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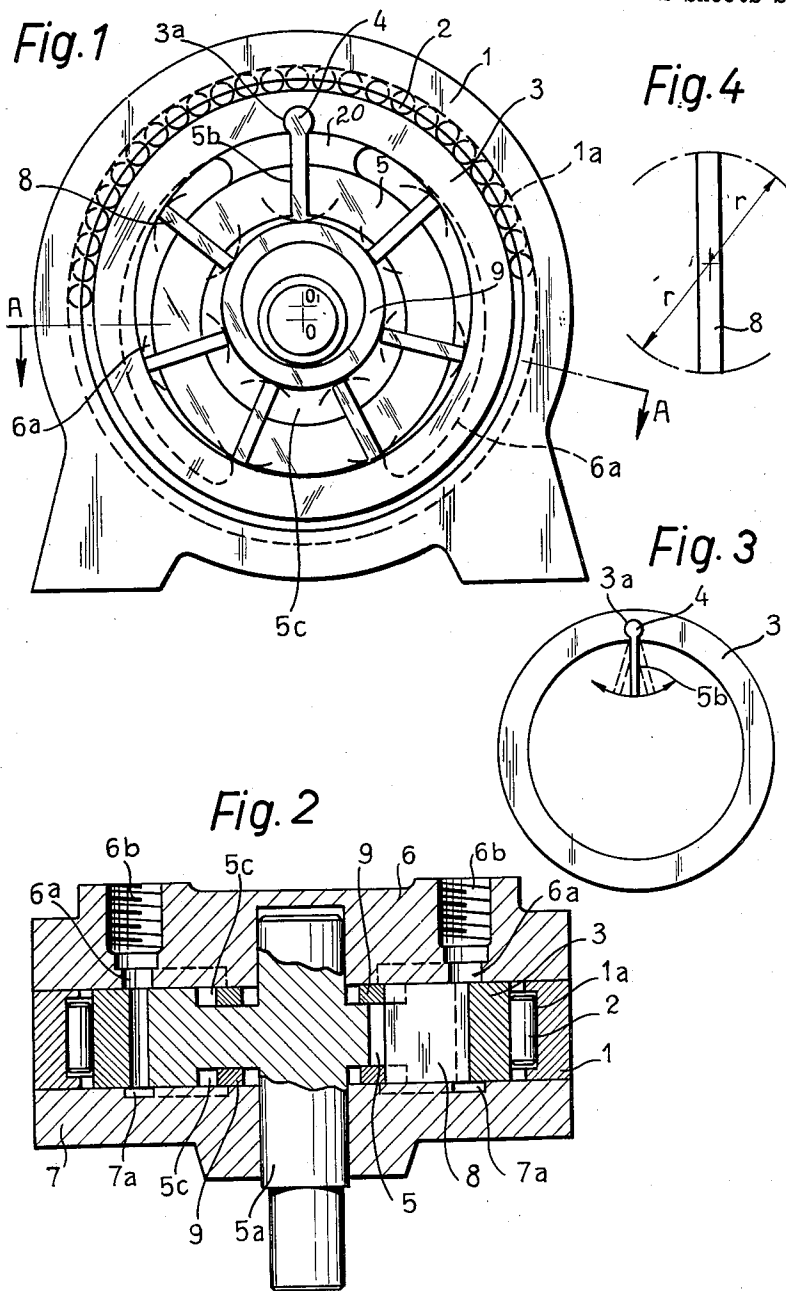
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ROTARY SLIDABLE-VANE MACHINES

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2 Sheets-Sheet 1



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Fig. 5

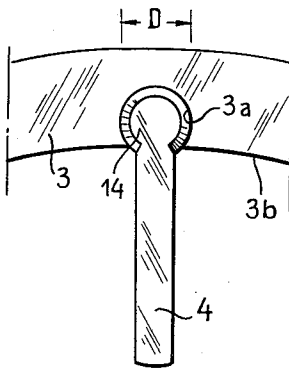


Fig. 6

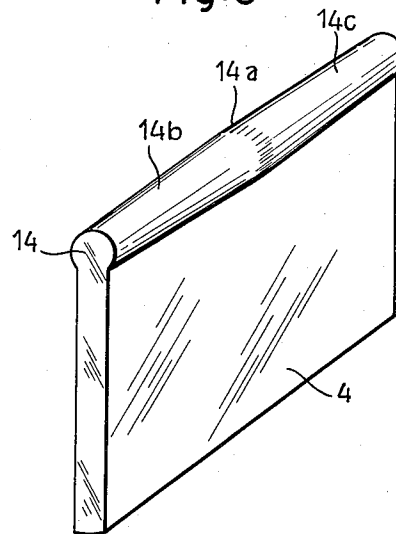
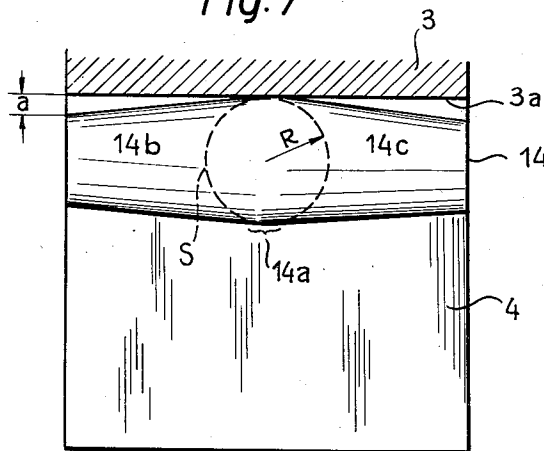


Fig. 7



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ROTARY SLIDABLE-VANE MACHINES

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3 Claims. (Cl. 103—121)

The present invention, a continuation-in-part of application Serial Number 699,227, filed November 27, 1957, now Patent Number 2,891,482, relates to a rotary machine, such as a pump, compressor or fluid operated motor, of the type comprising a cylindrical casing having end walls, a hollow cylinder rotatably supported from and in the casing coaxially with the latter and defining with said end walls a chamber, a rotor rotatable in the chamber eccentrically with respect of the cylinder and casing, said rotor having a plurality of radial slots, a driving vane radially slidable in one of the slots having its radially outer edge section anchored to the cylinder thereby coupling the cylinder and rotor for unison rotation in the casing, and a plurality of secondary vanes radially slidable in remaining slots in the rotor to bear on the inner surface of the cylinder.

It is quite difficult in practice that a slidable vane rotary pump of the type referred to above be also apt to operate as a slidable vane rotary motor. In operation as a pump, the rotor of the machine is rotated by a prime mover the power of which should be sufficient in any case for overcoming the inner frictions of the machine, even if they are of substantial value and obtaining the desired pumping action. As opposed thereto, in a fluid motor the pressure fluid should overcome the inner frictions and rotate the rotor. If the inner frictions are substantial, the working power on the rotor shaft is only a very small part of a total power supplied by the pressure fluid. Therefore, a motor of this kind becomes uneconomical. Unfortunately, the inner frictions in motors of this type increase when the fluid pressure increases. Therefore it is difficult to increase the working power by increasing the pressure of the working fluid.

Several complex structures have been devised up to now to obviate the disadvantages set out above in view of increasing the efficiency both of pumps or compressors and motors. Such structures depart however in a substantial degree from the simple arrangement and cooperation of parts as referred to above, thereby giving rise to difficulties in manufacture, assembly and maintenance, especially when applied to fluid operated motors.

I have found now that one of the most important reasons for the objectionably low efficiency of rotary machines of the type referred to above resides in the anchoring means structure between the driving vane and cylinder.

It is, therefore, an object of the present invention to provide a novel anchoring means for the driving vane and cylinder, whereby a certain freedom of relative displacements between the vane and cylinder can result, leading to a substantially reduced friction between the casing, cylinder, driving vane and rotor. To this end I provide in combination an axially directed groove in the inner surface of the cylinder having a constant circular cross-sectional profile to accommodate the radially outer section of the driving vane, and the said radially outer edge section of the driving vane consisting of a bead comprising a substantially limited annular surface section located

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longitudinally mid-way of the bead, and a pair of oppositely arranged frusto-conical sections on the bead coaxial with the said annular surface section to converge from the latter towards the opposite ends of the bead.

With this and other objects in view which will become apparent in the following detailed description, the present invention will be clearly understood in connection with the accompanying drawings, in which:

Figure 1 is a schematic end view of a fluid operated slidable-vane motor, an end wall of its casing being removed to show the inner structure of the motor;

Fig. 2 is a cross-sectional view substantially on the line 2—2 of Fig. 1;

Fig. 3 shows in an end view the driving vane and cylinder of the motor;

Fig. 4 is an end view of one of the secondary slidable vanes of the motor;

Fig. 5 shows in an end view a detail of the anchoring structure between the driving vane and cylinder;

Fig. 6 is a perspective view of the driving vane; and

Fig. 7 is a plan view of the driving vane accommodated in the cylindrical cavity in the cylinder.

Referring now to the drawings, and in particular to Figs. 1 through 4, a stationary casing 1 of the motor presents in its internal surface a rolling track 1a for the rollers 2, on which the cylinder 3 rotates. The cylinder 3 is internally grooved at 3a into which groove there is lodged an outer edge section of a driving vane 4, running radially free in the radial slot 5b of the rotor 5. The rotor 5 is rotatable on bearings formed in the end plates 6 and 7 of the motor, around the axis O, eccentric in respect to the axis O₁ of rotation of the cylinder 3 and forms jointly with the cylinder 3 and end plates 6, 7 a crescent shaped chamber 20. The rotor 5 has a number of radial slots in which slide freely a plurality of secondary vanes 8 in addition to the aforementioned driving vane 4; the inner edges of said vanes are guided by a pair of centering rings 9 lodged in two annular recesses 5c which are formed on the two opposite faces of the rotor. The rings 9 are set between two parallel flat surfaces constituted by the bottom of said recesses and by the inner sides of the end plates 6 and 7, and are free to run radially between said surfaces. The inner and outer edges of the secondary vanes 8 and the inner edge of the driving vane 4 present a cylindrical surface of radius r (see Fig. 4), this radius being equal to half the height of the vanes 8, whereby the centering rings 9 are always central in respect to the rotatable cylinder 3 and that the vanes 8 continuously contact by their inner and outer edges respectively the rings 9 and the internal surface of the cylinder 3. In the end plate 6 there are formed the intake and outlet ports 6a, having the shape of annular segments and symmetrically positioned in respect to the vertical axial plane of the motor. These ports are in communication with the chamber 20 and with the inlet and outlet pipes (not shown), through the orifices 6b. In the end plate 7 recesses 7a are formed in line and opposite to the ports 6a so as to balance the axial pressure on the rotor 5.

The embodiment described can operate equally well either as a pump or as a fluid motor, and, furthermore, can also operate in both directions, since the direction of rotation can be reversed in relation to the connections of the fluid inlet and outlet pipes. When the machine is operating as a motor, and providing the rotor should be driven in anti clock-wise direction in respect to Fig. 1, the pressure fluid intake pipe is connected with the right-hand side port 6b, and the discharge pipe with the left hand port 6b in Figs. 1 and 2. The fluid pressure acting on the vanes 4 and 8 causes the rotor 5 and cylinder 3 to rotate about their respective axes O and O₁. Power is taken-off from the rotor shaft 5a outwardly protruding through the end plate 7. The two centering rings 9 will

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always remain central in respect to the cylinder 3, as the rings 9 are being kept at a constant distance from the cylinder by virtue of the cylindrical configuration of the edges of the vanes 4 and 8 which are running free radially in the slots of the rotor. The two centering rings 9 will, therefore, also rotate around the axis O_1 , and the driving motion between the edges of the vanes and the surfaces of the rings 9 and of the cylinder 3 is reduced to a gentle oscillation depending upon the eccentricity of the two axes of rotation.

Referring now to Figs. 5 through 7, it will be seen that the groove 3a in the cylinder 3 is of a constant circular cross-sectional profile over its length, thereby, to provide a smooth cylindrical bearing cavity opening on the inner surface 3b of the cylinder. The length of the diameter of the groove is indicated by D in Fig. 5 and the groove is parallel with the axis O_1 of the cylinder 3. The groove 3a accommodates an outer edge section 14 of the driving vane 4, the said section being formed as a bead comprising a substantially limited annular surface section 14a located exactly midway between the ends of the bead, and a pair of oppositely arranged frusto-conical sections 4b, 4c, which are coaxial with the section 14a and are divergent from the latter towards the opposite ends of the bead, respectively. Both the annular and frusto-conical sections are of a circular cross-sectional profile. The diameter of the annular section 14a is substantially equal to D, allowance being made for a smooth fit of the section 14a in the groove 3a. Although the annular section 14a could be cylindrical in shape, I have found that best results are obtained when this section substantially corresponds to what will be now described with reference to Fig. 7. The annular section 14a shown in this figure consists of an equatorial zone of a sphere S having a radius R which is a half of the diameter D of the groove 3a (with a necessary allowance as specified above). The frusto-conical surfaces 14b, 14c should be imagined coaxially tangent to the sphere S along two parallels, thereby giving rise to the aforesaid equatorial zone, or spherical annulus 14a, which is in a bearing engagement with the groove 3a. The axial extent of the zone 14a is always very limited as compared with the axial length of the bead 14 and depends upon the conicity of the frusto-conical sections 14b, 14c, the said conicity being strongly exaggerated in Fig. 7 for illustrative purposes. Assuming that each of the frusto-conical sections is spaced from the groove 3a by a radial clearance a at the respective ends of the bead and groove, the optimal value of a has been found ranging between 0.1R and 0.3R. Values of a smaller than 0.1R lead to increasing frictional losses in the efficiency of the motor (or pump), whereas values greater than 0.3R have shown to result in a somewhat noisy operation, probably due to the fact that the bead 14 is too loose in its groove 13a in such conditions.

For what concerns the materials employed, I have successfully experienced steel for the driving vane 4 and cylinder 3, and bronze for the bead 14. I have experienced several pumps and motors of the character specified in the preamble and it is a matter of fact that hardly starting motors of low efficiency have showed to readily start at low operating fluid pressures when equipped with a driving vane substantially as described above. I suppose that such an advantageous performance is a result of a certain freedom of the driving vane and cylinder, whereby the cylinder, driving vane and rotor are somewhat independent in the casing and linked together for an extent just sufficient for a unison rotation. It should be observed that, owing to the rotary arrangement of the cylinder 3 in the casing 1, especially when a rolling bearing is employed, the unitary pressure between the annular section 14a and groove 3a is very low, whereby no objec-

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tion can be made as to the durability of the respective members.

While I have disclosed several embodiments of the present invention, it is to be understood that these embodiments are given by example only and not in a limiting sense, the scope of the present invention being determined by the objects and the claims.

I claim:

1. In a rotary slidable-vane machine of the type comprising a cylindrical casing having end walls, a hollow cylinder rotatably supported from and in said casing coaxially with the latter and defining with said end walls a chamber, a rotor rotatable in said chamber eccentrically with respect to said cylinder and casing, said rotor having a plurality of radial slots, a driving vane radially slidable in one of said slots having its radially outer edge section anchored to said cylinder, thereby coupling said cylinder and rotor for unison rotation in said casing, and a plurality of secondary vanes radially slidable in the remaining slots in said rotor to bear on the inner surface of said cylinder, said inner surface of said cylinder defining an axially directed groove and having a constant circular cross-sectional profile to accommodate said radially outer section of said driving vane, and said radially outer edge section of said driving vane consisting of a bead comprising a substantially limited annular surface section located longitudinally midway of said bead in a bearing engagement with said groove, and a pair of oppositely arranged frusto-conical sections on said bead coaxial with said annular surface section diverging from the latter towards the opposite ends of said bead.

2. In a rotary slidable-vane machine of the type comprising a cylindrical casing having end walls, a hollow cylinder rotatably supported from and in said casing coaxially with the latter and defining with said end walls a chamber, a rotor rotatable in said chamber eccentrically with respect to said cylinder and casing, said rotor having a plurality of radial slots, a driving vane radially slidable in one of said slots having its radially outer edge section anchored to said cylinder, thereby coupling said cylinder and rotor for unison rotation in said casing, and a plurality of secondary vanes radially slidable in remaining slots in said rotor to bear on said inner surface of the cylinder, the inner surface of said cylinder defining an axially directed groove, the latter having a constant circular cross-sectional profile accommodating said radially outer edge section of said driving vane, and said radially outer edge section of said driving vane consisting of a bead comprising a substantially limited annular bearing section, the bearing surface of the latter consisting of a spherical annulus in a smooth engagement with said groove, said annular bearing section being located longitudinally mid-way of said bead, and a pair of oppositely arranged frusto-conical sections on said bead coaxially tangent to the said annular bearing section and diverging from the latter towards the opposite ends of said bead.

3. The machine, as set forth in claim 2, wherein the smaller end of each of said frusto-conical sections has a radial clearance with respect to said groove ranging between 0.1 and 0.3 times the radius of said spherical annulus.

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