the present invention.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Preferred embodiments of the present invention will be described with reference to the drawings. First, an embodiment of the present invention will be described with reference to Figs. 1 to 20.

Referring to Fig. 1, reference numeral 1 represents an impeller, 2 represents a casing forming an annular passage 8, and 4 represents a motor for rotating the impeller 1. The impeller 1 and the casing 2 are formed to face each other and the impeller 1 is fastened in such a manner that it can rotate with respect to the casing 2. The motor 4 is placed on the base member 7a in such a manner that the motor 4 is secured to both the base member 7a and the casing 2. An end of the annular passage 8 is communicated to an inlet passage 6a and the other end of the same is communicated to an outlet passage (not shown in Fig. 1). The inlet passage 6a and the outer passage are formed in a muffler 7 which also serves as a base member. The annular passage 8 is formed in an annular shape around the rotational center of the impeller 1, that is, around the rotational shaft 3 of the motor 4, the cross sectional shape of the annular passage 8 forming a semicircular arc when it is cut by a plane passing through the axial center

of the rotational shaft 3. A partition wall is formed between an inlet port and an outlet port each of which is communicated with the annular passage 8, the partition wall being formed with a small gap maintained for the purpose of permitting a plurality of blades 5 formed in the impeller 1 to pass through. Thus, the communication between the inlet port and the outlet port is prevented by the partition wall. The impeller 1 is constituted by a wheel 9 and a shroud 11 which are secured to the rotational shaft 3 of the motor 4 and are capable of rotating with integrated to each other. The shroud 11 has a passage 10 formed therein, the passage 10 having a multiplicity of blades 5 formed in a direction traversing the passage 10.

In the vortex flow blower of this embodiment, the shape of the blade 5 is, as shown in Figs. 2 to 6, formed in such a manner that at least the inner portion thereof has a three dimensional shape.

The air flow in the annular passage 8 will be described before it is explained about the shape of the blade 5. The air flow in the annular passage 8 becomes as shown in Figs. 7 to 11. Air introduced through an inlet port 6c passes, as shown in Figs. 7 and 8, through a passage 2a in the casing 2 formed in the impeller 1, the passage 2a being in the form of a circular cross sectional shape. The air passes through the passage 2a while swirling around the center of the circular cross section and the pressure of which is being raised due to the rotation of the blades 5 until it reaches the outlet port through which the air is discharged.

It has been found that air passes as shown in Fig. 9 to 14 as a result of visual tests and measurements of the speed of the internal flow.

Assuming, as shown in Figs. 3 to 7, that the inner end of the blade 5 is 5b, the outer end of the same is 5a and the central portion between the outer end 5a and the inner end 5b is 5c, the distribution of speed of air passing through the annular passage 8 after it has been introduced through the inlet port 6c with respect to the speed of the blade 5 becomes as shown in Fig. 10. That is, the speed of internal flow becomes positive values with respect to the direction of rotation of the impeller 1 in the region from the outer end 5a to a position near the central portion 5c, while it becomes negative values in the region from the position near the central portion 5c to the inner end 5b.

Therefore, in this embodiment, at least the more the air approaches the central portion 5c from the inner end 5b, the larger becomes the speed component of air passing through the annular passage 8 in the inverse direction to the direction of the rotation, so that the shape of the blade 5 facing the annular passage 8 is formed to be retracted in the region from the inner portion to the central

portion in order that air can flow without separation even when it passes through the portion near the central portion 5c at which air passes at high speed.

That is, in this embodiment, angle β_1 in the circumferential direction is determined so as to retract the blade 5 to the central portion 5c at the inner portion thereof, thereby making the internal flow uniform.

On the other hand, as is known, the speed distribution of the flow passing through the annular passage 8 in the traverse direction toward the rotational shaft 3 has, as shown in Fig. 11, speed vector running toward the casing 2 in a region from the outer end 5a to a position near the central portion 5c, and it has speed vector running toward the impeller 1 in the region from the position near the central portion 5c to the inner end 5b.

Therefore, in this embodiment, the axial inlet angle of the blade 5 is determined so as to be adapted to the resultant vector of the speed vector of the air passing through the annular passage 8 with respect to the blade 5 as shown in Fig. 10 and the speed vector of the air passing in the traverse direction toward the rotational shaft 3 with respect to the blade as shown in Fig. 11, that is, the vector w_i in the speed triangle shown in Fig. 14.

That is, the resultant speed vector changes in such a manner that the inlet angle of the front edge of the blade 5 is about 90 degrees at the inner end 5b and it becomes smaller in going toward the central portion 5c, so that the axial inlet angle is determined so as to be adapted to this change.

Referring to Fig. 2, a shaft hole 20 for fastening the rotational shaft 3 is formed in a central portion of the impeller 1. As shown in Fig. 3, the impeller 1 has blades 5 and passages 10 between the blades 5 formed annularly in a space between radii R_1 and R_2 from the center of the shaft hole 20. In this case, the structure is arranged in such a manner that the cross sectional shape, which is obtained by cutting the passages 10 between the blades 5 with a plane passing through the center of the shaft hole 20, forms a semicircular arc.

The cross sectional shape of the blade 5 is formed so as to be adapted to the aforesaid resultant speed vector of air in such a manner, for example, as shown in Figs. 2 to 6.

It is assumed as shown in Figs. 2 to 6 that the radius of a circle connecting the inner end 5b of the blade 5 and the center (the rotational center of the rotational shaft 3) of the shaft hole 20 is R_1 , the radius of a circle connecting the outer end 5a and the center of the shaft hole 20 is R_2 and the radius of the midpoint between the inner end 5b and the outer end 5b is R_c and, under this assumption, position 5c of point R_c in the front edge of the blade 5 is delayed from the inner end 5b when

viewed in the direction of rotation of the blade 5. Further, it is assumed that the inlet angle at the inner end 5b of the blade 5 is γ_1 and the inlet angle at the position 5c is γ_c , both γ_1 and γ_c being smaller than 90 degrees and having the different values from each other with a relationship of $\gamma_1 > \gamma_c$ held and, under this assumption, the blade 5 is formed by smoothly curved surface. Furthermore, the axial exit angle γ is formed to be 90 degrees in a region from the central portion to the outer portion. In addition, as shown in Fig. 3, the front edge of the blade 5 is formed in such a manner that it is delayed with respect to the direction of the rotation of the impeller 1 in the region from its inner end to a position slightly outer than the midpoint and it extends radially with respect to the center of the shaft hole 20 in the region outer than the above-described region. That is, as shown in Fig. 3 it is arranged in such a manner that the angle β_1 formed between the line tangent to the inner end 5b and the line connecting the midpoint 5c and the inner end 5b is less than 90° and the angle β_2 formed between the line tangent to the outer end 5a and the line connecting the midpoint 5c and the outer end 5b is 90° . The reason for this lies in that the direction of air flow is inverted at a portion slightly outer than the central portion.

The "axial angle γ " is defined, here, to be an angle formed by the smoothly curved surface in the rotational direction side of the front edge portion of the blade 5 with respect to the plane in the front edge of the blade 5. Alternatively, it may be defined with respect to the center line of the cross section of the blade 5.

The "angle β in the circumferential direction" is defined to be an angle which is in the opposite direction to the direction of rotation, among the angles formed — at the intersections between concentric circles with respect to the axial center of the impeller 1 and the front edge of the blade 5 — between the lines tangent to the above-described circles and the above-described front edge.

By forming the shape of the blade 5 in this manner, air passes through the inside portion of the casing 2 while swirling from the outer portion of the annular passage 8 formed in the casing 2 before being introduced into the inner portion of the impeller 1 along the surface of the blade 5 in the casing 2, thereby forming an internal flow passing smoothly and three dimensionally along the surface of the blade 5 without any significant speed reduction. That is, since air is introduced so as to be adapted to the inlet flow including the counter flow component in the circumferential direction, the air flow can be introduced between the blades 5 with the resistance reduced satisfactorily. The air which has reached the outer portion changes in its flowing direction due to the axial exit angle of 90

degrees, so that the direction of the internal flow is changed into the forward direction with respect to the circumferential direction and, as a result, the work is given to the fluid from the blade 5 by one swirl, thereby causing the pressure of air to be raised. In this way, a smooth internal flow passing along the blade 5 can be formed three dimensionally in at least the inside portion without any significant speed reduction, so that a flow having no excessive swirls and stagnation can be created. As a result, the discharge pressure can be raised and a vortex flow blower whose noise is low can be obtained.

Fig. 15 illustrates the ratios between the pressure coefficients in the present invention and those of the conventional example when the value of β_1 of the impeller 1 according to the present invention is varied as 100, 90, 80, 60, 45 and 20 degrees and that of γ of the same is varied as 10, 20, 45, 70, 80 and 90 degrees. The pressure coefficient ϕ_0 of the conventional example is obtained when all of β_1 , β_2 , γ_i , γ_c and γ_0 are 90 degrees. The value of γ_c when obtaining the pressure coefficient ϕ in an embodiment of the present invention was set to a value which is smaller than γ_c by 13 degrees. The value of β_2 was fixed to 90 degrees and the value of R_1/R_2 to a constant value of 0.58.

If the values in the frame are larger than 1.0, it means that the pressure coefficient is higher than that of the conventional example. If it is somewhat larger than 1.7, the pressure coefficient corresponds to 14 or more.

Therefore, the pressure coefficient can be increased to a value higher than 14 when β_1 is 45 to 80 degrees, γ_i is 20 to 70 degrees and γ_c is smaller than γ_i by 13 degrees.

Similarly to Fig. 15, Fig. 16 illustrates the values of the pressure coefficient ratio when the value of β_2 was set to 70 degrees. As shown from Fig. 16, the pressure coefficient ratio obtainable when β_2 is 70 degrees is smaller than that when β_2 is 90 degrees. However, the pressure coefficient in this case is larger than that according to the conventional example when β_1 is 15 to 80 degrees, γ_i is 20 to 70 degrees and γ_c is smaller than γ_i by 13 degrees.

That is, a fact is shown that the axial inlet angle and the inlet angle in the circumferential direction of the front edge of the blade 5 are critical factors of the aerodynamic performance.

Fig. 17 illustrate the relationship between the flow rate coefficient ϕ and pressure coefficient ϕ in each of the embodiment of the present invention and the conventional vortex flow blower. It can be understood that both the flow rate coefficient and the pressure coefficient in the embodiment of the present invention are higher than those in the conventional one.

Fig. 18 illustrates the relationship between the flow rate coefficient ϕ and the pressure coefficient ϕ when the inlet angle β_1 in the circumferential direction is set to 20 degrees and 90 degrees. As seen from this drawing, both the flow rate coefficient and the pressure coefficient are higher when the inlet angle β_1 in the circumferential direction is set to 20 degrees.

Fig. 19 illustrates the ratios of the pressure coefficients when the inlet angle β_1 in the circumferential direction is varied. In this case, the exit angle β_2 in the circumferential direction is fixed to 90 degrees and they are compared with the case in which both β_1 and β_2 are 90 degrees. As seen from this drawing, in the range from 90 degrees to 20 degrees the lesser the value of β_1 is, the larger becomes the pressure coefficient ratio.

Fig. 20 illustrates the ratios of the pressure coefficients in the case where the axial inlet angle γ_1 in the front edge of the blade 5 is varied with both β_1 and β_2 set to 90 degrees, as a standard in the case where both γ_1 and β_2 is set to 90 degrees. As seen from this drawing, the lesser the value of γ_1 is, the larger becomes the pressure coefficient ratio.

As described above, at least the axial inlet angle in the inner portion in the front edge of the blade 5 and the inlet angle in the circumferential direction are determined so as to be adapted to the resultant vector of the speed vector of the air flow passing through the annular passage 8 and the speed vector of the air flow passing in the traversing direction toward the rotational shaft in the annular passage 8 and thereby to form the three dimensionally shaped blades. Therefore, turbulence of the internal flow such as swirls and stagnation of air introduced into the internal portion can be satisfactorily prevented and, as a result, the aerodynamic performance can be significantly improved in comparison with the conventional vortex flow blower. That is, an advantage can be obtained in that the aerodynamic performance can be significantly improved by forming the inner portion of the blade into a three dimensional shape which can be adapted to the flow of fluid. As a result, the drawback inherent in the conventional vortex flow blower in that the pressure coefficient is inevitably reduced when the ratio R_1/R_2 is reduced to 0.75 or less for the purpose of reducing the size of the vortex flow blower can be overcome. Therefore, even if the ratio R_1/R_2 is set to 0.75 or less and 0.3 or more, the discharge pressure can be significantly increased in comparison with the conventional vortex flow blower and, as a result, an advantage can be obtained in that the outer diameter of the impeller can be reduced and the size of the vortex flow blower can thereby be reduced.

Then, an another embodiment of the present

invention in which the discharge pressure is further raised will be described with reference to Figs. 21 to 35. In this embodiment, the shape of the blade 5 from the inner portion to the central portion thereof is formed as shown in Figs. 2 and 3. Further, as described before, the speed distribution of air with respect to the speed of the blade 5 in the annular passage 8 becomes, as shown in Fig. 10, positive values with respect to the direction of the rotation of the impeller 1 and, in the portion from the central portion 5c to the outer end 5a, the speed component in the annular passage 8 becomes steeply increased in the forward direction with respect to the direction of the rotation of the blades 5. Therefore, the shape of the blade facing the annular passage 8 is formed to project from the central portion 5c to the outer end 5b in the direction of the rotation of the blade 5.

That is, in this embodiment, the exit angle β_2 in the circumferential direction is determined to 90 degrees or more in order to make the air flow on the outer side uniform by forming the blade 5 in such a manner that it projects from its central portion 5c toward the outer end 5a.

On the other hand, as mentioned before, the axial outlet angle γ is determined so as to be adapted to the vector w_o in the speed triangle shown in Fig. 12.

Assuming that the inlet angle at the front edge of the blade 5 in the outer midpoint at which the radius is a value expressed by $R_o = (R_2 + R_c)/2$ and that in the inner midpoint at which the radius is a value expressed by $R_i = (R_1 + R_c)/2$ are respectively γ_o and γ_i , the shape of the blade 5 is formed by smoothly curved surface (see Figs. 22 to 26) formed in such a manner that both γ_c and γ_i are smaller than 90 degrees and the relationships of $\gamma_o > \gamma_c$ and $\gamma_i > \gamma_c$ are met, as shown in Figs. 24 to 26 and 27. Air introduced so as to be adapted to the inlet flow including the counter flow component in the circumferential direction and having reached the outer portion changes the direction of the internal flow into the forward direction between the blades 5 since the axial exit angle γ_o is provided. Furthermore, since the exit angle β_2 in the circumferential direction is provided, the slow speed flow near the midpoint and the high speed flows in the vicinity of the outer and inner ends of the blade 5 can be synchronized with one another. As a result, stagnation causing internal loss can be prevented, the swirling component can be increased and the change in air speed between the blades 5 can be reduced. Since the axial exit angle γ_o and the exit angle β_2 in the circumferential direction are provided as described above, the work obtainable by one swirl of the blade 5 can be made larger and the internal loss taken place in the action of the blade 5 can be restricted. As a result, the obtain-

able pressure can be raised.

The exit angle β_2 in the circumferential direction causes, between the blades, the flow near the midpoint whose internal speed is slow and the flows in the vicinity of outer and inner ends of the blade 5 whose internal speeds are high to be synchronized with one another. As a result, turbulence of the flow owing to stagnation, which causes the internal pressure loss, can be prevented.

As a result of the shape of the blade 5 in which the axial exit angle γ_0 and the exit angle β_2 in the circumferential direction are provided, the blade 5 acts to form a three dimensional smooth internal flow whose change in speed can be reduced in the passage 8, so that the aerodynamic performance exhibiting a significantly high pressure can be obtained.

The experimental results about the blade 5 whose outer shape is three dimensionally formed are shown in Fig. 28 in comparison with the conventional vortex flow blower in which the shape of blade is set in such a manner that $\beta_1 = 90$ degrees, $\gamma_1 = 90$ degrees, $\beta_2 = 90$ degrees, $\gamma_0 = 90$ degrees and $R_1/R_2 = 0.58$. As seen from this drawing, when the outer shape is three dimensionally formed as described above, the pressure coefficient can be improved twice or more. In an experiment about the vortex flow blower having the conventional two dimensionally formed blade in which only the exit angle β_2 in the circumferential direction was taken into consideration, as shown in Figs. 29 and 30 a satisfactory maximum pressure coefficient was displayed when β_2 was about 90 degrees. However, if the axial exit angle γ_0 is varied, the pressure coefficient becomes larger in comparison with the conventional vortex flow blower. Because of the above-described reason, with respect to the embodiment shown in Fig. 23 the pressure coefficient can be significantly improved by simultaneously changing the axial exit angle γ_0 to 45 degrees and the exit angle β_2 in the circumferential direction to 115 degrees.

Fig. 32 is a map showing the pressure coefficient ratios when the axial exit angle γ_0 and the exit angle β_2 in the circumferential direction are varied. As seen from this map, the pressure coefficient ratio can be significantly improved in the regions of $10^\circ \leq \beta_2 \leq 135^\circ$ and $20^\circ \leq \gamma_0 \leq 70^\circ$.

Fig. 33 is a graph which illustrates the experimental results when the outer portion of the blade 5 is three dimensionally formed in addition to the inner portion of the same which has been three dimensionally formed. As seen from this graph, the pressure coefficient can be further improved by three dimensionally forming the blade 5 as a whole, thereby making it possible to obtain a pressure coefficient of about 25.

In this embodiment, the impeller 1 is, as shown

in Fig. 23, arranged to have the blade 5 whose shape at the front edge is formed in such a manner that its central portion 5c connecting the inner portion and the outer portion of the blade 5 is steeply changed in its angle, but as shown in Fig. 34 the shape of the blade may be modified in such a manner that the angle is gradually changed from the inner end 5b to the outer end 5a.

In the above-described embodiments, the blade 5 is retracted with respect to the direction of the rotation of the impeller 1 in the inner side of the blade 5, while it is radially set or forwarded in the portion outer than the midpoint 5c, but as shown in Fig. 35 in the case where the blade 5 has been three dimensionally formed the front edge shape of the blade may be arranged to have such a shape as shown by symbols d and f in its inner portion and such a shape as shown by symbols a and c in its outer portion. Alternatively, these shapes shown by symbols a-c and d-f may be combined.

Another embodiment of the present invention in which the blade is formed in this manner will be described with reference to Figs. 36 to 40. In this embodiment, the front edge of the outer portion of the blade 5 is arranged to have the curve c shown in Fig. 35. That is, the cross section of the outer portion of the blade 5 is retracted with respect to the direction of the rotation of the impeller 1 as shown in Fig. 38 in comparison with position of that shown in Fig. 23. Namely, the blade is formed such that the axial edge γ in the inner portion becomes, as shown in Fig. 40, an angle which is similar to γ_1 shown in Fig. 26, that the angle in the central position becomes slightly smaller than 90 degrees as shown in Fig. 39 and that the axial exit angle in the outer portion becomes larger than 90 degrees as shown in Fig. 38.

A further another embodiment of the present invention will be described with reference to Figs. 41 to 44. As shown in Fig. 44, in this embodiment, the blade 5 is formed in such a manner that the axial inlet angle γ_1 is set to an angle similar to that shown in Fig. 23. The blade 5 is further formed to have at its central position 5c such an axial angle γ_c as shown in Fig. 43 and to have an intermediate portion 32 retracting with respect to the direction of the rotation of the impeller 1 as shown in Fig. 41. The outer portion of the blade 5 is formed to have the axial exit angle γ_0 which is larger than 90 degrees as shown in Fig. 42.

Since the front edge of the blade 5 is retracted with respect to the direction of the rotation of the impeller 1 and γ_0 is arranged to be larger than 90 degrees, the work given by the blade 5 to air can be reduced. As a result, an effect can be obtained in that the discharge pressure and the necessary power can be controlled to a low level.

Next, an embodiment in which in Fig. 35 the

front edge of the blade 5 is formed to have curve f in its inner portion and to be retracted outwards with respect to the direction of the rotation of the impeller 1 in its outer portion is shown in Figs. 45 and 46. Fig. 45 is a partial front elevational view which illustrates the impeller 1 according to this embodiment. Although the blade does not appear as a section, it is illustrated with hatching for the purpose of having the torsional direction understood easily. Referring to this drawing, the parts given the same reference numerals as those shown in Fig. 3 are the same parts.

Fig. 46 illustrates cross sectional shapes of the blade at the radii B, C, D, E and F of Fig. 45. The outer shape (cross sections B and C) of the blade 5 is formed in such a manner that the axial exit angle γ_2 in the front edge 35 of the blade 5 is larger than 90 degrees. As a result, the shape of the blade 5 is formed in such a manner that it forms a projecting curve in the direction opposite to the direction of the rotation with respect to a line connecting G and H. The front edge 35 of the blade 5 is formed in its inner portion (cross sections E and F) in such a manner that it projects with respect to the direction of the rotation. As a result, the shape of the blade 5 is formed in such a manner that it forms a projection curve in the direction of the rotation with respect to a line connecting H and I and, furthermore, the axial inlet angle γ_1 is smaller than 90 degrees. Furthermore, the front edge 35 of the blade 5 projects by θ_0 in the direction of the rotation with respect to a radiant line relative to the axial center on cross section D in the central portion. Since the axial inlet angle γ_1 in the front edge of the blade 5 in its inner portion is formed to be substantially the same as the incidental angle γ_{a1} of the flow and the axial inlet angle γ_1 in the front edge of the blade 5 is, on the other sections (sections D and E) in the inner portion of the blade 5, set to be adapted to the incidental angles γ_{e1} and γ_{d1} which differ from each other in the degree and the direction, swirls and stagnation can be reduced. Since the outer portion is formed as shown in section B of Fig. 46 in such a manner that the axial exit angle γ_2 is made sufficiently larger than 90 degrees, the velocity component cu_2 in the circumferential direction is reduced, the advance angle of the swirling flow in the circumferential direction is reduced and the number of swirls taken place from the inlet port to the outlet port can be increased. Furthermore, the discharge pressure can be controlled to a predetermined value.

Fig. 47 illustrates the shape of a partition wall 25 for partitioning the inlet port and the outlet port formed in the casing 2, the partition wall being capable of significantly eliminating noise. The casing 2 has a circular arc passage 8 whose cross

section facing in the direction running parallel to the axial line of the rotational shaft 3 is in the form of a semicircular arc groove. The groove is provided with a partition wall 25 in a part thereof, the partition wall 25 facing the impeller 1 with a small gap retained therebetween. An end of the circular arc passage 8 is connected to the inlet side passage 6a and the other end of the same is connected to the discharge side passage 6b. The inlet side passage 6a and the outlet side passage 6b are provided to run parallel to each other in the muffler 7 which also serves as the base member.

A guide 26 adjacent to the inlet port is provided in a portion of the partition wall 25 adjacent to the inlet port. A front portion 26a of the guide 26 adjacent to the inlet port is arranged to be substantially horizontal so as to make the blade 5 cut (intersect the front edge of the blade 5) from outside. It is considered that the front portion 26a acts to smoothly introduced air, which has been introduced into the circular arc passage 8 through the inlet port 6c, to the inlet port (the portion in which the arrows face the left hand direction in Fig. 11) of the blade 5. When viewed from the axial direction, the inlet port 6c is hidden behind the guide 26 adjacent to the inlet port. This acts to prevent noise generated in the circular arc passage 8 from being directly transmitted to the passage 6a adjacent to the inlet port for the purpose of insulating noise.

A guide 28 adjacent to the outlet port is provided with the partition wall 25 adjacent to the outlet port. The front end 28a of the guide 28 adjacent to the outlet port is formed in such a manner that its substantially central portion 28b (the portion which agrees with a point of the blade 5 at which the flow is inverted) projects in the direction opposite to the direction F of the rotation of the impeller 1 so as to make the blade 5 cut (intersect the front edge of the blade 5) from inside. It is considered that the front end 28a acts to guide air to be discharged from the circular arc passage 8 to the outlet port 6d so as to be smoothly discharged from the outlet portion (the portion in which arrows face the right hand direction in Fig. 11) of the blade 5. Further, when viewed from the axial direction, the outlet port 6d is substantially hidden behind the guide 28 adjacent to the outlet port. This acts to prevent noise generated in the circular arc passage 8 from being directly transmitted to the passage 6b adjacent to the outlet port for the purpose of insulating noise.

Fig. 48 is a graph which illustrates data about noise actually measured when a vortex flow blower composed by combining the casing 2 shown in Fig. 47 and the impeller 1 shown in Fig. 36 is operated.

It can be clearly seen that the guide 28 adjacent to the inlet port and the guide 26 adjacent to

the outlet port significantly assist to reduce noise when compared with noise data shown in Fig. 48 in the case where the vortex flow blower from which the guide 26 adjacent to the inlet port and the guide 28 have been removed is operated.

In an experiment in which dimension L from 26b to 28b (the portion which agrees with the point of the blade 5 at which the direction of the flow is inverted) in the circular arc passage 8 was selected to meet the following relationship:

$$L = \frac{1}{4} \lambda (2n + 1)$$

where $\lambda = C/f$

$f = Z \times N$

Z: the number of the blades 5

N: the rotational speed of the shroud

C: acoustic velocity

$n = 0, 1, 2, 3, \dots$

the maximum noise level shown in Fig. 48 was further lowered by 4 dB.

An another embodiment of the present invention is shown in Fig. 49. In this embodiment, the impeller 1 having the blades 5 is disposed on the side adjacent to the motor 4 and the casing 2 is disposed to face the impeller 1. As a result, the degree of the overhang of the impeller 1 can be reduced. In this manner, since the impeller 1, which is a body of rotation, is disposed adjacent to the bearing portion, vibrations of the impeller 1 can be significantly reduced, thereby causing the durability against the radial loads to be improved.

An another embodiment of the present invention is shown in Figs. 50 to 54. In the embodiment shown in Figs. 50 to 54, an impeller, which is a double blade impeller having on its both sides the shape of the blade shown in Figs. 23 to 27, is employed. Fig. 50 is a perspective view which illustrates the vortex flow blower in which the double blade impeller is mounted. The parts given the same reference numerals as those shown in Fig. 1 are the same parts. In this embodiment, the casing 2 is formed so as to cover the both sides of the double blades. The annular passage 8 is formed on both sides of the double blades. Partition walls are provided on both sides of the casing 2 so as to hinder the communication between the outlet port 6d and the inlet port 6c. The inlet side passage 6a and the outlet side passage 6b are provided adjacent to the motor 4.

By constituting in this manner, a vortex flow blower exhibiting a high pressure coefficient and capable of obtaining a large wind quantity can be provided. Furthermore, another effect can be obtained in that the outer diameter of the casing can be reduced and the size of the vortex flow blower can thereby be reduced.

Next, a method of manufacturing an impeller of the vortex flow blower according to the present invention will be described.

An embodiment of the impeller is shown in Figs. 55 to 63. In this embodiment, the blade 5 as shown in Figs. 64 to 66 and the shroud 11 are independently formed. Then, the shroud 11 having the annular groove 45 and a plurality of blades 5 are coupled and secured to each other so that the impeller 1 is manufactured.

In this manner, by forming the blades 5 and the shroud 11 independently, the shroud 11 can be manufactured by using a mold formed two dimensionally, so that it becomes possible to be mass-produced by the die-casting or metal mold casting process. Further, even if the blade 5 is in the form of a complicated shape, it becomes possible to be die-cast or press-formed, so that the impeller having the three dimensionally shaped blades can be easily manufactured.

Further, also in an another embodiment described later, the blade 5 can be made of a thin and light weight material since the blades 5 are independently manufactured as described above. Therefore, an effect can be obtained in that the secondary moment of inertia of the impeller can be reduced.

Further, as shown in Figs. 64 to 66, since only the blades 5 can be formed to have various shapes, impellers having different aerodynamic performances can be easily manufactured.

In a manufacturing method shown in Figs. 55 and 56, the shroud 11, in which the annular groove 45 is formed and a plurality of insertion holes 40 are formed, and the blade 5 provided with a plurality of caulking projections 41 are manufactured. The shroud 11 and the blade 5 are coupled to each other in such a manner that the caulking projections 41 formed on the blade 5 are inserted into the insertion holes 40 formed in the shroud 11, and then they are secured by plastically working the caulking projections 41.

The method of plastically working may be a cold working or a hot working. It is preferable in terms of the appearance after subjected to the plastic working that the following method be employed: namely, as shown in Fig. 57 an upper electrode 42 having a predetermined conductivity and high temperature strength and a lower copper electrode 43 are used and only the caulking projections 41 are plastically worked with heat generated by an electric current being applied thereto.

Further, as occasion demands, as shown in Fig. 58 when the impeller is manufactured by fitting the blade 5 within the annular groove 45 formed in the shroud 11 before being press formed, the blade 5 can be stabilized and further satisfactorily plastically deformed at the time of caulking, so that the airtightness between the blades 5 can be also improved.

Figs. 59 and 60 illustrate the cross sectional

shape of the impeller which has been cut in the circumferential direction relative to the rotational center. As shown in Fig. 59, an insertion groove 44 having a width which is slightly narrower than the width of the blade 5 is formed in the annular groove 45 formed in the shroud 11. The blade 5 is press-fitted into the insertion groove 44. As a result, the airtightness between the blade 5 and the shroud 11 can be maintained. Further, as shown in Fig. 60, the fastening force can be further increased when the blade 5 having the caulking projections 41 is press-fitted and the caulking projections 41 are plastically worked.

When the airtightness is desired to be improved, the corner portions between the blades 5 and the shroud 11 may be filled with a filler 46 as shown in Fig. 61. Since the filler 46 acts to permit air to smoothly flow in addition to improving the airtightness, it is preferable from a view point of improving the aerodynamic performance. As shown in Fig. 62, the filler 46 can be easily formed by brazing the blade 5, to which a skin material 47 of the low melting point has been brazed, in a furnace.

An another embodiment is shown in Fig. 63. When the brazing shown in Figs. 61 and 62 is performed, flux must be applied and then removed after it has acted its roll. However, as shown in Fig. 63 when the impeller 1, in which the blade 5 has been secured to the shroud 11 by being press-fitted or by caulking its projections, is ultrasonic soldered in a jet type soldering tank 17 provided with a ultrasonic oscillator 16 while rotating the impeller 1, an oxide film formed on the surface to be soldered is broken by the supersonic erosion action, so that the application of the flux becomes unnecessary, thereby making it possible to efficiently manufacture the impeller 1 exhibiting excellent airtightness.

An another manufacturing method can be employed in which an adhesive is applied to the insertion groove 44 formed in the annular groove 45. As a result, the shape shown in Fig. 61 can be easily formed. That is, when the blade 5 is press-fitted into the insertion groove 44 formed in the shroud 11, a part of the adhesive overflows to the corner portion and solidifies, thereby causing an effect similar to that obtainable when the filler has been filled.

According to the above-described manufacturing methods, impellers of complicated shapes can be easily manufactured and the thus obtained impellers can exhibit satisfactory airtightness.

An another embodiment of the method of manufacturing an impeller will be described with reference to Figs. 67 to 70. In this embodiment, the blade 5 and the shroud 11 which have been independently manufactured are coupled to each

other by using a screw.

In an embodiment shown in Fig. 67, the shroud 11 may be secured to the wheel 9 by a screw 48 or it may be secured as shown in Figs. 68 and 70 in such a manner that a part of the blade 5 is expanded so as to become an expansion portion 49 and a screw hole 50 is formed in the expansion portion 49 so as to be secured by the screw 48. It is preferable in terms of the performance that the expansion portion 49 be formed on the back side of the blade 5. Alternatively, a ring 51 connecting the outer front end of the blade 5 is manufactured integrally with the blade 5 and the wheel 9, and the shroud 11 is, as shown in Fig. 69, inserted between the ring 51 and the wheel 9 so as to be secured.

The wheel 9 may be integrally formed as a whole or only a part of the wheel 9 may be integrally formed with the blade 5.

In this way, since the blade and the shroud are independently manufactured and then they are coupled to each other, the mold can, of course, be manufactured easily and the mold can be readily removed after the casting has been completed. Therefore, impellers of a complicated shape can be readily manufactured.

Further, as described above, it is possible to form the annular groove 45 in such a manner that its width becomes smaller than that of the blade 5 and to provide the blade 5 in the groove 45 by inserting it while being elastically deformed, by using the adhesive or by using the filler.

An another embodiment of the method of manufacturing an impeller will be described with reference to Figs. 71 and 72. In this embodiment, cores 55 each of which has a shape partitioning the annular groove with the neighboring blades 5 are first manufactured. The thus manufactured cores 55 are positioned at a predetermined interval away from each other on the circumference and are placed in an outer mold 58 having a circular groove 57 formed therein. In this state, parts of the neighboring cores, that is, a projection 59 of either core 55 and a notch 60 of the neighboring core 55 are disposed to overlap each other at a certain interval. Incidentally, the reference numeral 61 represents a lower mold. Then, a melt is poured into a space 56 between the neighboring cores 55 and between the outer mold 58 through a pouring hole 62 so as to be subsequently solidified. As a result, the blade 5 and the shroud 11 can be integrally formed. In this embodiment, also the wheel 9 is integrally formed by a space 63.

In the case where the impeller 1 is made of aluminum casting, it is preferable that the core 55 be made a shell core. In the case where it is formed by injecting synthetic resin, the core 55 may be made of silicone rubber.

An another embodiment will be described with

reference to Figs. 73 to 75. In this embodiment, the core 55 is constituted by a first portion 65 positioned behind a core neighboring either core 55, a second portion 66 positioned behind a core 55 neighboring the other core 55, and a third portion 67 positioned between the first portion 65 and the second portion 66 and not positioned behind any core 55. After the melt has been solidified, the third portion 67 is firstly drawn out, and then the first portion 65 and the second portion 66 are drawn out so that the impeller is formed.

As a result, not only the impeller of a simple structure but also the impeller of a complicated structure can be readily manufactured since the same shape cores may be prepared by the number corresponding to the number of the blades.

An another embodiment is shown in Fig. 76. In this embodiment, a shroud 11a constituting a part 45a of the annular groove 45, which lies between a blade 5A and a neighboring blade 5B, is manufactured as a unit and, thereafter, a plurality of shrouds 11a thus manufactured are disposed on the circumference and are coupled to one another so that the impeller is manufactured.

As a result, every component constituting the impeller and having the same shape can be manufactured as a unit, so that the impeller can be readily manufactured by assembling these components. Therefore, impeller of a complicated shape can be readily manufactured.

The above-described methods of manufacturing an impeller can, of course, be applied to impellers having the conventional shape.

The first advantage according to the present invention can be obtained from the blade formed in such a manner that at least its inner portion is three dimensionally formed, thereby causing air to be smoothly introduced so as to be adapted to the speed vector of the swirling air flow. As a result, the discharge pressure can be significantly raised.

The second advantage can be obtained from the blade formed in such a manner that its shape is three dimensionally formed so as to be adapted to the speed vector of the swirling flow. Therefore, swirls and stagnation can be significantly prevented. As a result, a low noise vortex flow blower can be obtained.

The third advantage can be obtained from the partition wall formed in such a manner that its front end adjacent to the inlet port of the vortex flow blower is cut by the blade from the outside while the front end of the same adjacent to the outlet port is cut by the blade from the inside. Therefore, the air flow from the inlet port to the circular arc passage and the air flow discharged from the circular arc passage through the outlet port can be made smooth. As a result, noise can be extremely reduced.

The fourth advantage can be obtained from the blade formed in such a manner that the shape of the blade in the impeller is three dimensionally formed as mentioned before and R_1/R_2 can thereby be set to 0.75 or less and 0.3 or more. As a result, the size of the vortex flow blower can be reduced.

The fifth advantage can be obtained from the blade formed in such a manner that the shape of the blade at the outer portion of the impeller is retracted and the axial outlet angle is arranged to be 90° or more. Therefore, work given by the blade to air can be restricted. As a result, the discharge pressure and the required operating power can be controlled to a low level.

The sixth advantage can be obtained from the method of manufacturing an impeller, which is constituted in such a manner that the blade and the shroud are independently manufactured and then they are coupled to each other. Therefore, impeller of a complicated shape can be readily manufactured.

Further, as occasion demands, the airtightness can be improved and the flow can be made smooth by using a filler or an adhesive.

The seventh advantage lies in that the secondary moment of inertia of the impeller can be reduced and the starting torque required for the motor can be reduced since the blade can be independently manufactured and made of a thin and light material.

The eighth advantage lies in that impellers of different shapes can be readily manufactured since only the blade can be independently manufactured and, as a result, impellers of different aerodynamic performance can be readily manufactured.

Claims

1. A vortex flow blower comprising: a casing having an annular passage and inlet and outlet ports each of which is communicated with said annular passage; a rotatable impeller having a plurality of blades for raising the pressure of fluid by giving a swirling flow between them and said annular passage; and a motor for rotating said impeller, wherein the shape of said blade is formed by a curved surface at least in its region from the inner end to the central portion thereof so as to make both γ_1 and γ_c smaller than 90° and to meet the relationship of $\gamma_1 > \gamma_c$, where γ_1 and γ_c represent the axial inlet angles of said blade's front edge respectively at said inner end and at said central portion.

2. A vortex flow blower comprising: a casing having an annular passage and inlet and outlet ports each of which is communicated with said

annular passage; a rotatable impeller having a plurality of blades for raising the pressure of fluid by giving a swirling flow between them and said annular passage; and a motor for rotating said impeller, wherein the curved surface shape of said blade is formed in such a manner that at least the inner portion of said blade is formed so as to cause fluid to be introduced with more inclined at the central portion with respect to said impeller than at the inner end of said blades.

3. A vortex flow blower comprising: a casing having an annular passage and inlet and outlet ports each of which is communicated with said annular passage; a rotatable impeller having a plurality of blades for raising the pressure of fluid by giving a swirling flow between them and said annular passage; and a motor for rotating said impeller, wherein the shape of said blade is formed by a curve surface formed in such a manner that the axial angle of said blade's front edge is smaller than 90 degrees at the inner end thereof, is gradually decreased from said inner end toward the central portion thereof and is increased from said central portion to the outer portion thereof.

4. A vortex flow blower comprising: a casing having an annular passage and inlet and outlet ports each of which is communicated with said annular passage; an impeller having a plurality of blades rotating opposite to said annular passage; and a motor for rotating said impeller, wherein said blade is three dimensionally formed in such a manner that it retracts with respect to the direction of rotation of said impeller and the axial inlet angle becomes gradually decreased to smaller than 90 degrees from the inner end to the central portion, and the axial exit angle becomes gradually increased from said central portion to the outer end.

5. A vortex flow blower comprising: a casing having an annular passage and inlet and outlet ports each of which is communicated with said annular passage; an impeller having a plurality of blades rotating opposite to said annular passage; and a motor for rotating said impeller, wherein said blade has a shape formed in such a manner that all of γ_c , γ_0 and γ_1 become smaller than 90° and the relationships of $\gamma_c < \gamma_0$ and $\gamma_c < \gamma_1$ are met, where γ_c , γ_0 and γ_1 represent the axial angles respectively at the central portion, at the intermediate portion between said central portion and the outer end and at the intermediate portion between said central portion and the inner end of said blade.

6. A vortex flow blower according to Claim 1, wherein the inlet angle β_1 in the circumferential direction at the inner end and the axial angle γ_1 at the intermediate portion between said inner end and the central portion of said blade are set to meet the following ranges:

$$45^\circ \leq \beta_2 \leq 80^\circ \text{ and } 20^\circ \leq \gamma_1 \leq 70^\circ.$$

7. A vortex flow blower according to Claim 1, wherein the exit angle β_2 in the circumferential direction at the outer end and the axial angle γ_0 at the intermediate portion between said outer end and the central portion are set to meet the following ranges:

$$0^\circ \leq \beta_2 \leq 135^\circ \text{ and } 20^\circ \leq \gamma_0 \leq 70^\circ.$$

8. A vortex flow blower according to Claim 1, wherein said blade is formed so as to project in the direction of rotation of said impeller from the central portion to the outer portion.

9. A vortex flow blower according to Claim 2, wherein said blade is retracted with respect to the direction of rotation of said impeller from the central portion to the outer portion, thereby controlling the discharge pressure to a set value.

10. A vortex flow blower according to Claim 1, wherein the exit angle γ_2 at the outer end of said blade is larger than 90° , thereby controlling the discharge pressure to a set value.

11. A vortex flow blower according to Claim 8, wherein the central portion of said blade has an intermediate portion retracting with respect to the direction of rotation of said impeller.

12. A vortex flow blower according to Claim 1, wherein said impeller is positioned in said casing and has said blades on the both sides thereof.

13. A vortex flow blower comprising: a casing having an annular passage, inlet and outlet ports each of which is communicated with said annular passage and a partition wall partitioning the inner portion of said annular passage extending from said outlet port to said inlet port; an impeller having a plurality of blades rotating opposite to said annular passage; and a motor for rotating said impeller, wherein the shape of said blade is three dimensionally formed and the front end of said partition wall is formed such that it intersects the front edge of said blade from outer side at the portion adjacent to said inlet port and from inner side at the portion adjacent to said outlet port as said impeller rotates.

14. A vortex flow blower comprising: a casing having an annular passage and inlet and outlet ports each of which is communicated with said annular passage; an impeller having a plurality of blades rotating opposite to said annular passage; and a motor for rotating said impeller, wherein the shape of said blade is three dimensionally formed and said impeller is positioned at the side adjacent to said motor while said casing is positioned at the side remote from said motor.

15. A vortex flow blower comprising: a casing having an annular passage and inlet and outlet ports each of which is communicated with said annular passage; a rotatable impeller having a plurality of blades for raising the pressure of fluid by giving a swirling flow between them and said an-

nular passage; and a motor for rotating said impeller, wherein at least the inner portion of said blade is formed so as to cause fluid to be introduced with more inclined at the central portion with respect to said impeller than at the inner end of said blade, and the ratio R_1/R_2 between the radius R_1 at the inner end and the radius R_2 at the outer end of said blade is set to the range between 0.75 or less and 0.3 or more.

16. An impeller used in a blower for discharging fluid through an outlet port by sucking the fluid into an annular passage formed in a casing through an inlet port by means of rotating an impeller having a plurality of blades to give a swirling flow between them and said annular passage so as to raise the pressure of said fluid, characterized in that it comprises said blades the shape of each of which is formed by a curved surface at least in its region from the inner end to the central portion thereof so as to make both γ_1 and γ_c smaller than 90 degrees and to meet the relationship of $\gamma_1 > \gamma_c$, where γ_1 and γ_c represent the axial inlet angles of said blade's front edge respectively at said inner end and at said central portion.

17. An impeller used in a blower for discharged fluid through an outlet port by sucking fluid into an annular passage formed in a casing through an inlet port by means of giving a swirling flow so as to raise the pressure of said fluid, characterized in that it comprises a plurality of blades the shape of each of which is formed by a curved surface formed in such a manner that the axial angle of said blade's front edge is smaller than 90 degrees at the inner end thereof, is gradually decreased from said inner end toward the central portion thereof and is increased from said central portion to the outer portion, thereof.

18. A method of manufacturing an impeller having a shroud and a plurality of blades provided in said shroud, comprising the steps of independently manufacturing said shroud and said blades and combining the thus independently manufactured shroud and blades so as to compose said impeller.

19. A method of manufacturing an impeller having a shroud and a plurality of blades provided in said shroud, comprising the steps of independently manufacturing said shroud and said blades, forming insertion holes in said shroud, forming caulking projections on said blade, fitting said caulking projections into said insertion holes and plastically working the thus fitted portion so as to compose said impeller.

20. A method of manufacturing an impeller having a shroud and a plurality of blades provided in said shroud, comprising the steps of independently manufacturing said shroud and said blades, forming an insertion groove in said shroud and

press-fitting said blade into said insertion groove so as to compose said impeller.

21. A method of manufacturing an impeller having a shroud and a plurality of blades provided in said shroud, comprising the steps of positioning cores for forming said blades in an outer mold at a predetermined interval, pouring fluid between said cores and said outer mold and solidifying said fluid so as to compose said impeller.

22. A method of manufacturing an impeller having a shroud and a plurality of blades provided in said shroud, comprising the steps of manufacturing impeller component units each of which constitutes a part of said shroud between neighboring blades and has monolithically a pair of said blade and assembling said impeller component units so as to compose said impeller.

23. A method of manufacturing an impeller according to Claim 21, wherein said core is divided into a plurality of sections.

24. A method of manufacturing an impeller according to Claim 18, wherein a filler or an adhesive is filled in the corner portion between said shroud and said blade.

FIG. 1

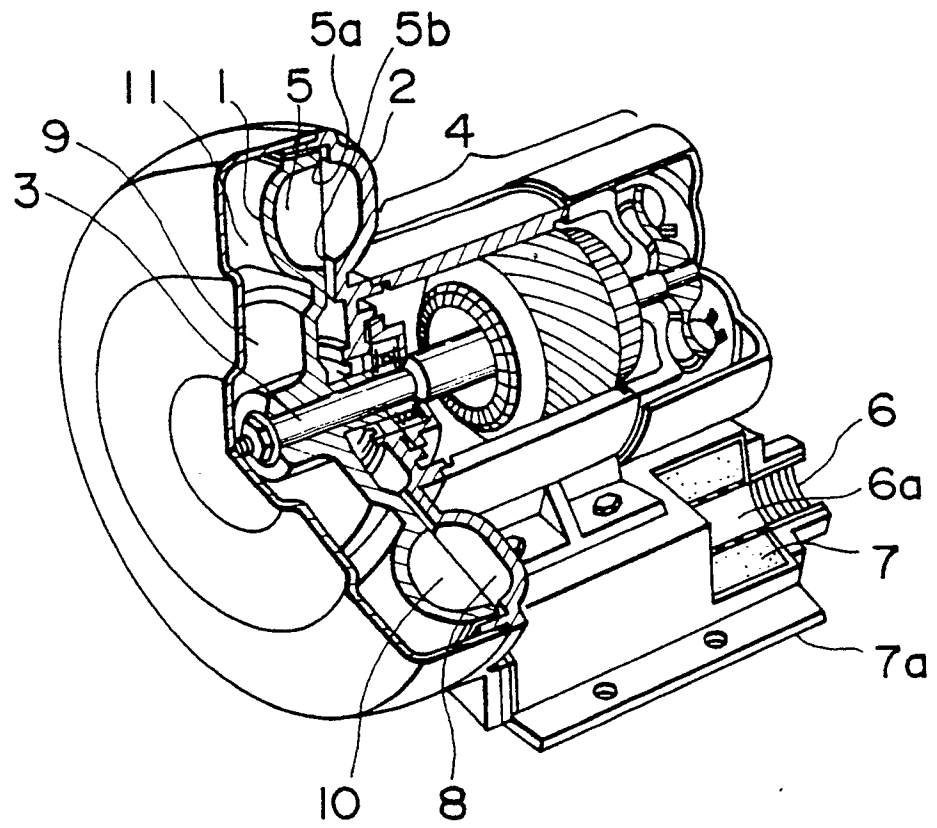


FIG. 2

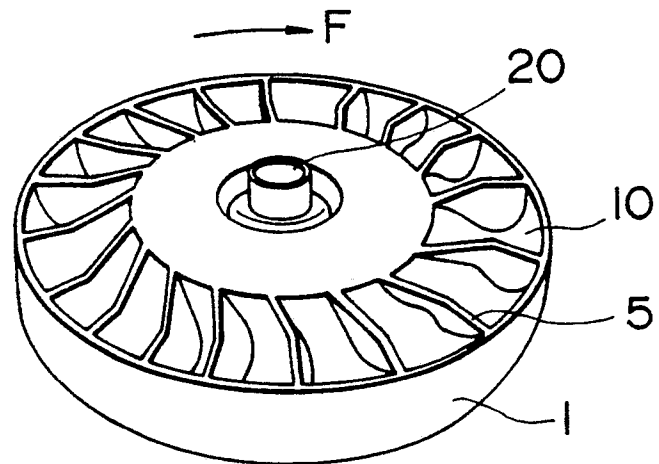
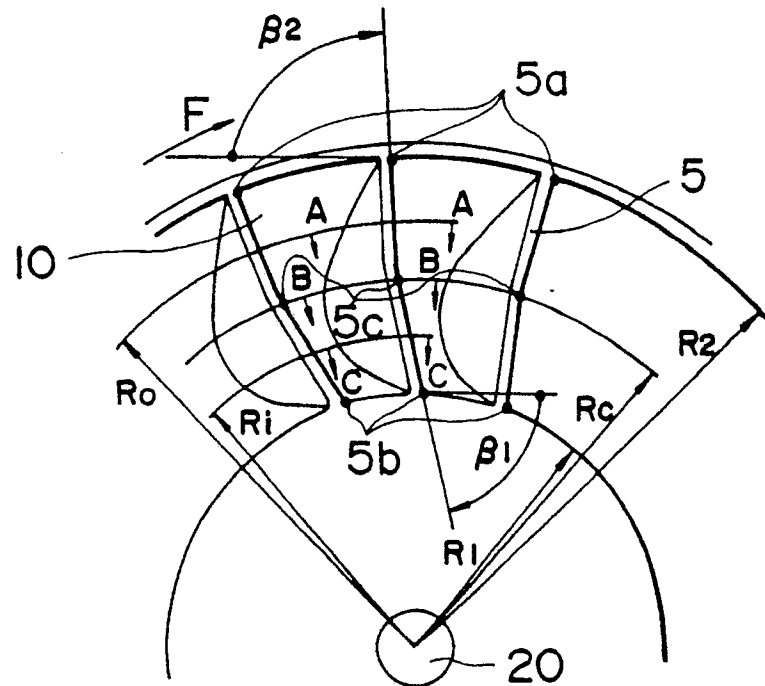


FIG. 3



RADIUS AT MIDPOINT
RADIUS OF OUTER
CROSS SECTION
RADIUS OF INNER
CROSS SECTION

$$R_c = (R_1 + R_2) / 2$$

$$R_o = (R_2 + R_c) / 2$$

$$R_i = (R_c + R_1) / 2$$

FIG. 4

SECTION
A - A

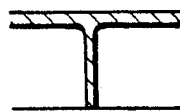


FIG. 5

SECTION
B - B

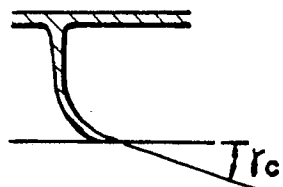


FIG. 6

SECTION
C - C

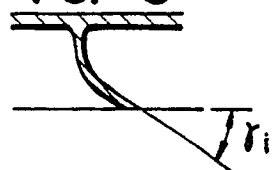


FIG. 7

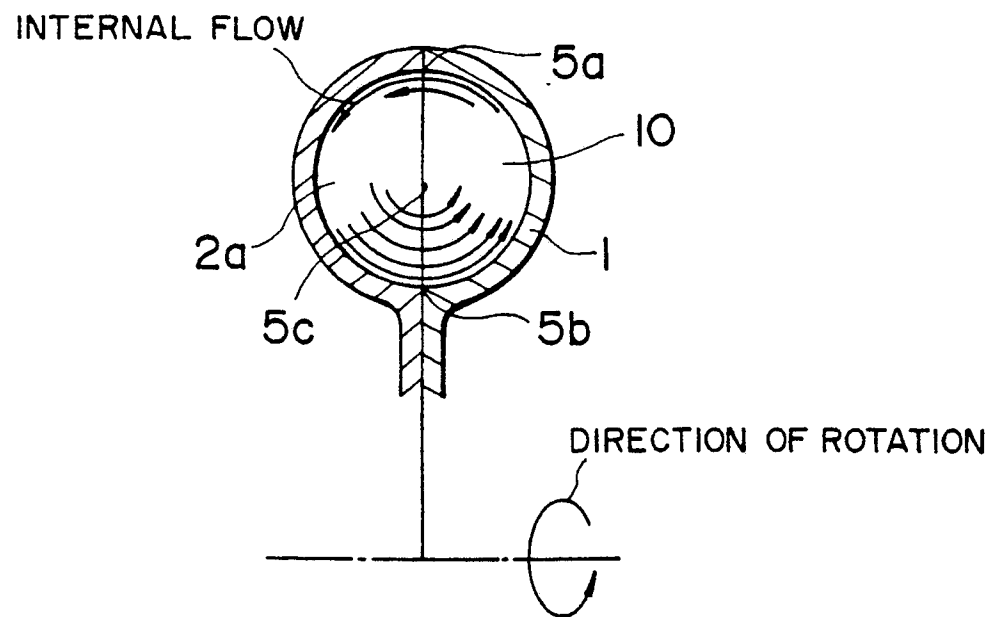


FIG. 8

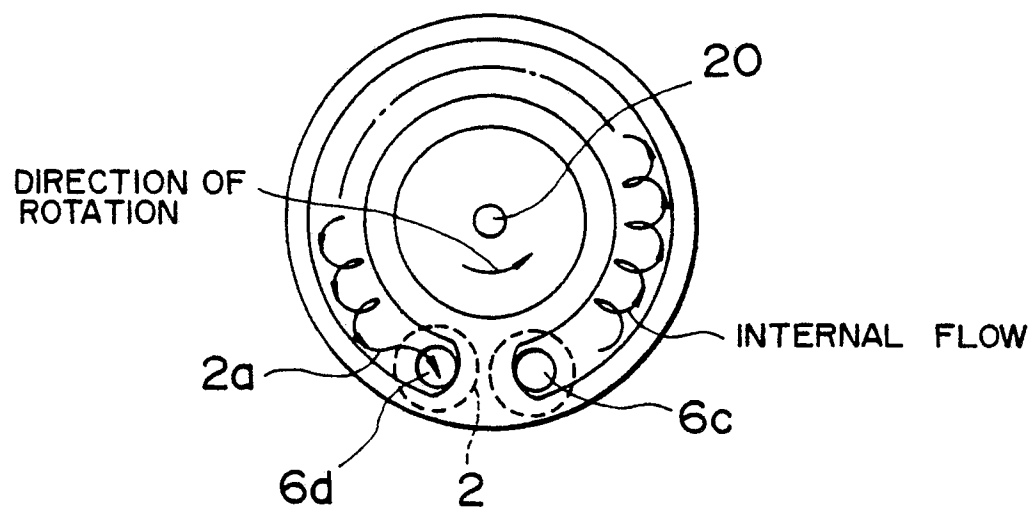


FIG. 9

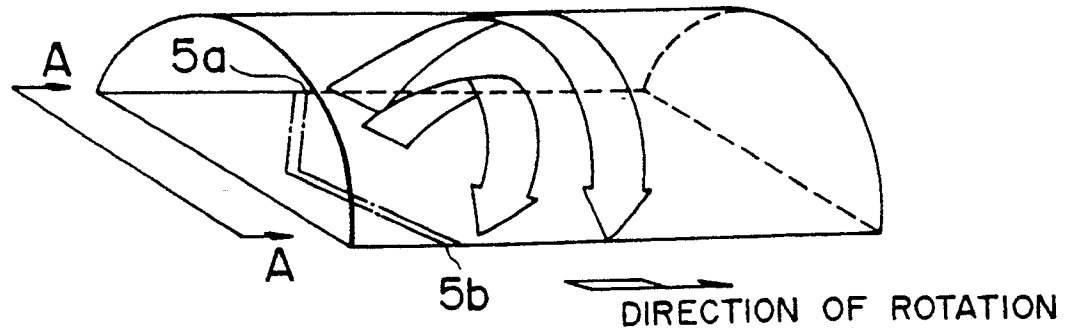


FIG. 10

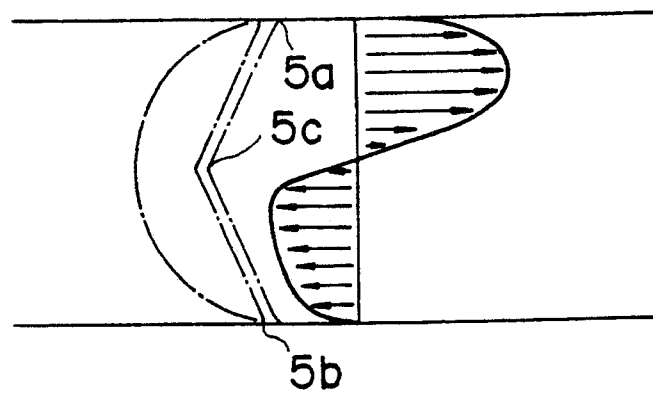


FIG. 11

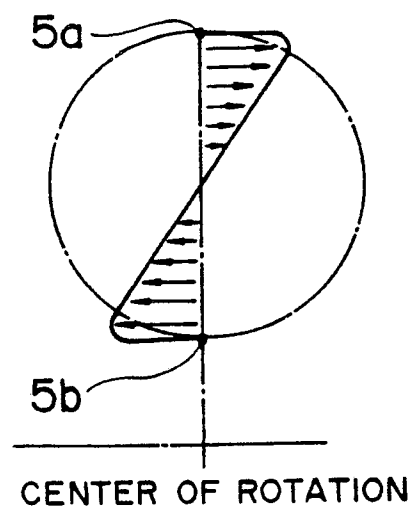


FIG. 12

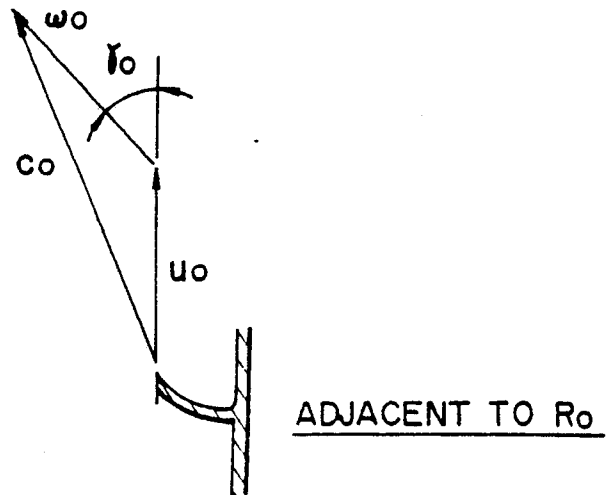


FIG. 13

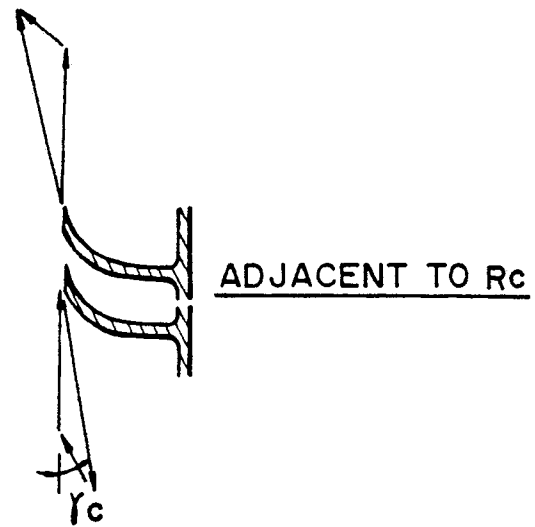


FIG. 14

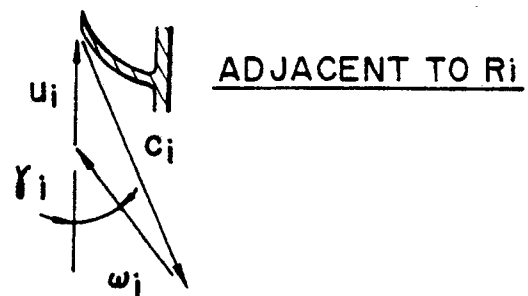


FIG. 15

PRESSURE
COEFFICIENT
RATIO ψ/ψ_0 DISTRIBUTION
($\beta_1=90^\circ, \gamma_1=90^\circ, \beta_2=90^\circ$)

$\beta_1 \backslash \gamma_1$	10°	20°	45°	70°	80°	90°
100°						0.8
90°	0.6	1.4	1.7	1.3	1.1	1.0
80°	1.0	1.8	2.0	1.8	1.6	1.1
60°		2.0	2.5	2.2	1.9	1.1
45°		1.8	2.0	2.0	1.6	1.3
20°			1.4			1.4

FIG. 16

PRESSURE
COEFFICIENT
RATIO ψ/ψ_0 DISTRIBUTION
($\beta_1=90^\circ, \gamma_1=90^\circ, \beta_2=90^\circ$)

$\beta_1 \backslash \gamma_1$	10°	20°	45°	70°	80°	90°
100°						0.7
90°	0.5	0.8	1.1	1.0	0.8	0.7
80°	0.6	1.3	1.3	1.4	1.2	0.8
60°		1.4	1.3	1.3	1.3	1.0
45°		1.3	1.3	1.3	1.0	1.0
20°		1.0				

FIG. 17

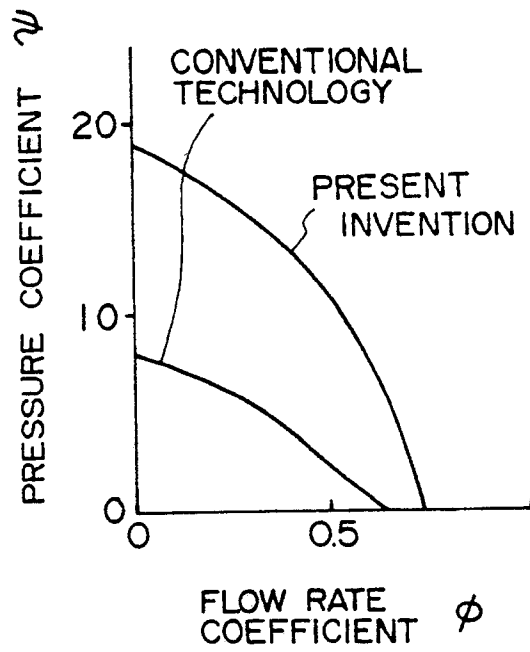


FIG. 18

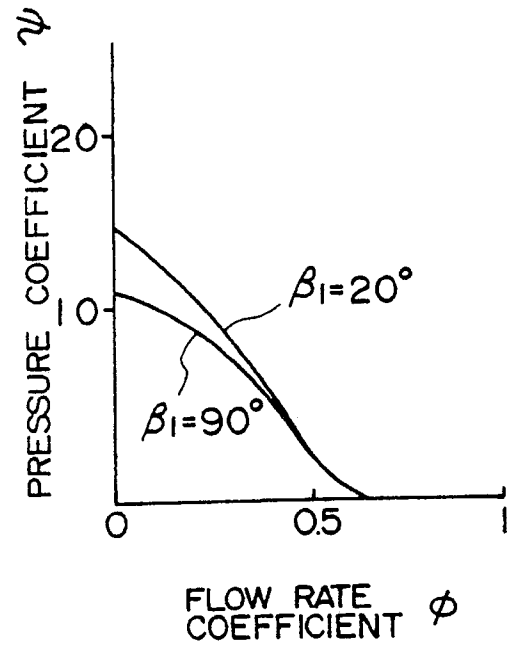


FIG. 19

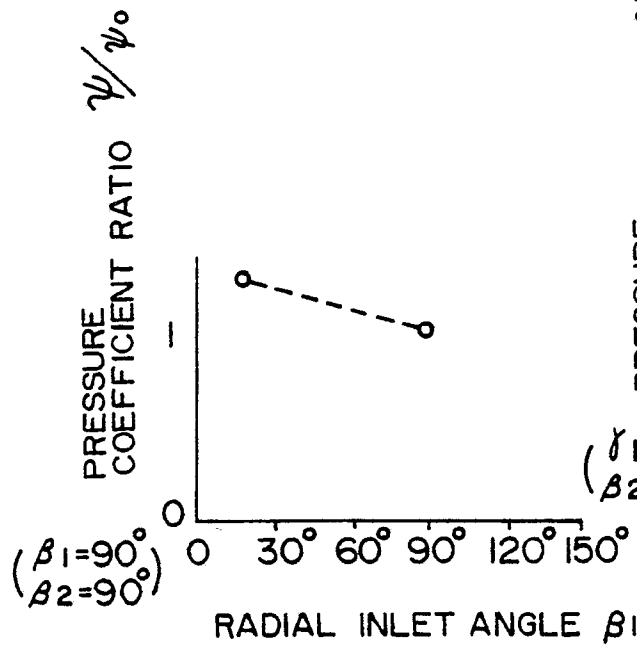


FIG. 20

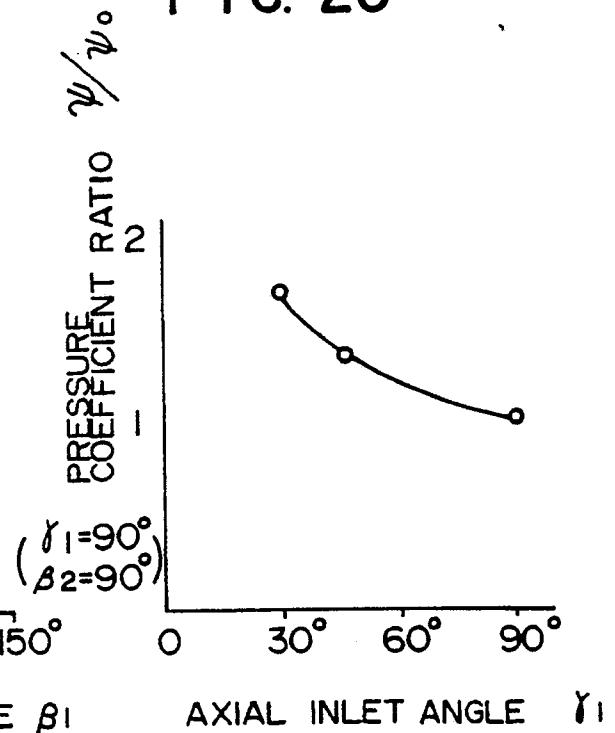


FIG. 21

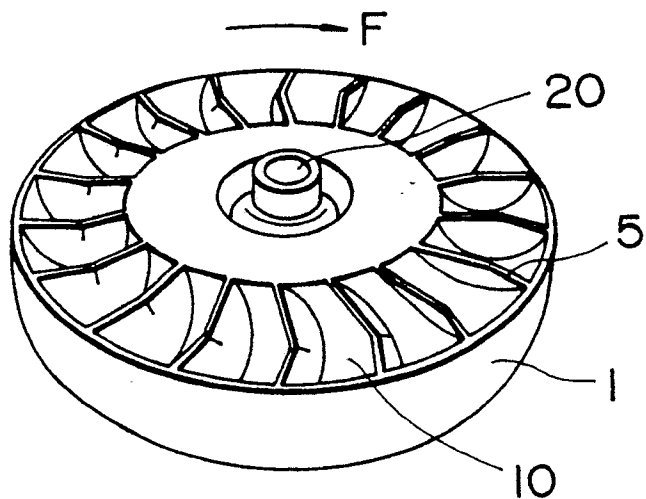


FIG. 22

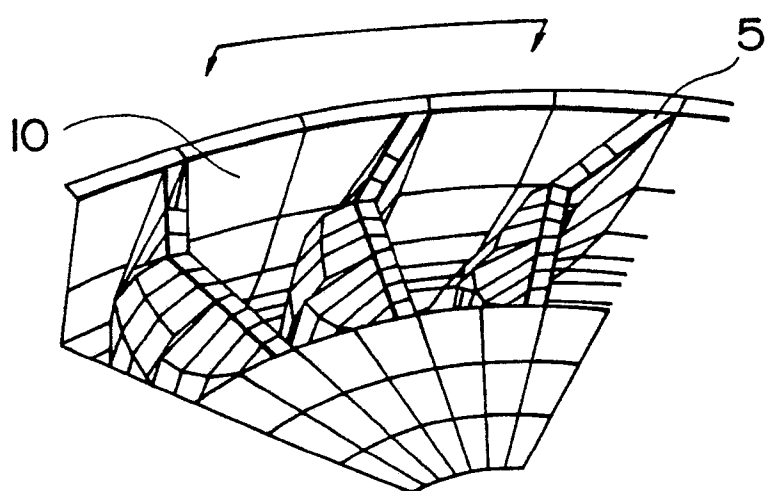
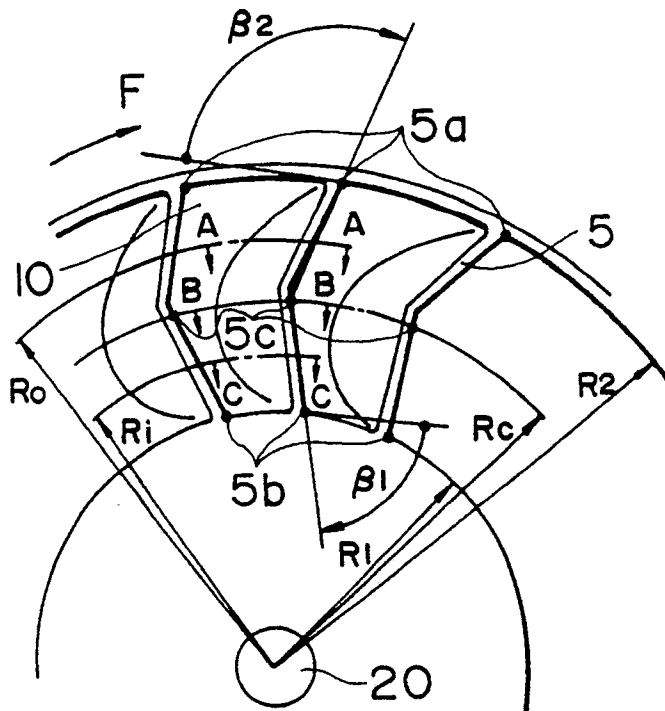


FIG. 23



RADIUS AT MIDPOINT $R_c = (R_i + R_2) / 2$
 RADIUS OF OUTER CROSS SECTION $R_o = (R_2 + R_c) / 2$
 RADIUS OF INNER CROSS SECTION $R_i = (R_c + R_1) / 2$

FIG. 24

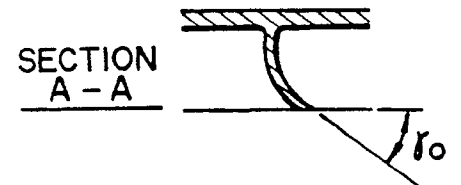


FIG. 25

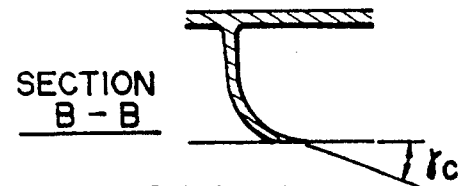


FIG. 26

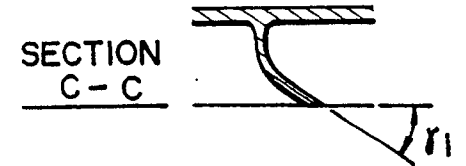
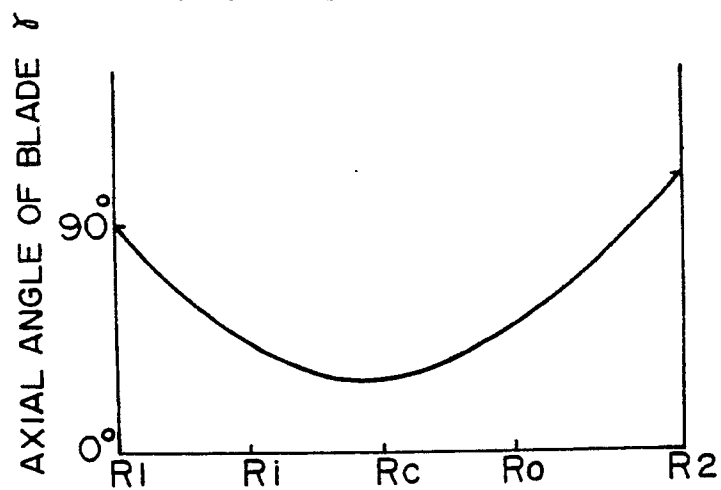


FIG. 27



$$r_c < r_o$$

$$r_c < r_i$$

FIG. 28

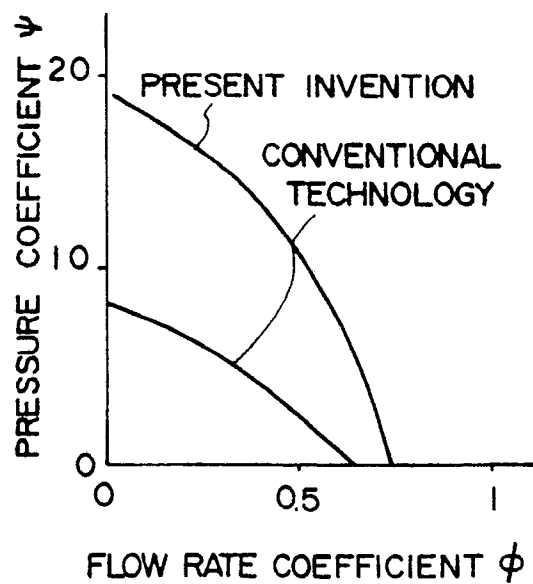


FIG. 29

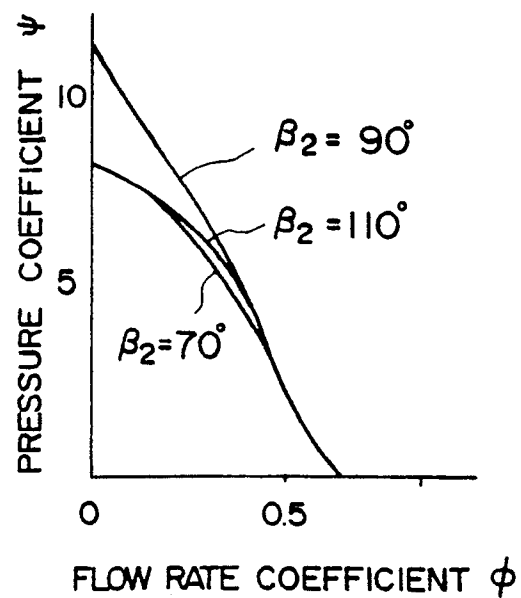


FIG. 30

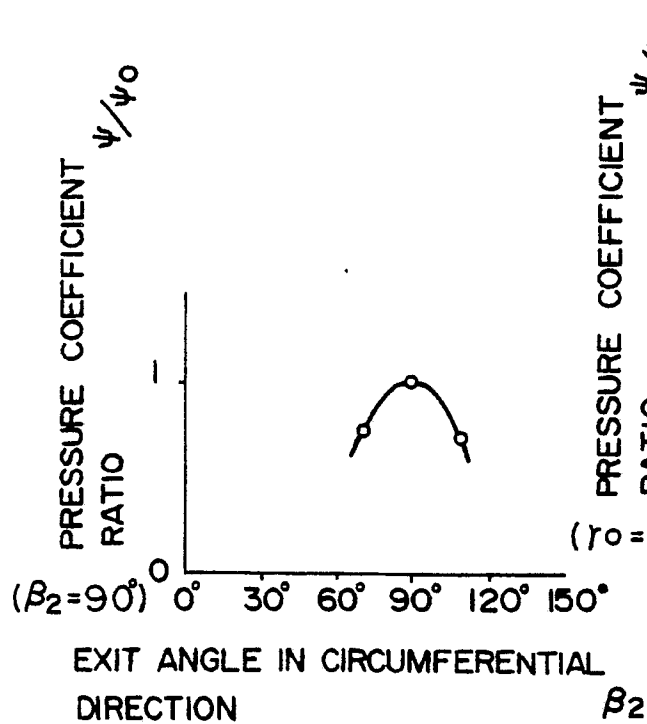


FIG. 31

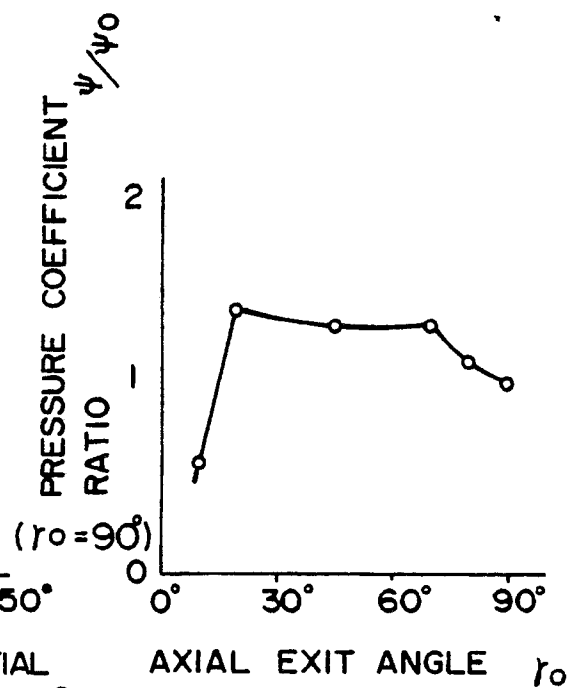


FIG. 32

PRESSURE COEFFICIENT
RATIO ψ/ψ_0 DISTRIBUTION
($\beta_2 = 90^\circ$, $\gamma_0 = 90^\circ$)

$\beta_2 \backslash \gamma_0$	10°	20°	45°	70°	80°	90°
80°						0.8
90°	0.6	1.4	1.3	1.3	1.1	1.0
100°	1.0	1.8	1.9	1.9	1.6	0.8
115°		2.0	2.4	2.3	1.8	0.7
135°		1.9	2.1	2.0	1.6	0.6

FIG. 33

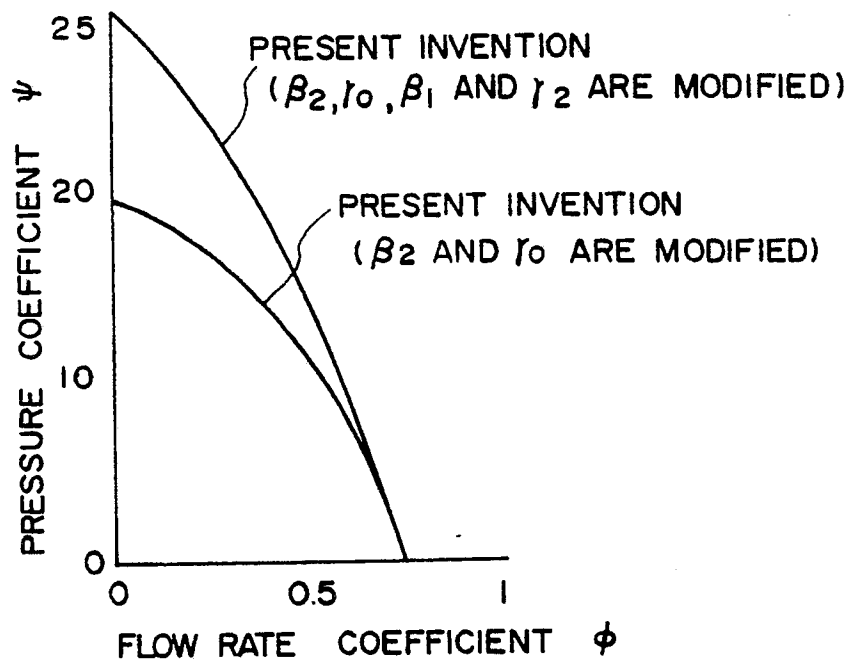
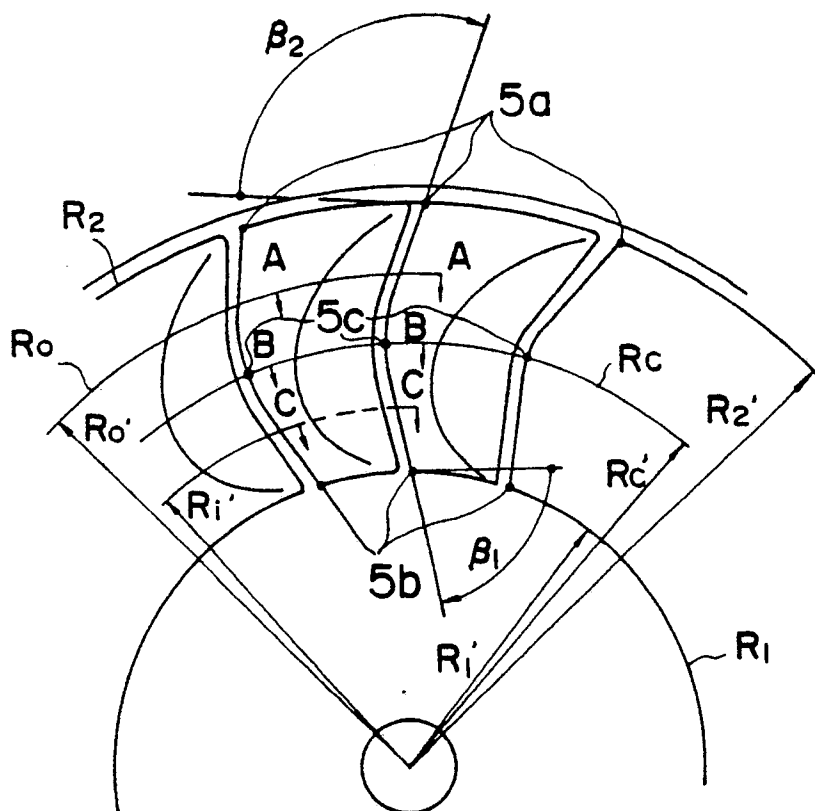


FIG. 34



RADIUS AT MIDPOINT

$$R_c = (R_1 + R_2)/2$$

RADIUS OF OUTER CROSS
SECTION

$$R_0 = (R_2 + R_c)/2$$

RADIUS OF INNER CROSS
SECTION

$$R_1 = (R_c + R_1)/2$$

FIG. 35

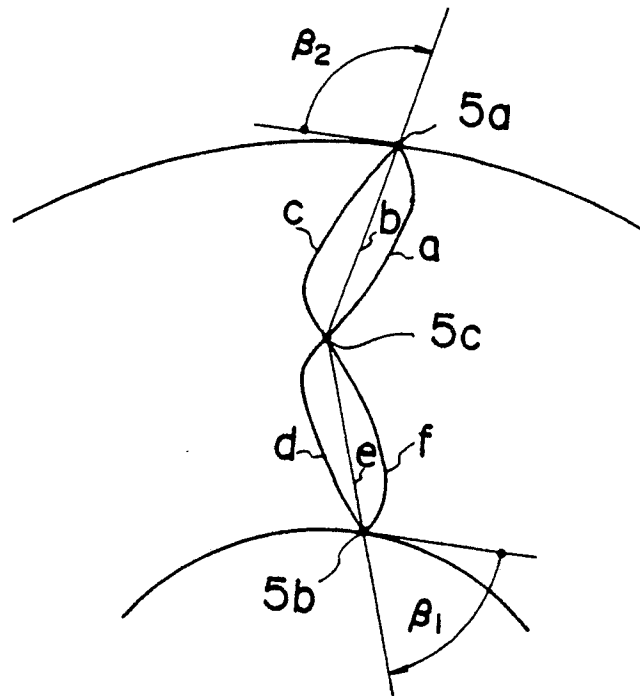


FIG. 36

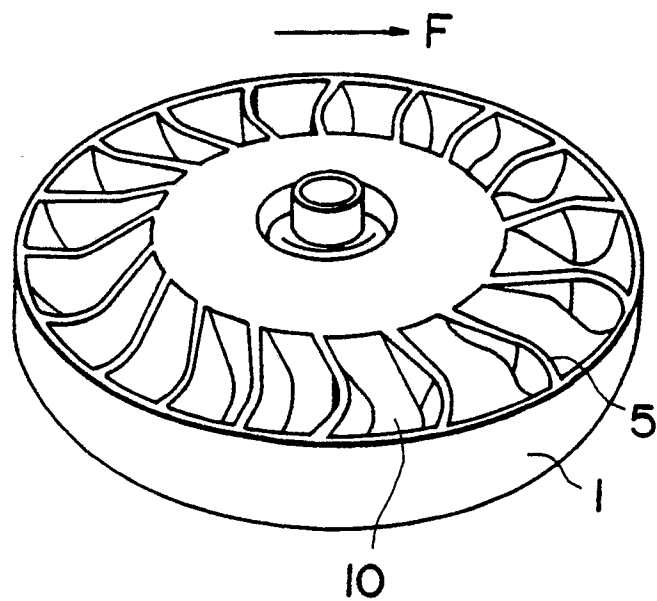


FIG. 37

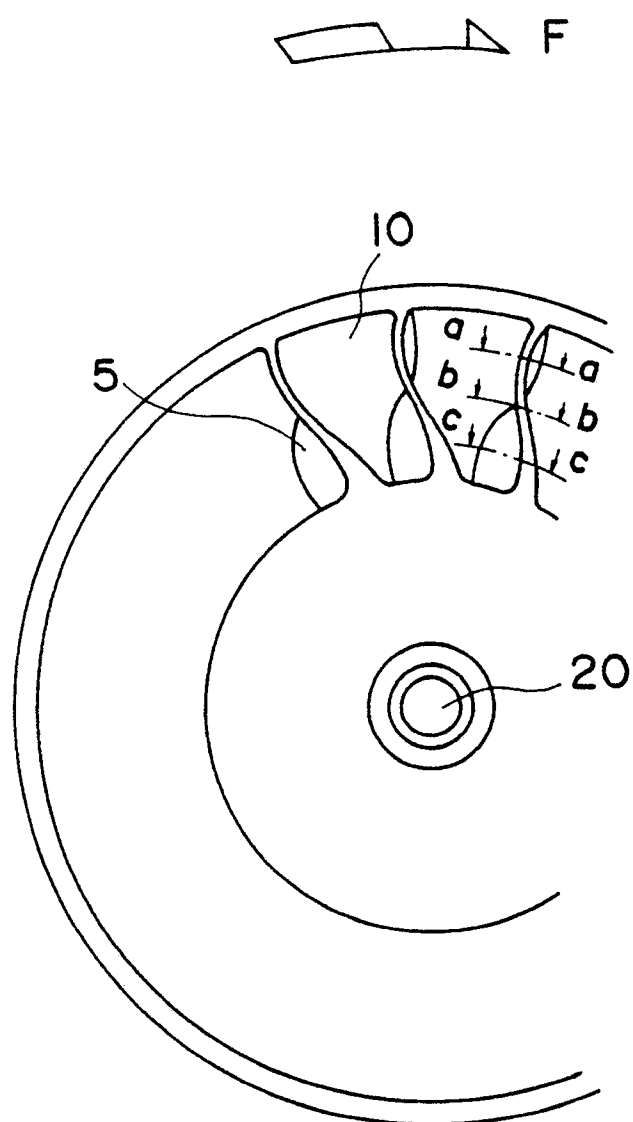
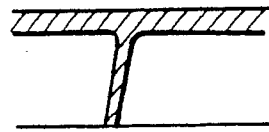
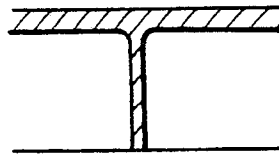


FIG. 38



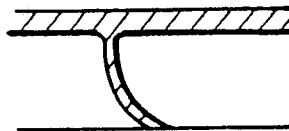
SECTION a - a

FIG. 39



SECTION b - b

FIG. 40



SECTION c - c

FIG. 41

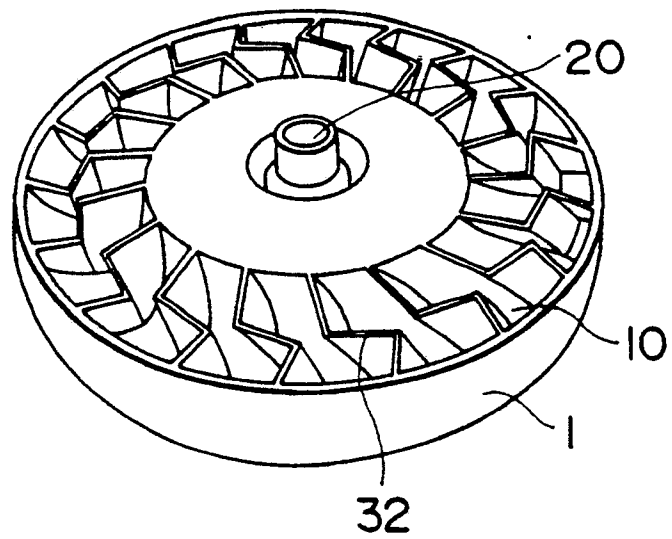


FIG. 42

R_o RADIUS
SECTION

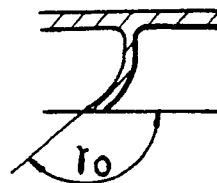


FIG. 43

R_c RADIUS
SECTION

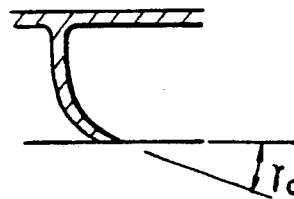


FIG. 44

R_i RADIUS
SECTION

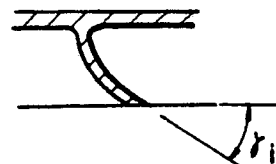


FIG. 45

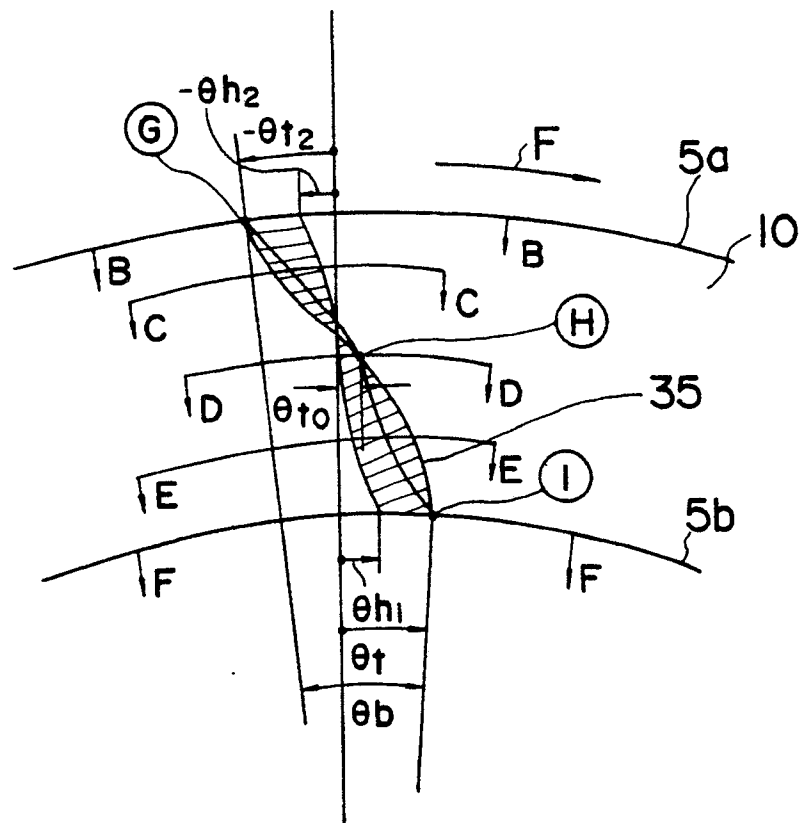


FIG. 46

DIRECTION OF ROTATION

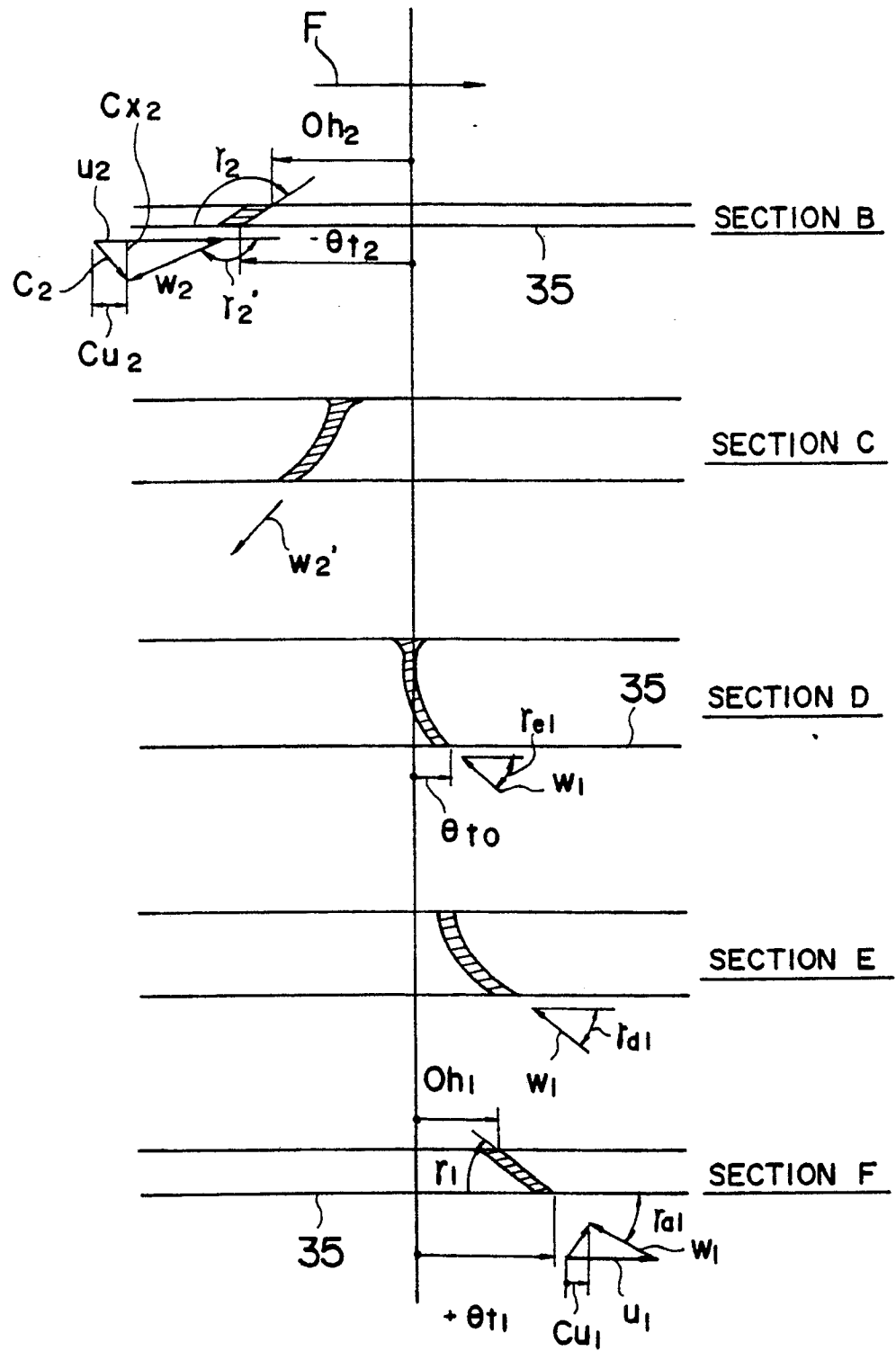


FIG. 48

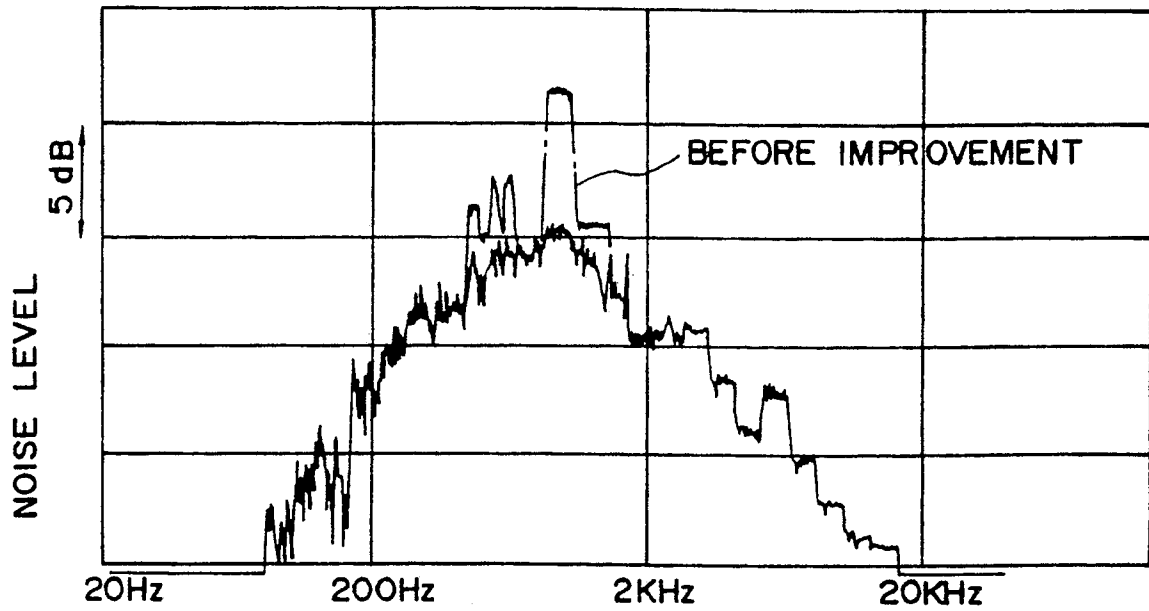


FIG. 49

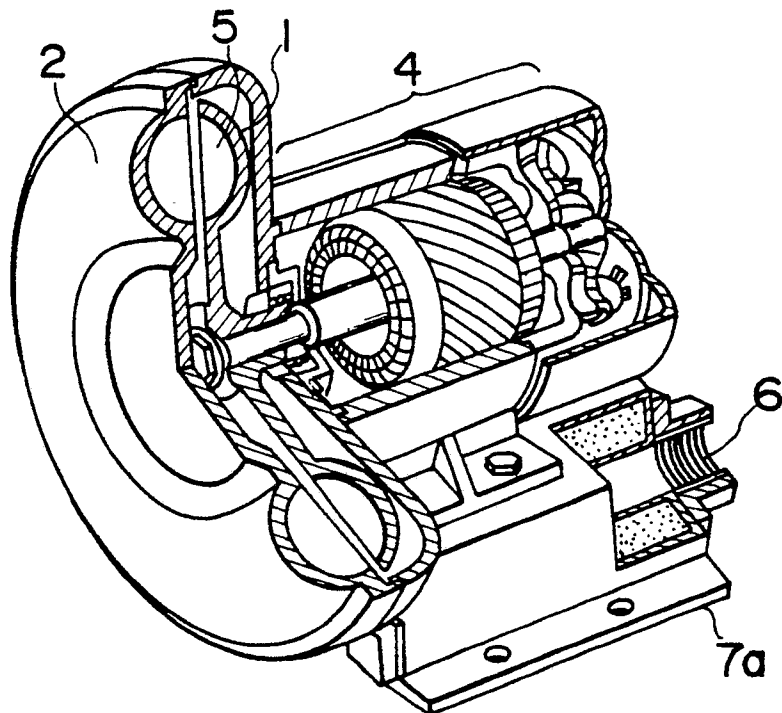


FIG. 55

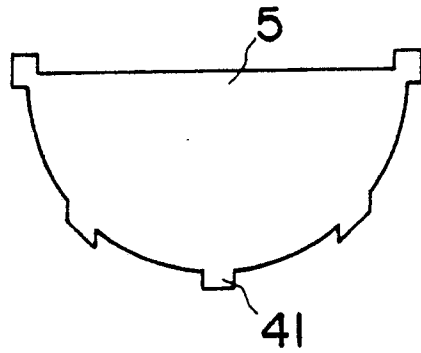


FIG. 56

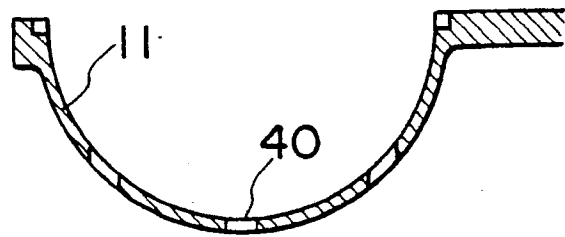


FIG. 57

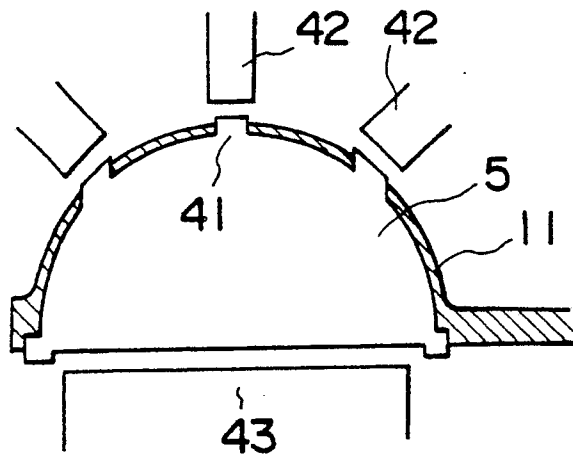


FIG. 58

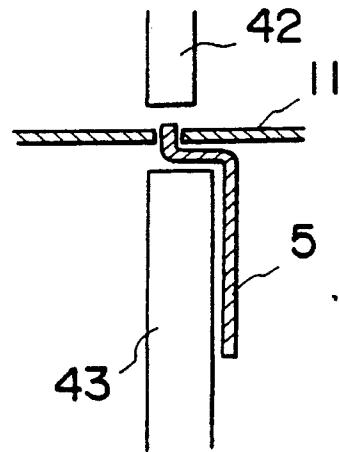


FIG. 59

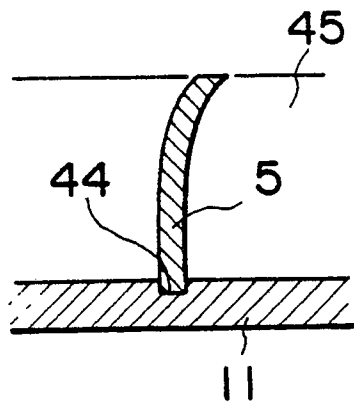


FIG. 60

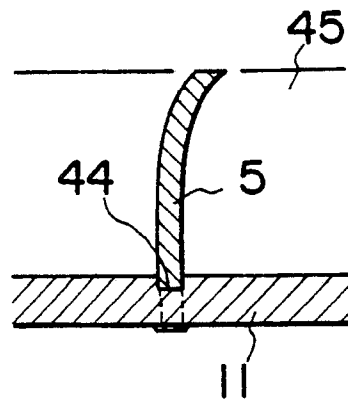


FIG. 50

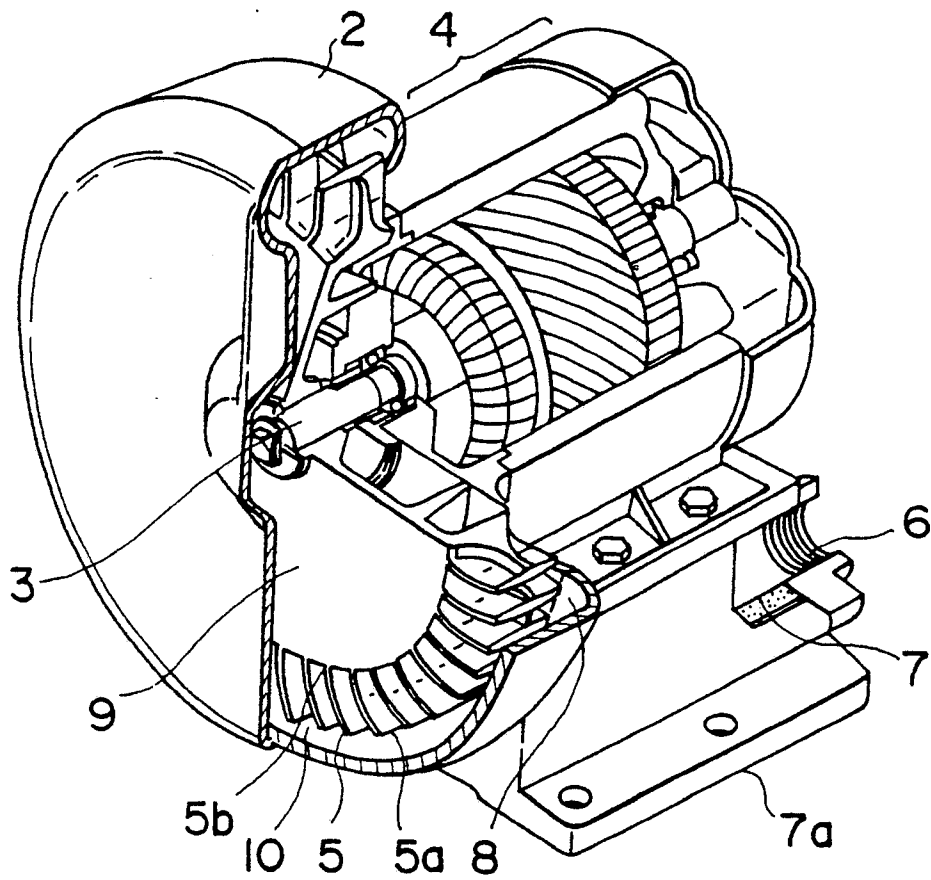


FIG. 51

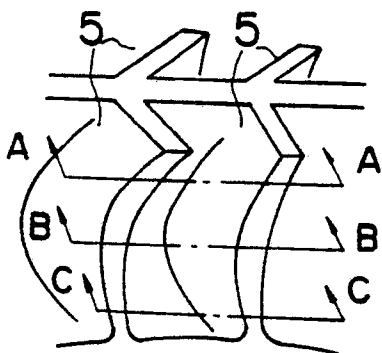
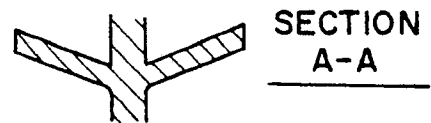
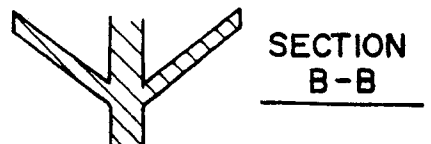


FIG. 52



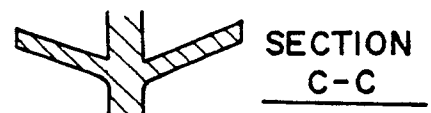
SECTION
A-A

FIG. 53



SECTION
B-B

FIG. 54



SECTION
C-C

FIG. 61

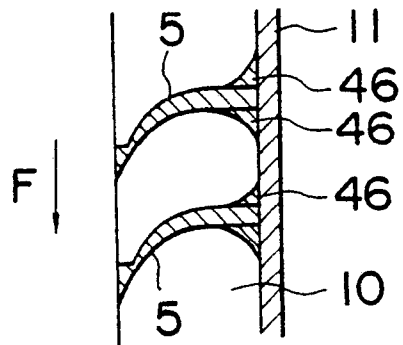


FIG. 62

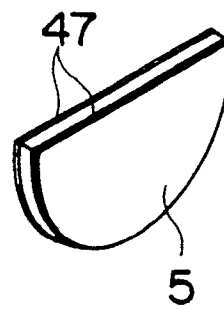


FIG. 63

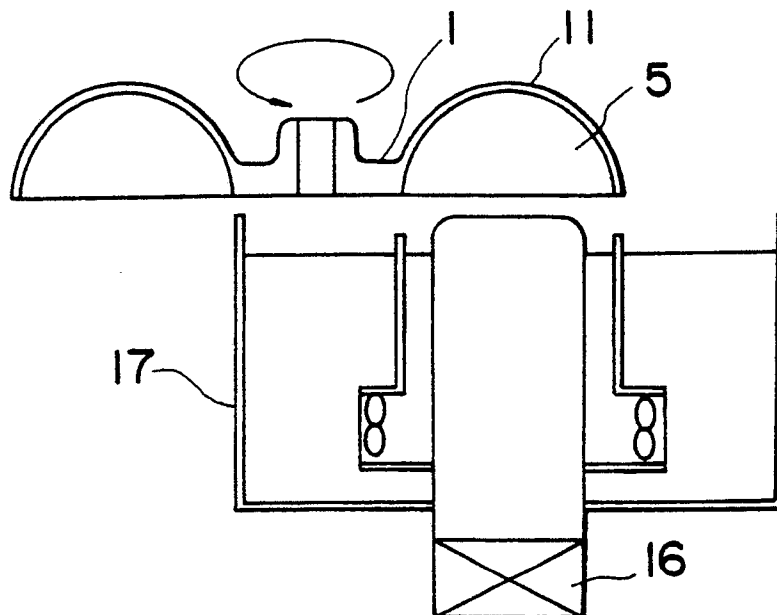


FIG. 64

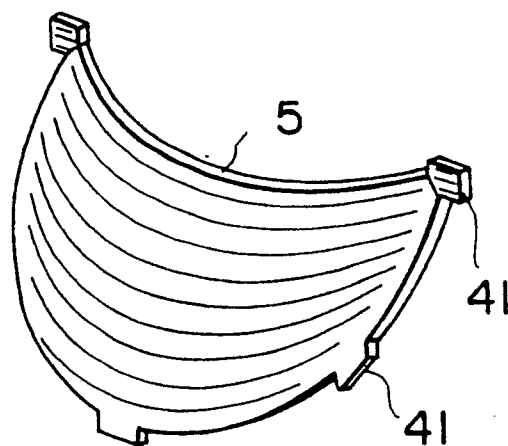


FIG. 65

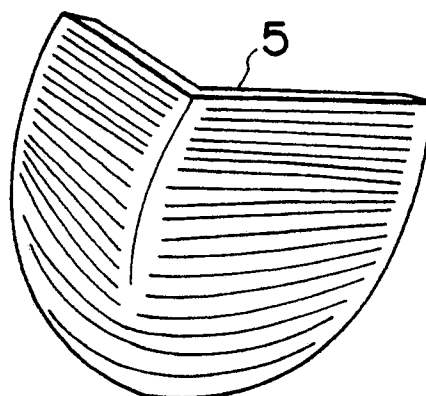


FIG. 66

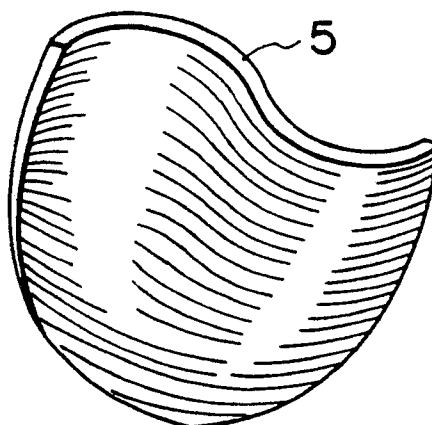


FIG. 67

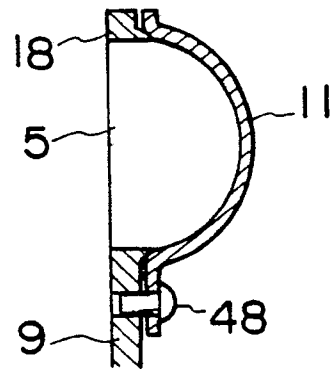


FIG. 68

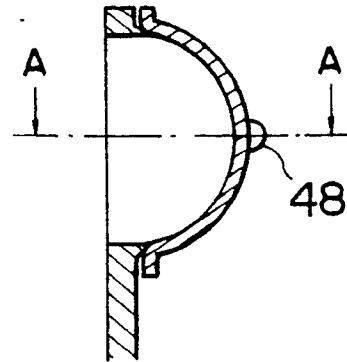


FIG. 69

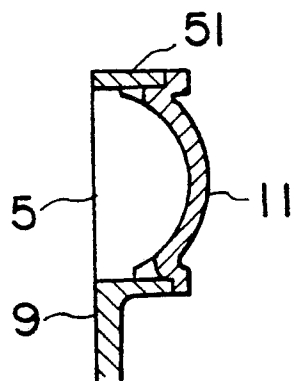


FIG. 70

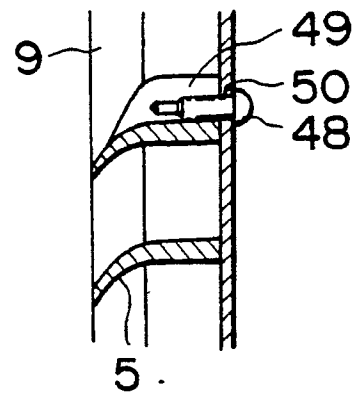


FIG. 71

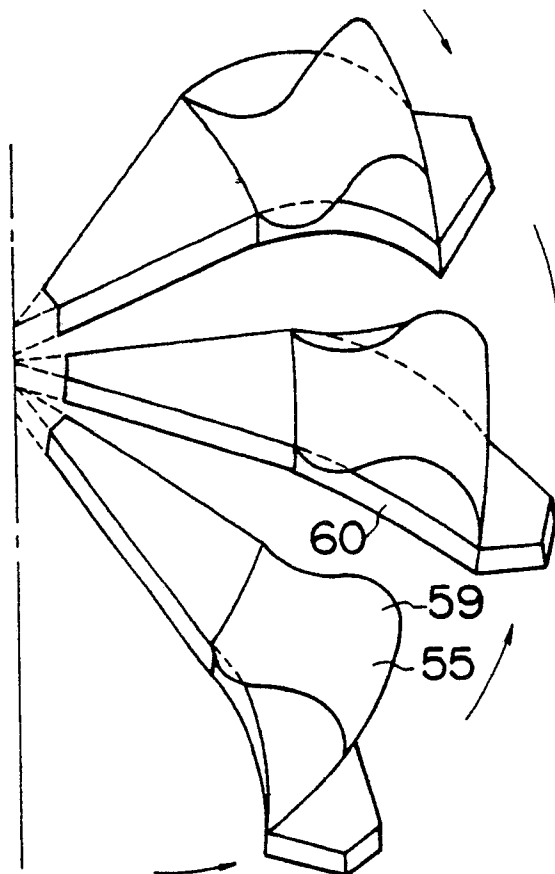


FIG. 72

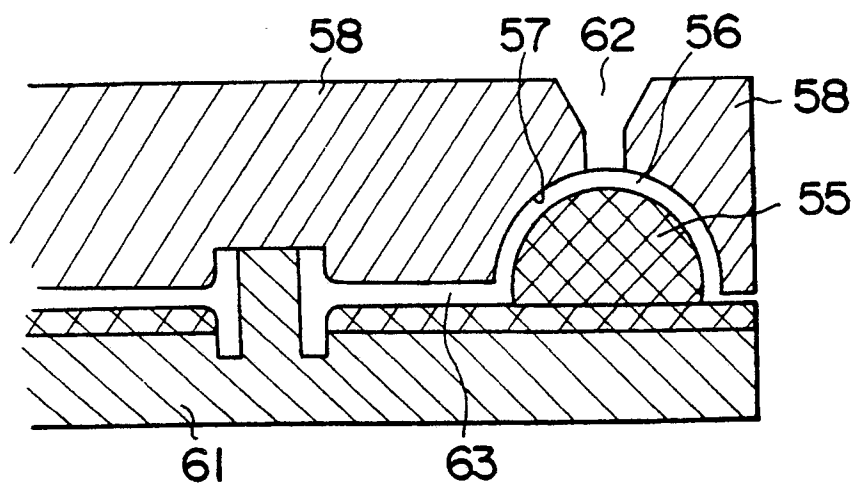


FIG. 73

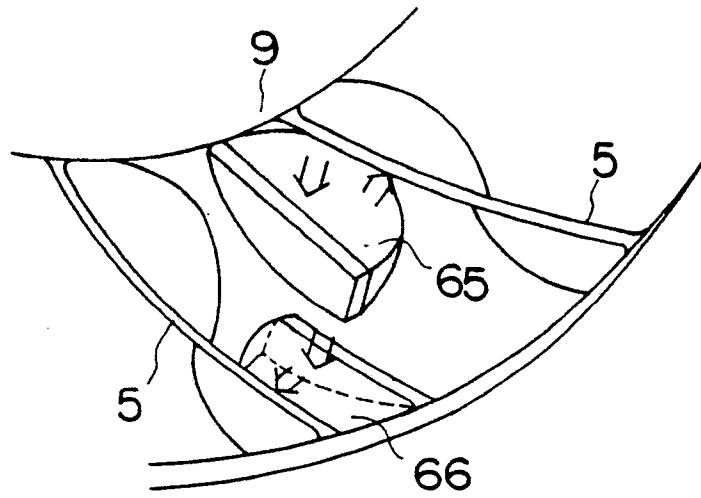


FIG. 74

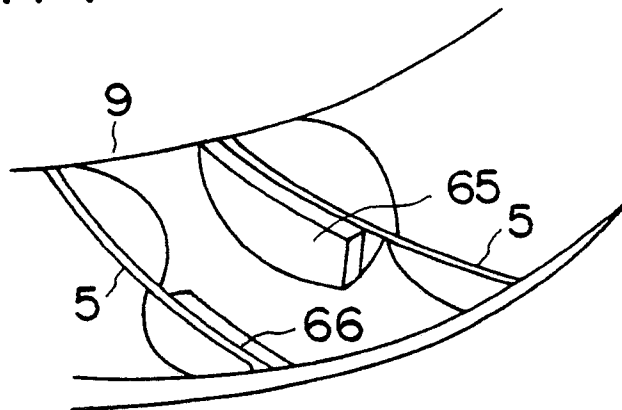


FIG. 75

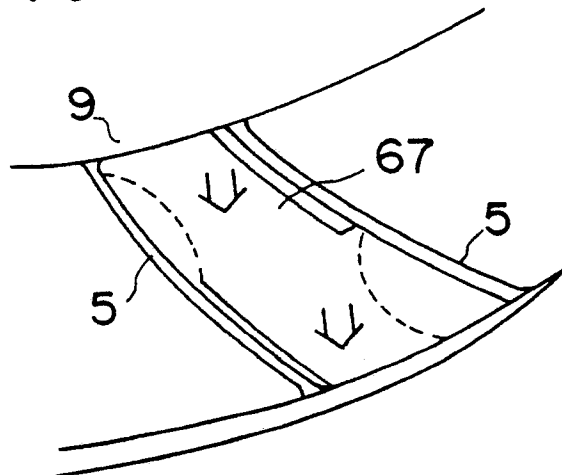


FIG. 76

