

May 14, 1946.

J. C. BUCKBEE

2,400,286

ROTARY MACHINE

Filed June 21, 1944

2 Sheets-Sheet 1

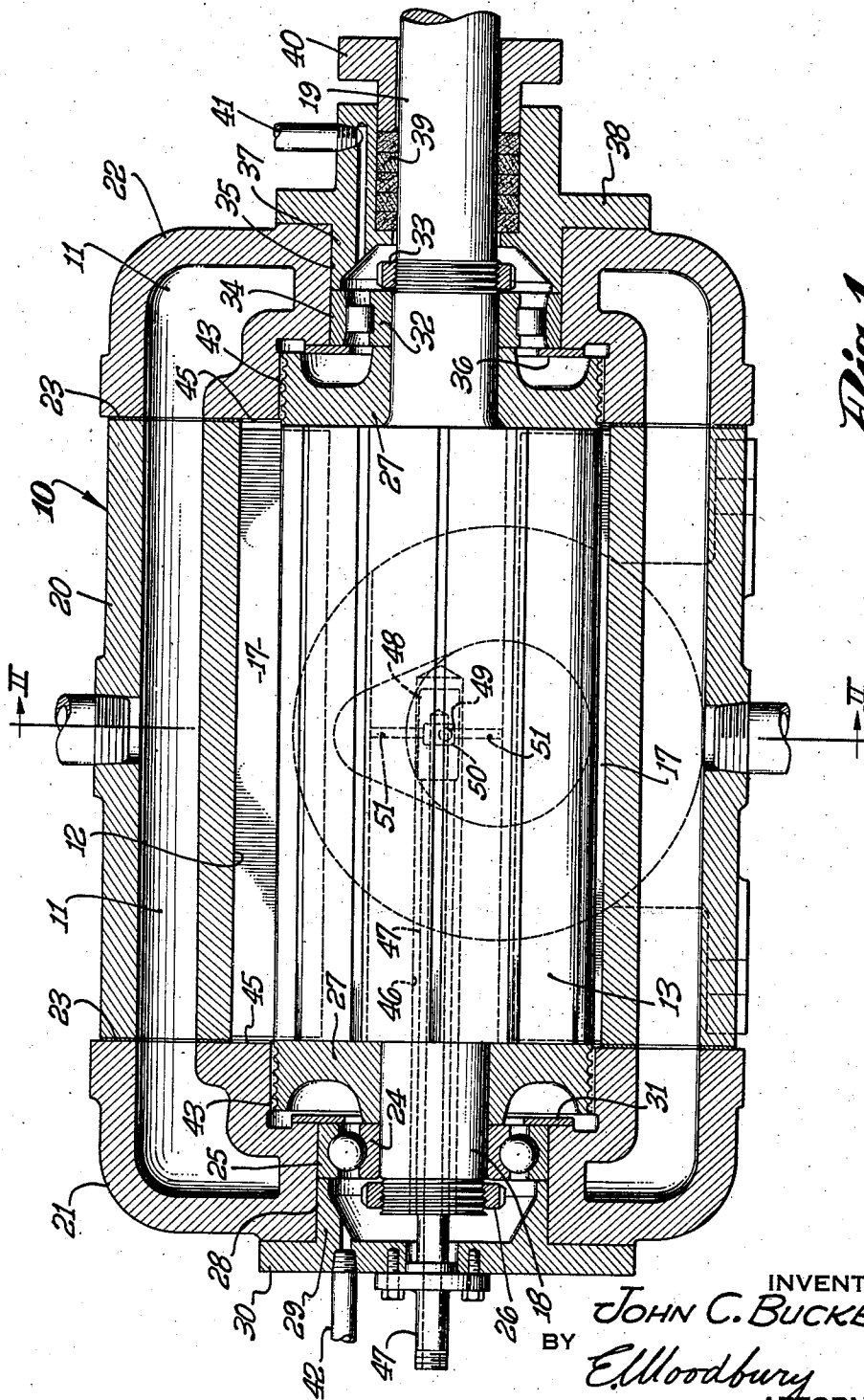


Fig. 1.

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2 Sheets-Sheet 2

Fig. 2.

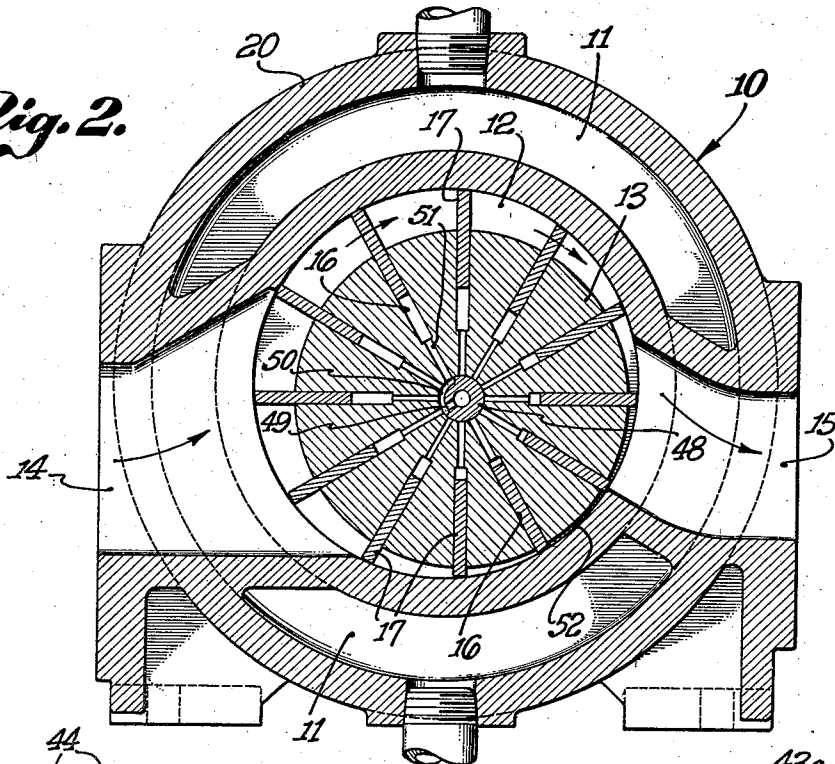
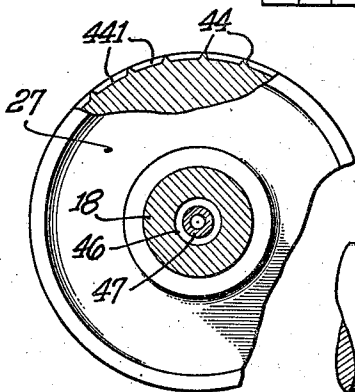


Fig. 3.

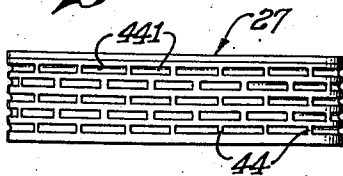


Fig. 4.

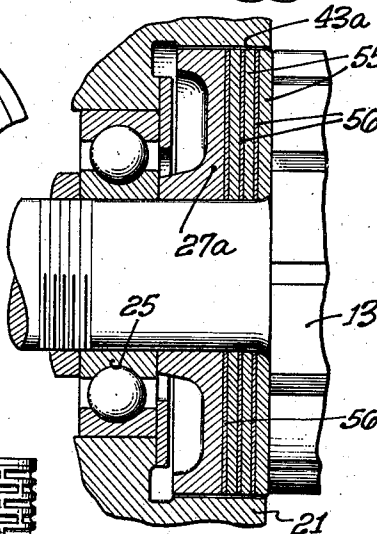


Fig. 5.

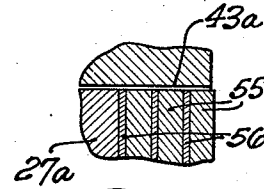


Fig. 6.

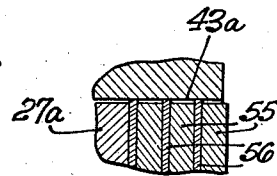


Fig. 7.

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UNITED STATES PATENT OFFICE

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ROTARY MACHINE

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13 Claims. (Cl. 230—152)

This invention relates to rotary pumps and engines of the type employing multiple-sliding-vanes and is particularly useful in connection with air compressors of that type.

An object of the invention is to provide a rotary pump or compressor of the multi-sliding-vane type in which the rotor is free to expand axially and in which the axial expansion does not affect the rotor end running clearances.

Another object is to reduce the leakage losses in a rotary compressor of the multi-sliding-vane type.

Another object is to provide an improved method of lubricating multi-sliding-vane type rotary machines.

Still another object is to provide a labyrinth type seal or packing of improved efficiency.

Other objects and features of the invention will appear from the detailed description to follow.

Heretofore it has been the usual practice in constructing rotary machines of the multi-sliding-vane type to confine the rotor between the fixed end heads of the stator and make the rotor short enough to allow for the expected axial expansion thereof when the machine is in service. In rotary air compressors the rotor often attains an operating temperature of 300° F. or more and the resultant expansion introduces a serious factor because the efficiency of such machines depends mainly upon the maintenance of close running clearances between the rotor and stator to reduce the leakage of air from the high pressure portions of the machine back to the low pressure portions, the latter usually being at the inlet pressure.

Experience has established certain minimum running clearances that are practical for machines of this type. For example, in a compressor having a rotor eight inches in diameter by twenty-four inches long, the minimum axial or end-clearances would have to be about .045 inch and the radial clearance about .006 inch. These clearances are reduced by the expansion resulting from the increase in temperature in service to a total end clearance of about .016 inch, which it is attempted to divide equally between the two ends because .008 inch of end clearance is about the minimum that can be used in a machine of such size without risking danger of contact due to small inequalities in expansion of the rotor segments and the stator. Contact cannot be tolerated because it leads immediately to ruinous friction because of the high speed at which such machines are operated.

In a machine of the dimensions referred to, there would be a leakage area at each end of the rotor equal to the diameter of the rotor multiplied by the clearance, or 8 x .008" which totals .064 square inch at each end of the rotor or .128 square inch for both ends of the machine, which is approximately the equivalent of a $\frac{3}{8}$ inch diameter hole through which air can leak from the high pressure side of the machine back to the low pressure side. Such a leakage area may reduce the capacity of the compressor by as much as 15% and such loss has been unavoidable in the machines as previously designed.

There has been another loss in machines as previously designed through the vane slots of the rotor due to the fact that the slots have been open at both ends, permitting high pressure air leaking into the slots past the surfaces of the vanes to escape into the end clearances of the rotor and be lost back to the inlet. The leakage resulting through the slots may reduce the capacity of a compressor by as much as 10% or more.

In accordance with the present invention, the above referred to losses are greatly reduced or eliminated by extending the ends of the rotor into concentric recesses provided in the end heads of the stator and closing the ends of the slots containing the vanes. With this construction, end clearances between the rotor and the stator is completely eliminated, so that unlimited axial expansion of the rotor can occur without interfering with the free operation of the machine. There is substituted for the axial end clearance a radial clearance between the end extensions on the rotor and the peripheral walls of the concentric recesses in the stator heads into which the rotor extensions project. The radial clearance between the rotor extensions and the walls of the stator head recesses can be made relatively small, and, in accordance with the present invention, leakage through these clearances is further reduced by providing a labyrinth packing on the rotor extensions. The vane slots are closed at the ends by the rotor extensions so that there can be no leakage from the inner portions of the slots below the vanes into the clearance spaces at the ends of the rotor. Instead, air that leaks past the vanes into the inner portions of the slots during outward movement of the vanes is returned to the compression space between the peripheral surfaces of the rotor and stator as the vanes are forced back into the slots by rotation of the rotor. In effect this design utilizes each vane, working as a piston in its slot, to com-

press air and deliver it to the discharge port of the machine.

Lubrication of a multi-sliding-vane type compressor is facilitated in accordance with the present invention by delivering oil through passages in the rotor to the bottom of each vane slot so that the oil is delivered at a point in the slot from which it will be readily circulated to all portions of the slot and vane by the air movements within the slot and by centrifugal force.

A complete understanding of the invention and the manner in which the objects thereof are achieved may be had from the following detailed description with reference to the drawings, in which:

Fig. 1 is a vertical longitudinal section taken through the center of the rotor of an air compressor in accordance with the invention, the rotor and its shafts being shown in full;

Fig. 2 is a cross section taken in the plane II—II of Fig. 1;

Fig. 3 is an end view of a rotor extension of the machine with portions shown in section;

Fig. 4 is a plan view of one of the rotor extensions showing the labyrinth in the peripheral surface thereof;

Fig. 5 is a detail view of a portion of Fig. 1 showing an alternative rotor end construction; and

Figs. 6 and 7 are greatly enlarged views of a portion of Fig. 5 showing the condition of the members at the rotor extension when cold and hot, respectively.

Referring first to Fig. 2, my invention comprises a stator 10 having water passages 11 therein for cooling and having a cylindrical chamber 12 in which a rotor 13 is eccentrically positioned for rotation in clockwise direction as indicated by the arrows. The cylindrical chamber 12 communicates on one side with an inlet passage 14 and on the other side with a discharge passage 15. The rotor 13 is provided with a plurality of symmetrically spaced apart radially extending slots 16 each containing a vane 17 which, during rotation of the rotor at high speed, is forced outwardly against the inner peripheral wall of the chamber 12 by centrifugal force. It will be observed from inspection of Fig. 2 that during rotation of the rotor in clockwise direction, air entering through the inlet 14 will be successively trapped within pockets defined by the rotor, the stator and the vanes 17, and will be compressed by the reduction in volume of the pockets as they pass from the inlet opening to the discharge opening and the air compressed in the pockets will be discharged through the discharge opening 15.

Referring now to Fig. 1, the rotor 13 has axial shafts 18 and 19 which extend from opposite ends of the rotor and serve to rotatably support it within the stator 10, the latter consisting of a cylinder portion 20 and a pair of end heads 21 and 22, respectively, which are secured to the stator cylinder by bolts (not shown), gaskets 23 being provided to prevent leakage.

The shaft 18 at the left end of the rotor (Fig. 1) is rotatably supported in the head 21 by a ball bearing capable of taking both radial and axial loading, this bearing having an inner race 24 mounted on the shaft 18 and an outer race 25 supported in the head 21. The inner race 24 is compressed by a nut 26 on the threaded outer end of the shaft 18 against a rotor extension 27 which is fitted on the shaft 18 and clamped against the end of the rotor 13. The outer race 25 is held in position within a cylindrical passage

28 in the head 21 by a flange 29 on a closure member 30 and a flange ring 31 welded or otherwise secured to the inner orifice of the passage 28.

The shaft 19 is supported by a cylindrical roller bearing capable of taking radial loading while permitting axial movement to permit axial expansion of the rotor with respect to the stator. This roller bearing comprises an inner race 32 pressed against the adjacent rotor extension 27 by a nut 33. The outer race 34 is supported in a cylindrical passage 35 in the head 22 by an inner flange member 36 welded or otherwise secured to the head 22 and a flange member 37 on a closure member 38, the latter containing a packing 39 for effecting a seal about the shaft 19 which is extended through the end of the compressor for connection to a driving means. The usual gland 40 is employed for compressing the packing 39 and pipes 41 and 42 may be provided for supplying lubricating oil to the roller bearing and the ball bearing, respectively.

As described, the rotor extensions 27 are formed as separate elements and securely locked to the rotor. However, it may be desirable under some conditions to form these extensions integrally with the rotor 13. It is usually desirable for manufacturing purposes to make them separate. Irrespective of whether they are formed integrally or separate, they may have the same outside diameter as the rotor and may be machined with the rotor after assembly on the rotor shafts, thereby contributing to economy in manufacture.

It will be observed that the rotor extensions extend into concentrically located cylindrical recesses 43 in the heads 21 and 22, respectively, and that they have a close running fit with the peripheral walls of these recesses so as to reduce to as great an extent as possible through such means the leakage of air, either axially or peripherally through the running clearance. To further reduce the leakage through the running clearance, the peripheral surfaces of the extensions 27 are provided with interrupted grooves or slots 44 as shown in Figs. 3 and 4, such slots constituting an improved seal of the well known labyrinth type. However, in the seals of this type that have been heretofore employed, the peripheral grooves were continuous, causing them to be substantially ineffective to prevent peripheral circulation of air between the parts. By interrupting the grooves by staggered ridges 44 as shown in Figs. 3 and 4, the resistance to peripheral leakage is substantially increased without decreasing the efficiency of the seal against axial leakage.

Obviously, the rotor can expand axially in response to increased temperature without altering the free running characteristics of the machine since there are no close end clearances between the rotor and the stator. Axial expansion simply shifts the rotor extension 27 slightly to the right with respect to the adjacent concentric sealing surface 43. However, running clearance must be maintained at all times between the ends of the vanes 17 and the faces 45 of the heads 21 and 22 surrounding the recesses 43 in the heads 22 and 21, respectively. Hence, the rotor and the vanes must be shorter than the distance between the faces 45 of the heads 21 and 22 to prevent binding of the ends of the vanes 17 and the faces 45 when the machine heats up in operation and the rotor and vanes expand.

It is also necessary to provide some clearance

between the ends of the vanes 17 and the ends of the slots in which they are positioned in order that the vanes may slide freely in the slots 16, the ends of the slots in the particular construction shown being closed by the rotor extensions 27. This clearance may be relatively small in the size of machine cited above, say of the order of .003 inch, since there will ordinarily be no great difference in temperature between the rotor and the vanes or in the expansion thereof. In order to illustrate the relative magnitude of the clearances between the ends of the vanes and the faces 45 and between the ends of the vanes and the rotor extensions 27, these clearances have been greatly exaggerated in Fig. 1.

When the machine is in operation, the vanes 17 are forced outward by centrifugal force into contact with the peripheral wall 12 of the stator and move in and out of their slots during each rotation of the rotor because of the eccentric mounting of the rotor in the stator. Furthermore (referring to Fig. 2), as the vanes pass the inlet opening 14, they are near their outermost positions and trap air at low pressure, as previously described, in the pockets defined by the rotor, vanes and stator. Low pressure air also flows into the space in the slots 16 below the vanes 17 through the clearance spaces between the vanes and the slots and at the ends of the vanes. As the vanes move inwardly during their passage from the inlet to the outlet opening, they compress the air in the portions of the slots below the vanes, which air is forced out past the ends of the vanes through the clearances between the vanes and the rotor extensions 27 and the clearances between the slots 16 and vanes 17. Hence, this air is returned under pressure into the pockets adjacent that vane and is delivered to the discharge opening 15. In prior constructions the air forced out from the slots below the vanes was lost into the rotor end clearance spaces of the machine and flowed back to the inlet.

During a large part of the return movement of the vanes from the discharge side to the inlet side of the machine and during their movement past the inlet opening, the vanes are moving outwardly and drawing in air at low pressure in the space in slots 16 below the vanes 17 through the clearances at the ends of the vanes and the side clearances of the vanes in their slots. The overall effect is that in addition to the major compression produced by the contraction of the pockets defined between the rotor, vanes and stator during the movement of those pockets between the inlet and outlet openings, there is an auxiliary piston action of the vanes in their slots which functions to draw air into the slots below the vanes during outward movement of the latter and compress this air and discharge it into the pockets for delivery to the discharge opening 15 during inward movement of the vanes.

In addition to the increase in efficiency resulting from the utilization of the air compressed in the slots below the vanes, there is a further increase in efficiency resulting from the extension of the rotor into the head recesses 43 due to the fact that longitudinal expansion of the rotor relative to the stator does not affect the radial clearances between the rotor extensions 27 and the peripheral walls 43 of the concentric recess in which they fit. Hence, these clearances can be made very small, greatly reducing the leakage of pressure air from the pockets at the rotor ends. When the rotor ends are provided with labyrinth packing, the leakage is further reduced.

To provide adequate and positive lubrication for the vanes 17 in the rotor slots, I provide for the direct delivery of oil to the bottom of the slots 16 in the rotor 13 at a point substantially midway the ends of the rotor. To this end, I provide a cylindrical passage 46 extending through the axial center of the rotor shaft 18 to a point beyond the longitudinal center of the rotor and extend into this passage 46 an oil pipe 47 which is non-rotatably supported on the closure member 30 and is adapted to be connected to a source of oil under pressure. The inner end of this pipe 47 has an enlarged portion 48, the outer surface of which has close running clearance with the surface of the passage 46. A lateral passage 49 extends from the central passage in the member 48 to the surface of the latter where it connects with a recess or oil distribution pocket 50 extending circumferentially approximately 90° facing towards the inlet 14 of the machine. Juxtaposed to the oil pocket 50 in the enlarged portion 48 of the pipe 47 are a plurality of radial passages 51 in rotor 13 extending from the passage 46 to the bottom of the rotor slots 16.

It will be observed from inspection of Fig. 2 that the radial passage 51 associated with each vane slot moves into communication with the recess 50 about the time the associated vane begins to move across the inlet opening 14 so that while the vane is moving outwardly in its slot, oil may flow freely through the associated radial passage 51 into the vane slot. After the vane has moved past the inlet opening and is being forced back into its slot, the associated passage 51 moves clear of the recess 50 so that the oil previously supplied to the slot cannot be returned through the passage 51. Part of the oil is thrown outwardly by centrifugal force between the sides of the vane and the sides of the slot and lubricates the vanes and partially seals the clearances at those points with oil. Another portion of the oil is carried longitudinally in the slot by the movement of the air as it is forced out of the slots around the end clearance in response to the inward movement of the vane. All the oil escaping through the clearances between the sides of the vanes and the slots and past the ends of the vanes is thrown out against the peripheral wall 12 of the stator and lubricates the edges of the vanes 17 bearing thereon.

The left end bearing may be lubricated by oil supplied through the pipe 42 (Fig. 1) and the right end bearing by oil supplied through the pipe 41. Excess oil from the bearings finds its way into the labyrinth seals on the rotor extensions 27 and increases the sealing efficiency thereof.

It has been mentioned that the oil was supplied to the pipe 47 under pressure. This pressure need not be great and, in fact, may, under some conditions, be substantially at atmospheric pressure since during the outward movement of the vanes in their slots, they create a suction therebelow tending to draw the oil into the slots.

As shown in Fig. 1, the rotor extensions 27 are of substantially the same diameter as the rotor itself. However, it may be desirable under some conditions to fit the rotor extensions 27 more closely within the recesses 43 than the rotor is fitted to the stator at the point 52 of closest approach thereto because the rotor, being relatively long, may distort slightly when heated due to the inequality of the structure of the metal of which the rotor is made. Under certain conditions it may be desirable to make the rotor ex-

tensions of a material having a greater coefficient of expansion than the rotor, thereby enabling the extensions and the rotor to be machined to the same diameter at normal temperature but resulting in a larger diameter for the extensions 27 than for the main body of the rotor when the rotor and the extensions heat up in service.

It is to be understood that machines of the type herein referred to are often operated at relatively high speeds and it is conceivable that the vanes may move in and out of their slots so rapidly as to give insufficient time for much air to be displaced through the clearance at the ends of the vanes into and out of the slots below the vanes. However, this is immaterial since if air is not drawn into the slots, no energy is required to compress such air. On the other hand, when a compressor or pump of the general type herein referred to is operated under conditions in which it is impossible to obtain a perfect seal at the ends of the vanes, the present invention utilizes the leakage air that would be lost with prior constructions.

An alternative rotor end construction is shown in Figs. 5, 6 and 7. In the modified structure, instead of employing a solid rotor extension 27, I employ a laminated structure consisting of a plurality of disk elements 55 of a material such as steel, semi-steel or alloy cast iron having about the same coefficient of expansion as the main body of the rotor 13 and the outer portion 27a of the rotor extension. These disks 55 are interleaved with a plurality of relatively thin disks 56 made of some material having a higher coefficient of expansion such as brass or bronze or aluminum.

In manufacturing, the disks are compressed against the rotor by means of the nuts 26 and 33 (Fig. 1) as previously described and the whole assembly is ground or turned down to the same diameter; that is, the diameter of the rotor.

The recess 43a (Fig. 5) in the end head is of diameter to accommodate the rotor extension with sufficient running clearance with the disks 55 and the element 27a when the rotor is hot.

Since the interleaved disks 56 have a larger coefficient of expansion than steel or cast iron, they will become larger in diameter than the disks 55 and the element 27a when the rotor is placed in service and heats up, thereby causing these disks 56 to extend slightly beyond the peripheries of the disks 55 and automatically from a labyrinth packing against axial flow of air.

The first time the machine is placed in service, the disks 56 are expected to expand sufficiently to actually contact the peripheral wall 43a of the head recess, but because of the relative softness of the material employed and the thinness of the disks 56, they will quickly wear to establish a minute running clearance.

Figs. 6 and 7 show how the edges of the disks 56 are flush with the disks 55 and the element 27a when the device is cold and protrude beyond the disks 55 and element 27a when the device is hot.

The labyrinth packing resulting from the use of the disks 55 and 56 having different coefficients of expansion has no restriction against circumferential flow as does the packing shown in Figs. 3 and 4, but by making the disks 55 relatively thin, the area of the circumferential channels defined between the protruding edges of the disks 56 can be made so small that circumferential flow will be of little moment. Obviously, the number of the disks 55 and 56 and their thickness may

be varied between wide limits, depending upon the design factors of the compressor such as the pressure to be developed.

Various changes from the exact construction shown and described can be made without departing from the invention which is to be limited only to the extent set forth in the appended claims.

I claim:

1. A device of the type described comprising: a stator having a peripheral wall and end walls defining a rotor chamber, said end walls having aligned cylindrical recesses of smaller cross sectional area than said chamber; a cylindrical rotor of slightly smaller diameter than said recesses extending through said rotor chamber and into said recesses at each end, said rotor having axially extending slots, and vanes slidably supported in said slots for movement toward and away from the peripheral wall of said rotor chamber, the vanes being of axial length slightly less than the axial length of said rotor chamber to seal therewith; the peripheral surface of said rotor adjacent ends of said rotor sealing with the peripheral surfaces of said recesses but the ends of the rotor being spaced from the ends of the recesses whereby the rotor is free to expand axially with respect to said stator without affecting the running clearances between the rotor and stator.

2. A device as described in claim 1 including bearing means at opposite ends of said rotor for rotatably supporting said rotor in concentric relation with said recesses.

3. A device as described in claim 1 in which the end portions of said rotor projecting into said recesses have interrupted, circumferentially extending grooves in their peripheral surfaces constituting labyrinth type seals with the juxtaposed peripheral surfaces of said recesses.

4. A device as described in claim 1 in which the end portions of said rotor projecting into said recesses have pluralities of circumferentially extending grooves in their peripheral surfaces constituting labyrinth type seals with the juxtaposed peripheral surfaces of said recesses, said grooves being interrupted at circumferentially spaced intervals and in staggered relation with respect to adjacent grooves.

5. A device as described in claim 1 in which the ends of said slots are closed and the axial length of said slots is greater than that of said vanes but less than that of said chamber.

6. A device as described in claim 1 in which the ends of said slots are closed and the clearance between the ends of said vanes and the ends of said slots is sufficient to permit escape therepast of fluid from below the vanes during movement of the vanes into the slots.

7. A device as described in claim 1 in which the rotor comprises a central, slotted portion extending through said rotor chamber and separate, solid portions secured to opposite ends of said central portion and constituting said end portions that seal with the peripheral surfaces of said recesses.

8. A device as described in claim 1 in which the rotor comprises a central, slotted portion extending through said rotor chamber, and separate, solid portions secured to opposite ends of said central portion and constituting said end portions that seal with the peripheral surfaces of said recesses, said solid portions being of material having a higher coefficient of expansion than the said central portion.

9. A device of the type described comprising:

a stator having a peripheral wall and end walls defining a rotor chamber; a rotor having axially extending slots and vanes in said slots supported thereby for movement toward and away from the peripheral wall of said rotor chamber; means rotatably supporting said rotor in eccentric relation to said rotor chamber, whereby said vanes are alternately moved inwardly and outwardly of their slots in response to rotation, said rotor having an axial passage extending thereinto and distributing passages extending from said axial passage into said slots; a stationary lubricating fluid supply conduit extending into said axial passage and fitted therein with a close running fit adjacent the inner orifices of said distributing passages and having a lateral discharge opening adapted to successively register with said distributing passages during rotation of said rotor, said lateral discharge opening being so oriented with respect to said stator that it is connected with said distributing passages while the vanes associated with said passages are moving outwardly in their slots.

10. A device of the type described comprising: a stator having a peripheral wall and end walls defining a rotor chamber, said end walls having aligned cylindrical recesses of smaller cross sectional area than said chamber; a cylindrical rotor of slightly smaller diameter than said re-

cesses extending through said rotor chamber and into said recesses at each end; vane means on said rotor within said chamber; said rotor having at least one end portion extending into one of said recesses which end portion consists of laminations of material of low thermal expansion alternated with laminations of material of high thermal expansion adapted to more closely approach the peripheral wall of said recess when the device heats up in service.

11. A device as described in claim 10 in which said laminations are all of the same diameter at normal room temperature.

12. A device as described in claim 10 in which said laminations of high thermal expansion are relatively thinner than said laminations of low thermal expansion.

13. A device as described in claim 10 in which the clearance between the peripheral surfaces of said one end portion and the recess into which it extends when the device is cold is initially less than the expansion of the laminations of high thermal expansion, whereby when the device heats up in its first operation the said laminations of high thermal expansion expand into contact with said wall member and wear to a minute running clearance.

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