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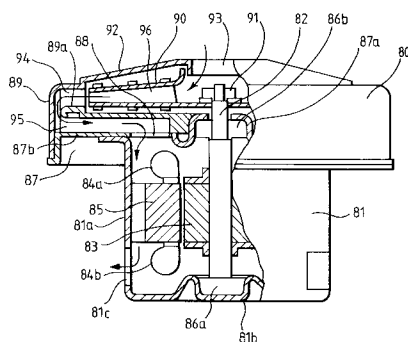
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York House
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(57) A vacuum cleaner blower assembly has a blower motor (81), a vaned centrifugal impeller (90) driven by the blower motor (81) and an air diffuser (89) radially beyond the periphery of the impeller. In order to improve air flow efficiency at the impeller inlet and outlet, the air diffuser (89) has vanes (94) with an inlet angle in the range 1 to 4°.

*Fig. 1.***EP 0 602 007 A2**

This invention relates to vacuum cleaners and particularly to impellers and diffusers in vacuum cleaners.

In conventional electric blowers of household vacuum cleaners such as shown in JP-A-59-74396, the configurations of the diffuser vane, return guide vane, etc. of the centrifugal impeller are analogous to those of a large-size blower or compressor, but such components are limited in size and shape in the case of an electric blower used in the vacuum cleaner. In general in centrifugal blowers or compressors, the angle formed between the flow coming out of the impeller and the circumferential direction is of the order of 10 to 30°, and the inlet angle of the diffuser vane is designed correspondingly. However, the specific speed of the electric blower for use in the vacuum cleaner is low (a small flow rate is provided in spite of a high pressure with respect to a relative rotational speed) and generally, the outlet width of the impeller is designed to be small; therefore, since the friction loss within the impeller becomes large as the outlet width of the impeller is decreased, the width and outlet angle of the vanes are made comparatively large. Accordingly, in the electric blower for use in a household vacuum cleaner, the outlet absolute flow angle of the impeller is designed to be about 6°, and the inlet angle of the diffuser is set to as large as 5° in practice.

The object of the present invention is at least partly to avoid the disadvantages described above, and to improve the air flow efficiency through the blower of a vacuum cleaner.

The present invention is set out in claim 1.

Embodiments of the invention will now be described by way of non-limitative example with reference to the accompanying drawings, in which:-

Fig. 1 is a side view, partly in cross section, of an electric blower including motor and blower, embodying the present invention;

Fig. 2 is an axial sectional view of part of the blower of Fig. 1;

Fig. 3 is an axial sectional view showing another embodiment of the blower according to the present invention;

Fig. 4 is an axial view of the impeller and diffuser of the electric blower shown in Fig. 1; and

Fig. 5 is an enlargement of the circled part of Fig. 4 showing the diffuser vanes;

Fig. 6 is a graph showing the characteristic of electric blowers when the inlet angle of the diffuser is varied;

Fig. 7 is a graph showing the characteristic of electric blowers when the ratio of diffuser vane throat width to a diffuser inner diameter is varied;

Fig. 8 is a graph showing the characteristic of electric blowers when the ratio of the total area of the diffuser vane throats to the diffuser inlet area is varied;

Fig. 9 is an axial view showing diffuser vanes of yet another embodiment of the present invention; and

Fig. 10 is a graph showing the aerodynamic characteristic of the embodiment of Fig. 9.

Embodiments of impellers and vacuum cleaner blowers of the invention will now be described. They may be fitted in conventional vacuum cleaners. Examples of vacuum cleaners in which they may be mounted are shown in European Patent Applications 91303152.2 and 91303496.3.

The electric vacuum cleaner blower shown in Figs. 1 and 2 is composed of a blower portion 80 and a motor portion 81. Disposed inside a housing 81a of the motor portion 81 are a rotor 83 secured to a rotating shaft 82 and a stator 85 including coils 84a and 84b. The housing 81a has a bearing-retaining portion 81b formed at the centre of its end wall, and a bearing 86a for rotatably supporting one end of the rotating shaft 82 is disposed in the bearing-retaining portion 81b. The housing 81a also has exhaust ports 81c in its peripheral surface. The housing 81a has an end bracket 87 at the opposite end, and this end bracket 87 connects the blower portion 80 and the motor 81 together.

The end bracket 87 has a bearing retaining portion 87a at its centre and a flat portion 87b around its circumference. The flat portion 87b is formed with suction ports 88 through which the air from the blower 80 is sent into the motor 81 to cool it. Disposed in the bearing-retaining portion 87a is a bearing 86b for rotatably supporting the other end of the rotating shaft 82. The end bracket 87 carries a diffuser 89, and on the upstream side of the diffuser, a centrifugal impeller 90 is secured to the rotating shaft 82 by means of a nut 91. The centrifugal impeller 90 and the diffuser 89 are covered by a blower casing 92 pressure-fitted to the circumference of the end bracket 87. The blower casing 92 has a suction port 93 formed in its central portion to provide an inlet to the central inlet region of the impeller.

The diffuser 89 is composed of a plurality of diffuser vanes 94 arranged radially outside the circumference of the centrifugal impeller 90. A plurality of return guide vanes 95 are arranged on the back of a wall 89a lying adjacent the impeller 90 and supporting the diffuser vanes 94. The wall 89a has a rounded outer peripheral edge to smooth the air flow from the diffuser vanes 94 to the return guide vanes 95, and in conjunction with the wall 89a and the end bracket 87, the return guide vanes 95 define a return

guide passage through which the air flow is guided to the suction ports 88.

The general operation of the electric blower in the embodiment will now be described. When the motor 81 is energized so that the impeller 90 is rotated, air flows as indicated by the arrows in the drawing, through the suction port 93 and into the impeller 90. After discharge from the impeller 90, the air passes
 5 between the diffuser vanes 94, and after passing through the return guide passage, goes through the suction ports 88 into the housing 81a. The air flow introduced into the housing 81a cools the rotor 83, passes through an air passage defined by the stator 85 and the inner surface of the housing 81a, cools the coils 84a and 84b, and goes through the exhaust ports 81c formed in the periphery of the housing 81a to the outside.

Fig. 2 shows the configuration of the centrifugal impeller 90 and the diffuser region in more detail. The impeller 90 is composed of a plurality of vanes 96, a shroud plate 97 and hub plate 98. Each vane 96 has on each edge three protrusions which are fitted in holes formed in the shroud plate 97 and the hub plate 98 and then caulked or upset, so that these components are rigidly and tightly secured together at these connection points. As Fig. 6 shows, the vanes 96 are curved as they extend outwardly, but for convenience
 15 this is not indicated in Fig. 2.

For further description of the shroud plate 97 and its shape, reference may be made to EP-A-467557 which is the publication of the application from which the present application is divided.

Another embodiment of the present invention will be described with reference to Fig. 3 showing a blower in partial sectional view. The shroud plate 97 is straight in its outer diameter portion, as viewed in
 20 the axial plane, and has a rounded portion 97a inwardly from the innermost point of connection 99, as in Fig. 2. The shroud plate 97 in this case is provided with a cylindrical portion 97b extending axially from the end of the rounded portion 97a. Furthermore, the blower casing 101 has an inwardly bent flange 101a at its inner diameter region, so that the gap 100 is left between the flange 101a and the cylindrical portion 97b of the impeller 90. Since the length of the gap 100 is much larger than the thickness of the shroud plate 97,
 25 the friction loss of the leak flow can be made very large, the leak flow can be reduced remarkably, and the efficiency of the electric blower can be improved.

Figs. 4 and 5 show the diffuser 89 of Figs. 1 and 2 with its vanes 94, as viewed from the suction port 93 of the electric blower. In this embodiment there are seventeen diffuser vanes 94 and eight return guide vanes 95. The inlet angle β_3 of the diffuser vane 94 as shown in Fig. 5 is 3° . The inlet angle β_3 is the angle
 30 between the inner face of the vane at its leading edge and the tangential line at this point. The throat width w_s is 2.2 mm, and its ratio to the inner diameter of the diffuser is 0.02. The radius of the rounded leading edge of the vane 94 is 0.5 mm. The air flow coming out of the impeller 90 is decelerated in a semi-vaneless space of the vaned diffuser 89 and further decelerated in each passage defined between two vanes 94. In the foregoing embodiment, the air discharge velocity of the blower can be made large, particularly about 0.8
 35 times the peripheral speed of the impeller. Accordingly, the size of the impeller can be reduced.

Fig. 6 shows the relative efficiency of an electric blower including the impeller according to the embodiment of Figs. 3 to 5, relative to a varying diffuser inlet angle β_3 . The efficiency under the condition that the diffuser inlet angle β_3 is 5° was taken as a reference. Where the diffuser inlet angle β_3 is smaller than 2° , the length of the semi-vaneless space is longer, the friction loss increases, and the efficiency
 40 decreases. Where the diffuser inlet angle β_3 is larger than 3° , it tends to come out of the flow angle from the impeller; thus, the performance degrades. As will be appreciated, where the diffuser inlet angle β_3 is within the range of 2 to 3° , the efficiency is about 2% greater than that in the prior art based on an angle of 5° , and even where the diffuser inlet angle is within the range of 1 to 2° or within the range of 3 to 4° , the efficiency is 1% greater.

Fig. 7 shows the efficiency of the same electric blower relative to a varying throat width w_s . Where the ratio of the throat width w_s to the diffuser inner diameter is smaller than 0.017, the deceleration is insufficient in the semi-open portion but increases in the passages defined between two vanes 94; thus, the flow breaks away in such a passage, thereby decreasing the efficiency. Where the ratio of the throat width
 45 w_s to the diffuser inner diameter is larger than 0.025, the deceleration becomes too significant in the semi-open portion; thus, the flow deviates remarkably as it flows into each passage defined between two vanes, thereby decreasing the efficiency. In the embodiment, where the ratio of the throat width w_s to the diffuser inner diameter is 0.02, the efficiency is high. In addition, since the flow angle of the air discharged from the impeller 90 is small and the air discharged from it travels a long distance until it enters the diffuser 89, the inlet diameter of the diffuser can be reduced as shown in Figs. 4 and 5, and the energy loss compared with
 50 a diffuser with no vanes can be reduced. Further, since the relative velocity at the outlet of the impeller can be decreased, noise can be reduced.

Fig. 8 shows the relative efficiency of this electric blower obtained when varying the ratio

$$\frac{\text{total area of diffuser vane throats}}{\text{real area of diffuser inlets}}$$

given by

$$\frac{Z_{vd} \cdot b_3 \cdot ws}{D_3 \cdot b_3 \cdot \sin \beta_3}$$

where

- Z_{vd} = number of diffuser vanes,
 b_3 = axial width of diffuser vanes,
 ws = diffuser vane throat width,
 D_3 = diffuser vane inlet diameter,
 β_3 = diffuser vane inlet angle.

When this ratio is smaller than 1.75, since the number of the diffuser vanes increases, the throat width decreases, surging occurs at a low flow rate, and pressure loss increases at a large flow rate, tending to narrow the serviceable range. When this ratio is larger than 3.5, the number of vanes of the diffuser 89 decreases, tending to cause interference with the number of blades of the impeller, so that a peak sound is generated, and the noise level is increased. When this ratio is 2.1 as in the actual embodiment, the efficiency is high.

Fig. 9 shows the diffuser 89 in another embodiment of the present invention. Each passage of the diffuser is defined by the vane portions overlapped. The outer end of each vane 94 is rounded while the inner end is tapered, and by this tapering, the throat width ws can be kept within an optimum range. The air discharged from the impeller 90 flows along the vane 94 at about the set flow rate, but the air flow at a small flow rate breaks away in the semi-vaneless space, as indicated by the arrows in the drawing, on the suction pressure side of the diffuser vane; therefore, the direction of the air stream is forcibly changed by the taper portion on the pressure side of the adjacent vane, thereby alleviating the broken air stream, so that the zone of surge generation is shifted more to the side of a small flow rate.

Fig. 10 shows the result of experiments on the relationship between the flow rate and pressure (static pressure) of the electric blower, in which the solid curve corresponds to the case including a diffuser based on the embodiment of Fig. 9. The broken curve corresponds to the case for comparison including a diffuser whose inlet angle is 5° . Although the comparison case shows the surge generation zone in the vicinity of a design point, the embodiment with a diffuser inlet angle of 3° can shift the surge generation zone to a small flow rate range.

Claims

1. A vacuum cleaner having an impeller (90), a blower motor (81) coupled to the impeller (90) and a diffuser (89) having a plurality of diffuser vanes (94) arranged radially outside the impeller, characterised in that the inlet angle of said diffuser vanes (94) is in the range 1 to 4° .
2. A vacuum cleaner according to claim 1 wherein the ratio of the throat width (ws) between adjacent pairs of said diffuser vanes (94) to the inlet diameter of said diffuser vanes (94) is in the range 0.017 to 0.025.
3. A vacuum cleaner according to claim 1 or claim 2 having a return guide passage for guiding air from said diffuser vanes to said blower motor for cooling the motor.
4. A vacuum cleaner according to claim 3 wherein said return guide passage extends radially inwardly and is separated from the region of said impeller (90) and said diffuser vanes (94) by a wall (89a) having a rounded outer peripheral edge.

5. A vacuum cleaner according to any one of claims 1 to 4 wherein the ratio

$$\frac{\text{total area of diffuser vane throats}}{\text{real area of diffuser inlets}}$$

given by

$$\frac{Z_{vd} \cdot b_3 \cdot ws}{D_3 \cdot b_3 \cdot \sin \beta_3}$$

is in the range 1.75 to 3.5,

where

- Z_{vd} = number of diffuser vanes,
- b_3 = axial width of diffuser vanes,
- ws = diffuser vane throat width,
- D_3 = diffuser vane inlet diameter,
- β_3 = diffuser vane inlet angle.

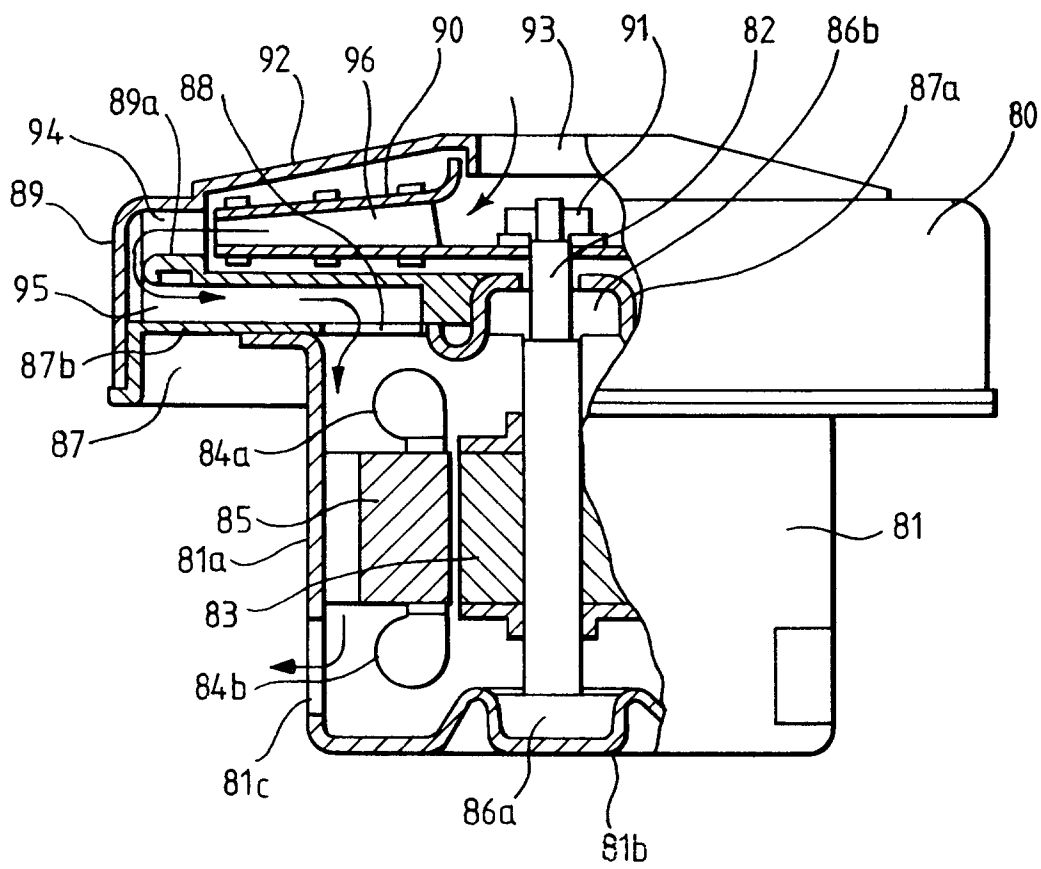


Fig. 1.

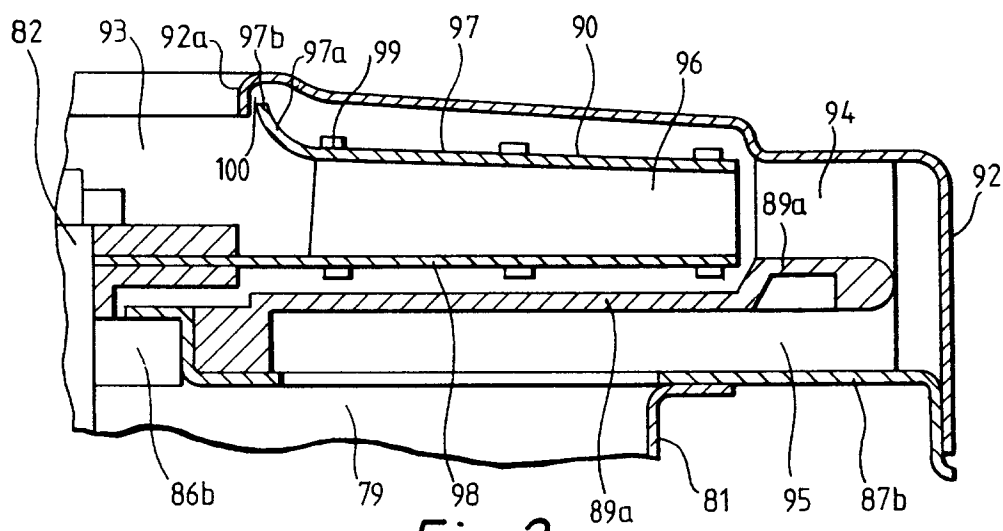


Fig.2.

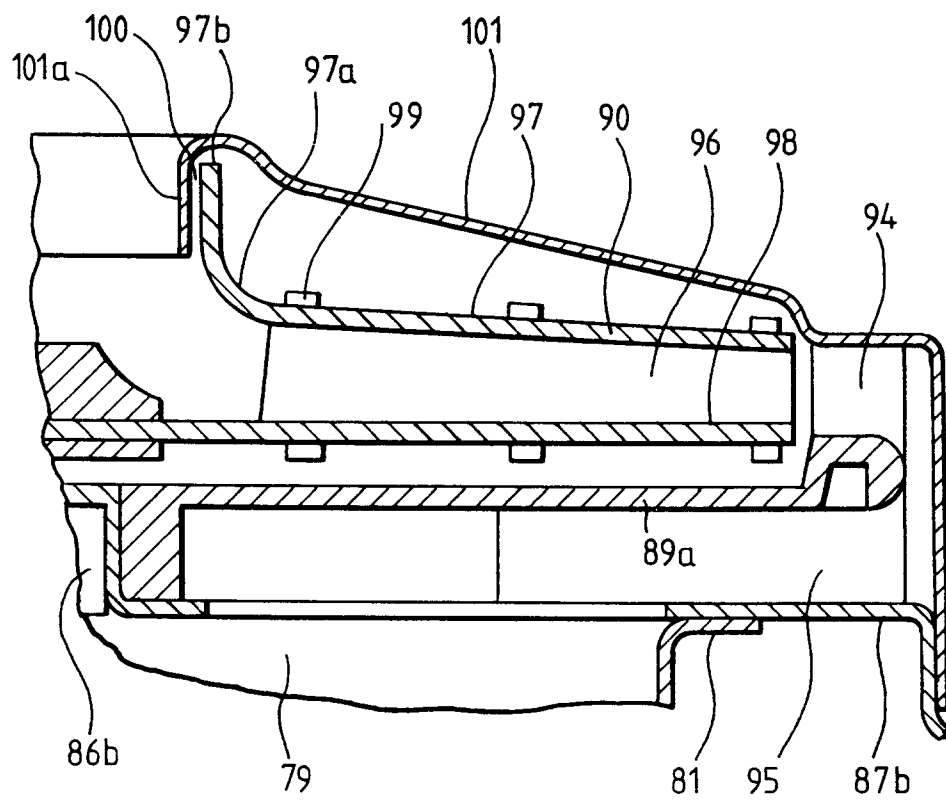


Fig.3

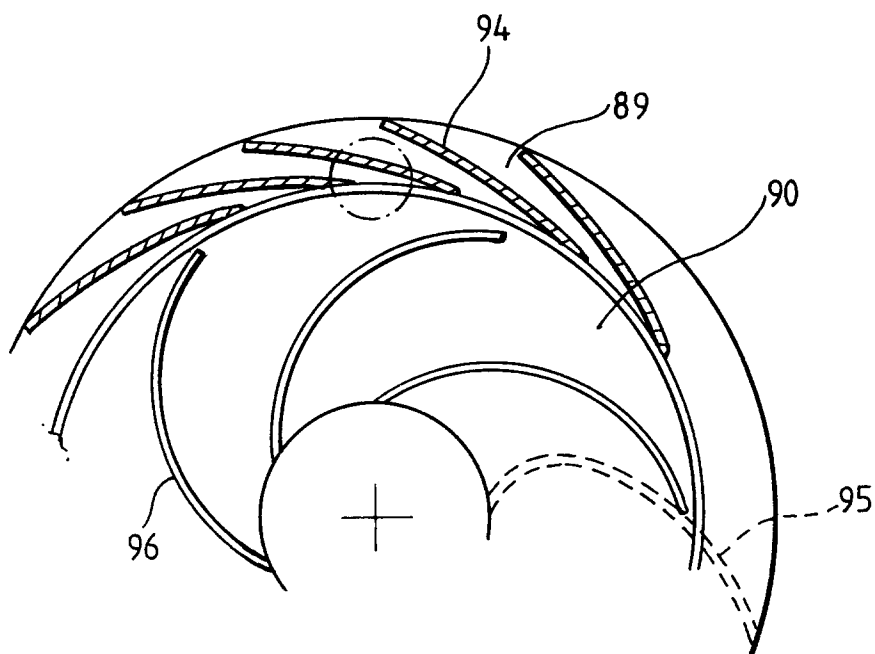


Fig. 4

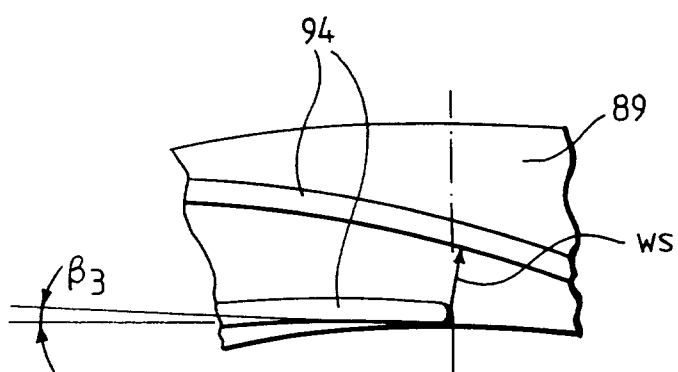
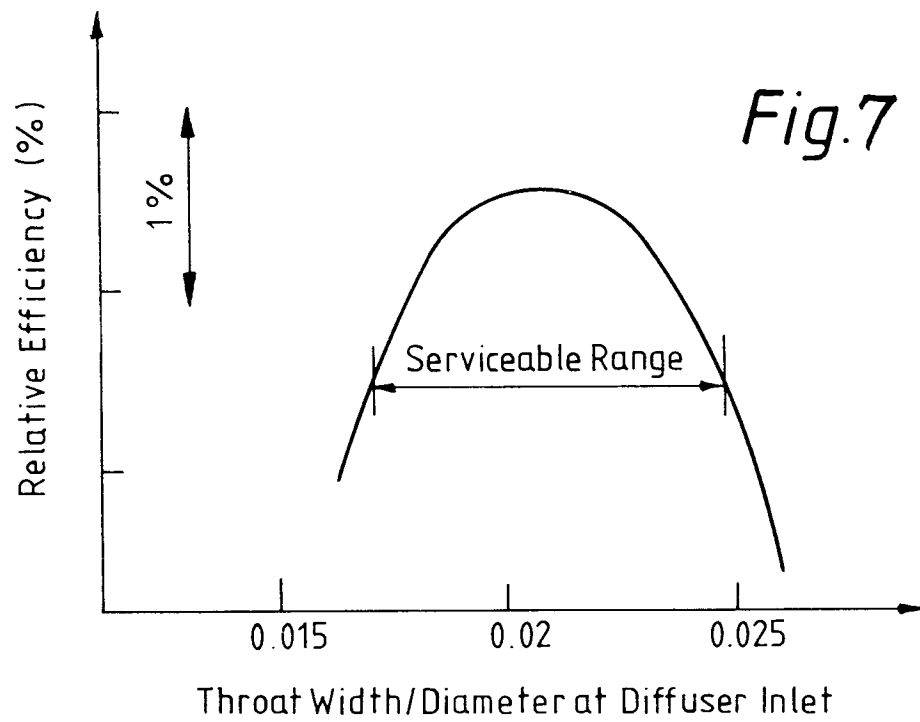
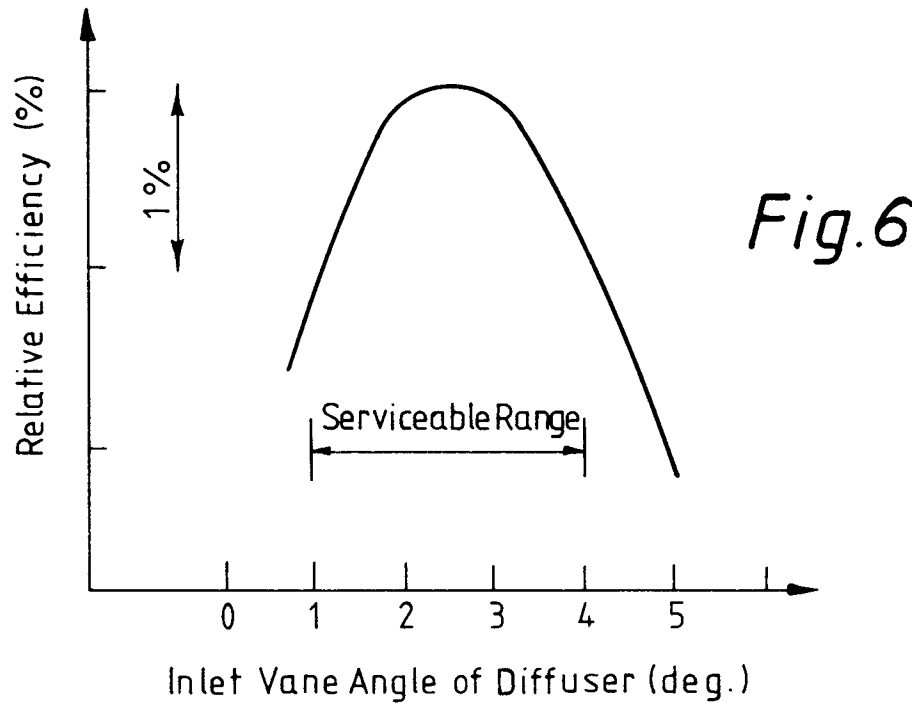


Fig. 5



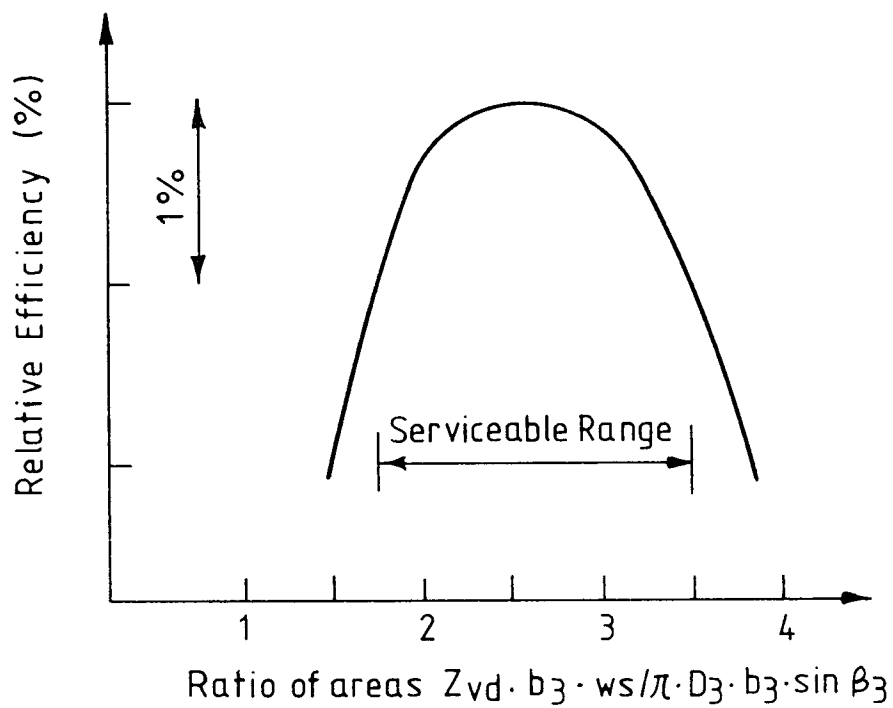


Fig. 8.

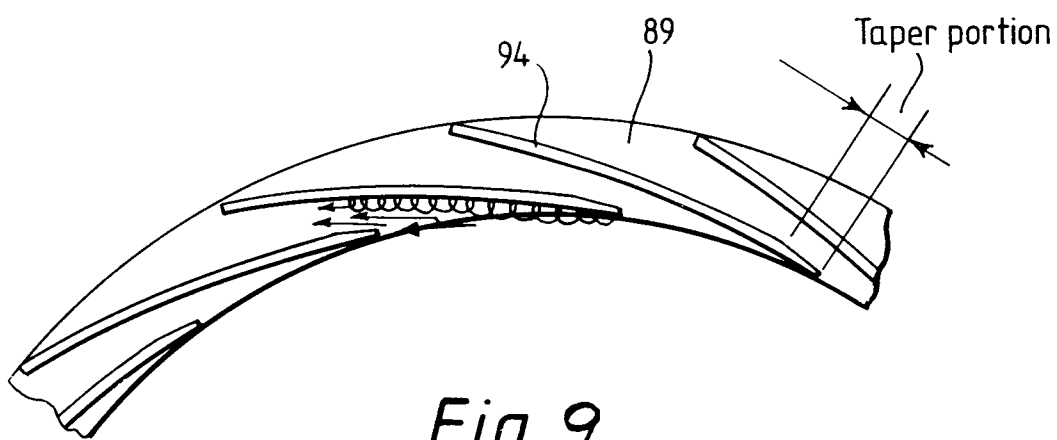


Fig. 9

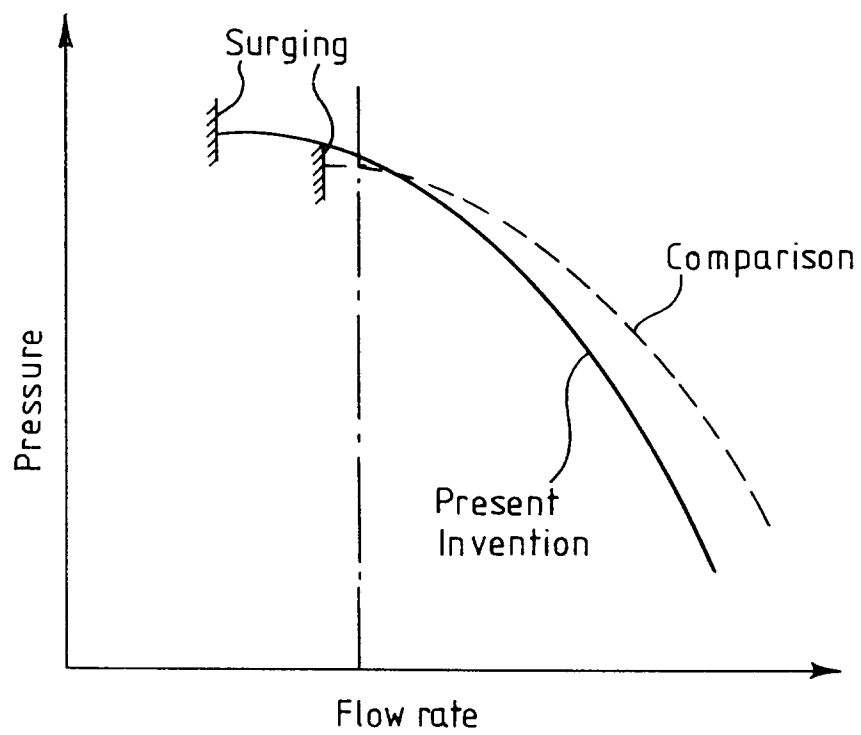


Fig.10.