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ROTOR FOR CENTRIFUGAL COMPRESSORS

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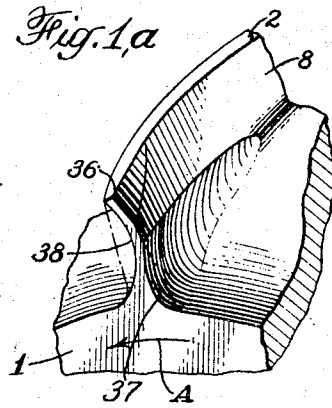
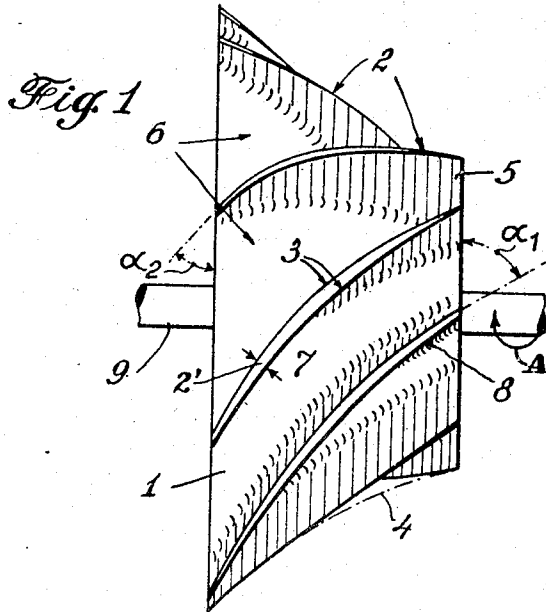
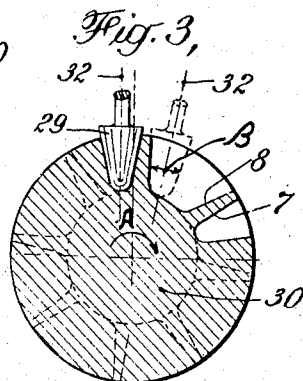
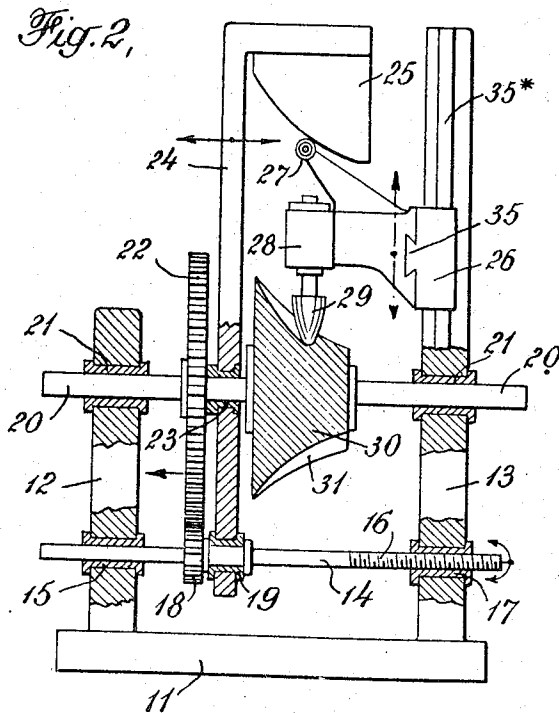
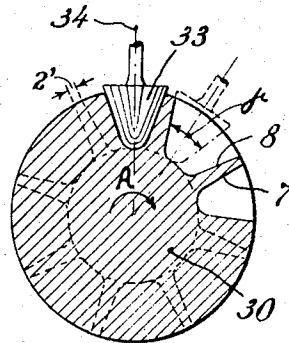


*Fig. 4.*



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## ROTOR FOR CENTRIFUGAL COMPRESSORS

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4 Claims. (Cl. 230—134)

1

The present invention relates to a rotor for a centrifugal compressor wherein the flow takes place diagonally to the rotor axis, having blades bent backwards in a screw shape and having boundary lines derived from meridian curves varying from flat to concave. The invention consists in that on the pressure flank, as well as on the suction flank, the screw of the blades is of uniform pitch, the pitch of the blade surface on the pressure flank being by so much less than the pitch of the surface on the suction flank, that the blade thickness at the blade inlet is approximately the same as at the blade outlet.

Preferably the tip-width of the blades along the periphery of the rotor is almost constant. In preferred forms of execution, and in particular in rotors for high and extra-high peripheral speeds of 400 m./sec. and more, the blades may extend radially to the rotor axis in planes at right angles to the rotor axis and have a profile giving approximately uniform strength. A stream-line taper of the delivery end of the blades may be effected essentially at the top and at the base of the suction flank, and half-way up the pressure flank.

One example of a form of execution according to the invention is represented on the accompanying drawing, in which Fig. 1 gives the side view of a rotor and Fig. 1a shows the blade end at the outlet side to a greater scale. Fig. 2 is a diagrammatic view, partly in section, of a profile milling machine suitable for making the rotor. Figs. 3 and 4 show details of the rotor in cross-section.

The rotor shown in Fig. 1 has screw-shaped blades 2, which are backwardly bent with respect to the direction of rotation A and whose boundary lines 3 are derived from the meridian lines 4 on the rotor 1, these lines running flat or concave. Because of the backward bending of the blades 2, the aim is attained of reducing at the spot 6 the outlet speed of the medium compressed. This prevents the absolute velocity at the outlet, in spite of the high peripheral velocity of the rotor 1 at the spot 6, from reaching the velocity of sound at that spot; this prevents the occurrence of Mach pressure waves which would impair the efficiency. At the inlet the rotor blades 2 make with the tangent at the periphery an angle  $\alpha_1$  of about 60° and at the outlet an angle  $\alpha_2$  of about 45°.

Both the pressure flank 7 and the suction flank 8 of the blades 2 are formed as screw-surfaces which have a uniform pitch. The pitch of the blade-surface 7 on the pressure flank is smaller than that of the blade-surface 8 on the suction flank. The difference in pitch is chosen in such a way that the width 2' (Fig. 4) along the top of the blade, i. e. in the direction of flow from inlet

2

5 to outlet 6 is approximately constant. Technical advantages are thus obtained, not only in connection with the construction, but also with regard to the strength and service qualities. In actual service it is desirable that a blower of certain dimensions can handle as big a quantity of gas as possible without any reduction in efficiency, i. e. the blade passages should have the greatest possible cross-sectional areas, for which purpose a minimum width along the top of the blade is required. With respect to flow, it is technically of advantage to endeavor to have the width of the top of the blade a minimum in order that the boundary-layer friction on the wall of the casing may be kept low. From the point of view of strength, a narrow width along the top of the blade has the advantage that, in a blade stressed by centrifugal forces, the width at the base of the blade is many times greater and is available to take the centrifugal forces. In practice, however, the width along the top of the blade may not be less than a certain dimension, since difficulties would otherwise be encountered in making the blades and they would also be easily damaged during erection and when in service. These different requirements can best be simultaneously fulfilled when the width along the top of the blade remains approximately uniform.

The difference between the pitch of the blade surface on the pressure flank and that on the suction flank amounts to between 1 and 4% of the pitch, depending on the ratio of the inlet diameter to the outlet diameter. When the difference in diameter is great, this percentage amount will be greater than when the difference in diameter is small. For instance, when the ratio of the inlet diameter to the outlet diameter is 1:2, the percentage amount may be in the neighbourhood of 1-2%. One and the same rotor may be employed for different working conditions, when it is preceded at the inlet end by different preliminary rotors specially designed to suit the actual working conditions at the moment.

The mutual position of the screw surfaces on the pressure flank and on the suction flank of one and the same blade 2 is chosen in such a way that blade cross-sections at right angles to the axis of the rotor extend radially, and are therefore subjected only to tensile stresses but not to bending stresses. In order to reduce the tensile stressing, the rotor 1 has no bore for the shaft. The shaft 9 is attached to the rotor 1 by a flange. At the outlet 6 the blades 2 are, as shown in Fig. 1a, gradually reduced in width in a stream-lined manner, a uniform distribution of the speed of the flow being obtained by the blade end at the suction flank being reduced in thickness at the tip 36 and at the base 37, whilst this reduction of the blade end at the pressure flank is made

principally in the middle zone 38 of the blade height.

The profile milling machine according to Fig. 2 has a bedplate 11 which is provided with two columns 12 and 13. The lead spindle 14 is carried in the plain bearing 15 of column 12 and has a screw-thread 16 passing through the nut 17 fitted in the column 13. On the lead spindle 14 the toothed pinion 18 is fixed and the guide bush 19 is arranged. The gear and driving shaft 20 is carried in the plain bearings 21 in columns 12 and 13. On the driving shaft 20 the toothed wheel 22, meshing with the pinion 18, is fixed; the sliding bush 23 is also arranged on shaft 20. On the bushes 19 and 23 is arranged the control bar 24, which can slide between parallel guides (not shown in the drawing), this bar is fitted with a copying template 25. On column 13 is arranged the headstock 26 which can be moved vertically on the guide 35\* and horizontally along the guide 35. The headstock 26 is fitted with a guide-roller 27 and the milling motor 28 which drives the milling cutter 29. On the two-piece shaft 20, the blank 30, which is to be machined to form the rotor, is fixed between column 13 and the control bar 24. The process of making the blading 31 on the rotor blank 30 can be performed as follows by means of the profile milling machine shown in Fig. 2:

At first the shaft 14 is with its threaded-spindle 16 in the starting position at the right-hand side in Fig. 2. The toothed-wheel gear 18, 22, the control bar 24 and the shaft 20 with the rotor blank 30 fixed to it are therefore also in their end positions at the right, whilst the roller 27 and the head-stock 26 are in their highest positions. If the shaft 14 turns, it as well as the toothed pinion 18 and the guide bush 19 are pushed to the left (Fig. 2). This axial displacement is transmitted by the control bar 24 to the shaft 20 and is also performed by the toothed wheel 22 and the rotor blank 30. The pitch of the blades is given by the pitch of the threaded spindle 16, as well as by the ratio between the toothed wheels 18 and 22. The axial displacement of the control bar 24 causes the template 25 to press on the guide roller 27 of the headstock 26, so that the milling cutter 29 moves vertically in accordance with this template. Because of the turning of shaft 14, the shaft 20 along with the rotor blank 30 is through the toothed gear 18, 22 also caused to rotate, and in this way a screw-shaped groove of constant pitch is milled out. On the rotor blank with its diameter decreasing from a large to a small value, it is evident that in this case the angle which the blade makes with the tangent at the periphery increases from a minimum value at the greatest diameter to a maximum value at the smallest diameter. Preferably the ratio of the axial displacement of the shaft 14 to the angle of rotation of the shaft 20 is chosen in such a way that the angle made by the blade with the tangent at the periphery amounts to about 45° where the diameter of the rotor is greatest and to about 60° where the diameter is smallest.

Considerations of solid geometry show that for certain diameter-ratios at the inlet and outlet ends 5, 6 of the rotor 1 and for certain meridian lines 4, it is necessary to displace the axis of the milling cutter with respect to the axis of the rotor in order that the tops of the blades may be of uniform width over their whole length.

This displacement can also be used with advantage to provide favourable conditions for cutting the tapered milling profile, as is the case with vertical milling cutters with a small angle of taper.

According to Fig. 3 of the drawings a vertical milling cutter 29 is employed which has a double taper angle  $\beta$  amounting to about 15° and gives technically advantageous conditions for the cutting. When milling the blade surface 7 on the pressure flank with respect to the direction of rotation A of the rotor, the axis 32 of the milling cutter is displaced forwards in the direction of rotation with respect to the rotor axis by a definite amount, for instance by 20 mm. in the case of a rotor with a diameter of 30 cm. at the inlet and 60 cm. at the outlet. For milling the blade surface 8 on the suction flank the axis 32 of the same milling cutter 29 is displaced backwards by the same amount with respect to the rotor axis.

By foregoing the possibility of choosing the ratio of the diameters at the inlet and outlet ends of the rotor as well as the meridian lines of the rotor in some desired manner, i. e. in accordance with some technical considerations to favour the flow or to facilitate construction, the blades of the rotor may be constructed by simplified milling methods with the top of the blades of approximately uniform width over their entire length. As illustrated in Fig. 4, the milling operation of the blade surfaces 7, 8 on the pressure and suction flanks is effected without any displacement of the axis of the milling cutter, the axis 34 of the cutter 33 remaining directed towards the rotor axis. The double milling cutter angle  $\gamma$  amounts in this case to 30° and more.

I claim:

1. Rotor for a centrifugal compressor wherein the flow takes place diagonally to the rotor axis, having blades bent backwards in a screw-shape and with boundary lines derived from meridian-curves varying from flat to concave, the pressure flank as well as the suction flank of the said blades being formed each by a helix of uniform pitch, the pitch of the pressure flank being by so much less than the pitch of the suction flank, that the tip width of the blades at the inlet is approximately the same as the tip-width at the outlet.

2. A rotor for a centrifugal compressor as set forth in claim 1, in which the tip-width of the blades along the periphery of the rotor is almost constant.

3. A rotor as set forth in claim 1, in which the blades in planes at right angles to the rotor axis extend radially to the rotor axis and have profiles giving approximately uniform strength.

4. A rotor as set forth in claim 1, in which the tip-width of the blades along the periphery of the rotor is almost constant and a streamline taper of the delivery end of the blades is effected essentially at the top and at the base of the suction flank and halfway up the pressure flank.

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#### REFERENCES CITED

The following references are of record in the file of this patent:

#### UNITED STATES PATENTS

| No | Number    | Name | Date          |
|----|-----------|------|---------------|
| 70 | 2,189,767 | Vose | Feb. 13, 1940 |
|    | 1,797,455 | Vose | Mar. 24, 1931 |