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ROTARY COMPRESSOR

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FIG-1

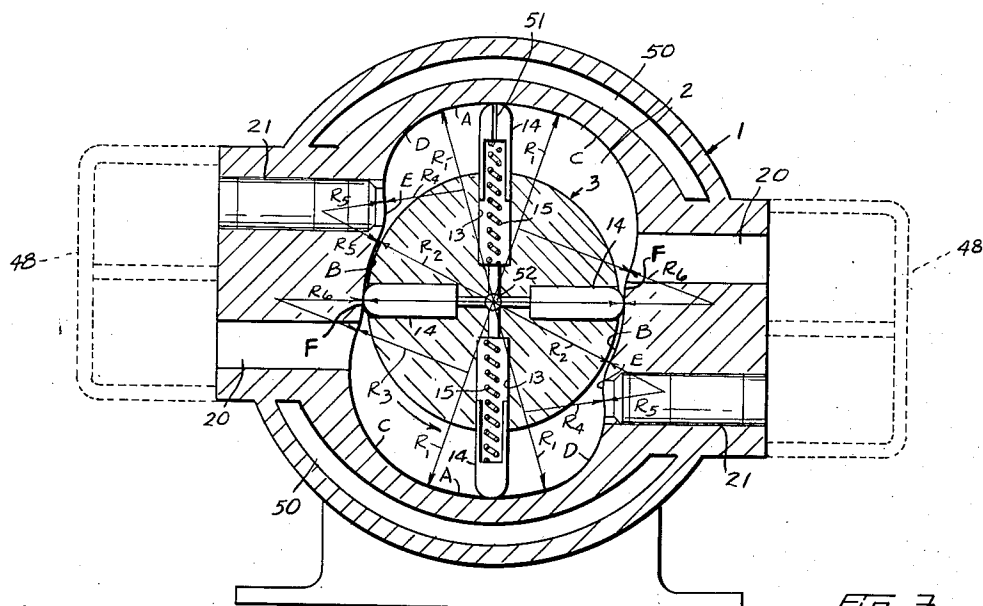


FIG-5

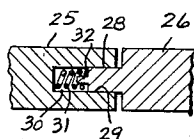


FIG-2

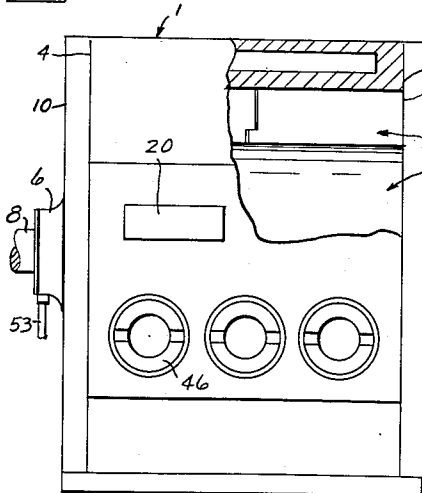


FIG-3

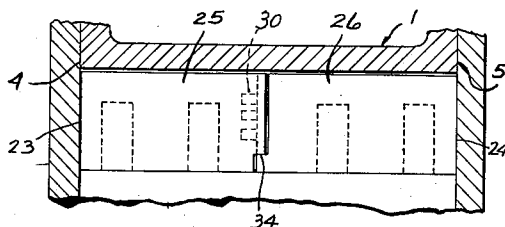
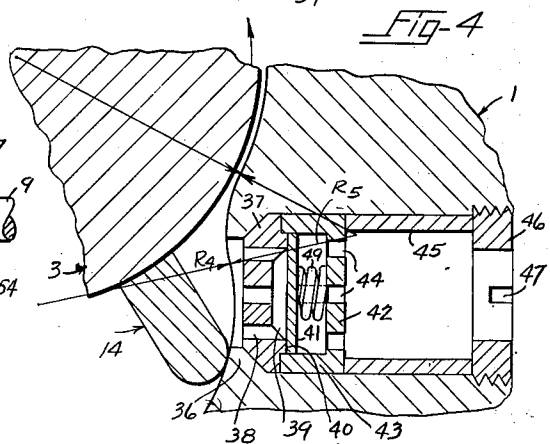


FIG-4



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2,831,631

## ROTARY COMPRESSOR

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2 Claims. (Cl. 230—152)

This invention relates to rotary air compressors and more particularly to a type of rotary compressor in which the housing and rotor are concentric as opposed to the type in which the rotor is eccentric to the housing.

Heretofore, and especially in the case of eccentric rotary compressors, it has not been possible to obtain a very high discharge pressure and such prior art devices are more properly designated blowers for this reason.

The main object of the present invention is to provision of an improved rotary compressor which is adapted to create a much higher discharge pressure than heretofore possible.

Another object of the invention is the provision of a rotary compressor in which the inner contour of the housing is concentric with the axis of the rotor and therefore relatively simple to manufacture.

Still another object of the invention is the provision of a rotary compressor having a novel housing contour adjacent the inlet and discharge parts which permits the device to develop relatively high capacity and pressure.

Other objects and advantages will be apparent from the following specification and the drawings.

Fig. 1 is a semi-schematic vertical cross sectional view through a rotary compressor embodying the present invention.

Fig. 2 is a side elevational view, partly broken away and in section to show internal structure.

Fig. 3 is a fragmentary vertical cross sectional view through the compressor showing the structure of a vane.

Fig. 4 is a greatly enlarged semi-schematic fragmentary cross sectional view through the portion of the compressor adjacent the discharge port showing the check valve and illustrating how the vanes cooperate with the inner sidewalls of the housing to create the discharge pressure.

Fig. 5 is transverse cross sectional view through a vane showing the spring means for urging the two halves of the vane apart axially.

In detail the compressor comprises a substantially cylindrical housing generally designated 1 which is hollowed to provide an inner chamber 2 of uniform cross section throughout the length of the housing (Fig. 1).

A rotor, generally designated 3 is provided within chamber 2 and has a relatively large diameter portion which extends between opposite ends 4, 5 of said housing.

End plates 10, 11 respectively are provided for closing ends 4, 5 and these plates are formed with central bearings 6, 7 respectively which are adapted to rotatably receive the relatively smaller diameter ends 8, 9 of the rotor 3 (Fig. 2).

The rotor 3 is provided with a plurality (preferably four) of axially extending radial slots 13 which are adapted to receive generally rectangular vanes 14 therein. Extending between the bottom of slots 13 and the vanes 14 are springs 15 which urge the vanes 14 radially outwardly at all times into sliding engagement with the inwardly directed surfaces of housing 1 which define the chamber 2.

With reference to the contour of the inner sidewalls

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of the housing 1, it will be apparent that the same is symmetrical about a generally horizontal plane. For this reason like numerals designate similar portions of the contour. At diametrically opposite points on the contour there are provided a pair of relatively short portions A which are concentric with the central axis of rotor 3 and have a radius of curvature  $R_1$  greater than any of the other radii to which the remainder of the contour is formed.

About centrally between the portions A are portions B of radius  $R_2$  which is only a few thousandths greater than the maximum radius of the cylindrical part of the rotor 3.

On opposite sides of each portion A is a relatively long portion C of a radius  $R_3$  somewhat smaller than radius R, and a relatively shorter portion D of a still shorter radius  $R_4$ .

On opposite sides of each portion B are relatively shorter arcuate portions E, F of radius  $R_5$ ,  $R_6$  respectively which, in turn, are respectively adjacent portions D and C. It should be noted that the centers of curvature of portions E, F are spaced outwardly of the inner sidewalls of the housing.

At the junctures between arcuate portions F and C, intake ports 20 are provided through which atmospheric air is drawn into the compressor chamber 2 by the action of vanes 14 in a manner to be described.

At the junctures between arcuate portions D and E, discharge ports 21 are provided through which the air is discharged under pressure from the compressor chamber 2.

It will be apparent from Fig. 1 that adjacent vanes 14 define chambers therebetween which are bounded in a radial direction by the outer periphery of rotor 3 and the inner periphery of the housing 1.

In an axial direction, the compression chambers are defined by the end plates 10, 11. In order to maintain a seal between end plates 10, 11 and the opposite axially directed ends 23, 24 of each vane 14 the vane may be made in two halves 25, 26 as seen in Fig. 3.

Referring to Fig. 5, half 26 may be provided with a tongue portion 28 which is adapted to be slidably received in a complementarily formed groove 29 in the other half 25 in order to effect an air tight seal.

At radially spaced points along the bottom of groove 29, axially opening recesses 30 are provided in vane half 25 for receiving helical compression springs 31 which coact with the tongue 28 to urge the vane halves away from each other and into sealing relationship with the end plates 10, 11. Relatively small projections 32 on tongue 28 may be provided coaxial with the recesses 30 to act as retainers for springs 31.

To insure the vane halves 25, 26 acting in unison as one vane they are preferably provided with an offset portion 34 adjacent the rotor 3 (Fig. 3).

Referring again to Fig. 1 it will be seen that counterclockwise rotation of rotor 3 causes air to be drawn in through inlet ports 20 to each of the four chambers defined by each pair of adjacent vanes 14. This action is due to the fact that the volume defined by each pair of adjacent vanes increases as the leading vane of each adjacent pair moves away from the inlet port.

It will further be apparent that the volume of the chamber within which each charge of air is trapped will remain substantially the same until the leading vane of an adjacent pair reaches one of the discharge ports 21.

In the position of Fig. 1, the retracted vanes are in engagement with the portions B of the housing contour and the extended vanes are each pushing a charge of air ahead of them against the abutment defined by the juncture of portions D and E. This creates a high pressure

at the discharge ports 21 urging the compressed air outwardly therethrough.

Each discharge port 21 is provided with a reduced diameter end 35 adjacent the inner contour of the housing. This end is formed to provide an axially directed frusto-conical shoulder against which a complementarily formed plate 37 abuts. This plate 37 is provided with a plurality of apertures 38 permitting the air within the housing chamber to communicate with the outer side of said plate. Said outer side is recessed as at 39 and is provided with an annular shoulder 40 against which a valve disc 41 seats as seen in Fig. 4.

Spaced axially outwardly of valve disc 41 is a spider 42 having a plurality of axially extending legs 43 along the inner sides of which the valve disc 41 is adapted to reciprocate from the closed position shown to an open position spaced from the valve seat 40. Spider 42 is provided with a plurality of apertures 44 through which the compressed air passes. Spring 49 urges valve disc 41 to closed position at all times.

To releasably secured spider 42 within port 21 a sleeve 45 may be provided which is secured in place by a centrally apertured, externally threaded washer 46 which is screwed into threads formed in the housing 1. A screw-driver slot 47 may be provided in the marginal portion of washer 46 to facilitate removal of the same.

From ports 21 the compressed air may be conducted to any desired accumulator (not shown) by means of piping, or a combined intake and discharge manifold generally designated 48 and indicated in dotted lines in Fig. 1 may be provided.

Radially outwardly of the rotor chamber 2 a cooling water chamber 50 (Fig. 1) may be provided and water may be conducted therethrough in the conventional manner.

Lubrication is readily provided for the inner contour of the housing by radially extending passageways 51 in each vane which communicate with an axially extending passageway 52 in rotor 3. Oil is forced longitudinally through the rotor shaft from one end thereof from an inlet 53 (Fig. 2) out through an outlet 54. It will be noted that the action of the vanes provides a pumping action for forcing oil out through the outer ends of the vanes. Thus, upon inward movement of each vane oil is pumped through the passageway 51 which is desirable because the greatest friction is experienced when the vanes are in engagement with the portions of smaller radius.

It is extremely important to note the shape of the contour adjacent the inlet and discharge ports, and especially the latter. As best seen in Fig. 4, the portion D, having a relatively small radius, causes the vanes 14 to start moving radially inwardly at a relatively rapid rate, especially if the speed of the rotor is about 1000 R. P. M. As seen in Fig. 4, the pressure build-up adjacent the discharge valve is very rapid which is essential. Furthermore, it is seen that there is a relatively abrupt change in direction of the contour at the juncture between portion D and portion E and the center of curvature of the latter is outside of the compressor chamber. Thus in passing from the concentric portion A to the concentric portion B it is seen that the vane traverses a plurality of portions (A, D and E) of respectively diminishing radius with the center of curvature of the last portion spaced outwardly from the contour of the inner sidewalls.

The abrupt acceleration of the vane in passing between the concentric portions A and B is essential, not only to build up the pressure effectively, but also to prevent separation between the vanes and the sidewalls which results in leakage past the vanes. Heretofore devices of this character have been characterized by gradual changes in the sidewall contour which, it was believed, reduced friction and prevented leakage past the vanes during radial movement of the latter. However, I have found that once a vane is started on its radially inwardly move-

ment, as when it starts to traverse portion D (Fig. 1) its inertia effect at high speeds permits the contour to be curved relatively sharply as best seen in Fig. 4. By positioning the center of curvature of portion D at a distance from the axis of the rotor about equal to the outer radius of the rotor 3 optimum build-up in pressure is achieved without jamming the vane against the housing and at the same time avoiding separation of the vane from the housing.

This structure necessitates joining the pressure portion D of the contour and the concentric portion B by a portion (E) which has its radius of curvature spaced radially outwardly from the inner sidewalls of the housing.

It will be understood of course that all adjacent curved portions of the sidewalls are tangent to each other so that the center of curvature of each portion is on a radial line through the center of curvature of an adjacent portion.

If the radius  $R_4$  of the pressure portion D of the contour were considerably increased pressure build-up would, of course, be more gradual, but the acceleration of the vanes in a radially inwardly direction would be that much less and said vanes would have less inertia to carry them past a relatively sharp abutment such as formed by the juncture of portions D and E. For this reason the likelihood of jamming would be increased without benefit of any substantial increase in pressure.

The efficiency of the present invention therefore depends on relatively high speed (1000 R. P. M.) and an abrupt change in contour to achieve the desired high pressure.

That the result obtained is unexpected is indicated by the fact that, from a consideration of Figs. 1 and 4, the vanes would appear to jam against the discharge abutment; but tests have proved that this does not result even at low rotational speeds.

Although the invention is not restricted to particular sizes it has been found that a compressor constructed as shown in Fig. 1 with radius  $R_1$  equal to 3 inches, and radius  $R_2$  equal to 2 inches has a capacity of 120 cubic feet of free air at 120-150 pounds per square inch at a speed of 1000 R. P. M. This is considerably better performance than may be obtained by the "blower" type of eccentric rotary compressors. Furthermore the weight of the device of the above dimensions need not exceed 130 pounds. The "blower" type air compressors cannot achieve the above noted pressure in a single stage.

In the above detailed example the radii  $R_3$ ,  $R_4$ ,  $R_5$ ,  $R_6$  are preferably  $1\frac{3}{16}$ ",  $1\frac{1}{4}$ ",  $1$ " and  $1\frac{3}{8}$ ". A reduction of  $R_4$  will result in too abrupt a change in direction for the radial acceleration of the vanes and if the same is increased beyond  $1\frac{1}{2}$ " the change is too gradual thus preventing the desired pressure build-up.

It will be understood that Fig. 1 discloses two compressors in one housing served by the same rotor and vanes. This structure results in an extremely compact unit with relatively high capacity. With conventional eccentric type compressors pressures up to 40 or 45 pounds are all that can be achieved.

It will be understood, of course, that the close clearance of about .003 inch between the rotor and the adjacent minimum contour B prevents discharge from the pressure side of one compressor unit into the suction of the other unit. Obviously labyrinth grooves and the like may be formed in the abutments to reduce the likelihood of such leakage.

The above described compressor has been tested at lower speeds without a substantial reduction in pressure although the volume is of course reduced.

I claim:

1. In a rotary air compressor, a housing, a rotor supported for rotation about an axis within said housing and provided with a plurality of radially extending vanes slidably disposed in said rotor, said housing having end walls and an inner sidewall, means in said rotor for urging said

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vanes radially outwardly into sliding engagement with said sidewall, said rotor during its rotation successively engaging first, second and third sections of said sidewalls with said sections being arranged concentric relative to said axis, said first and third sections being formed to the same radius and said second section being of a lesser radius, said sidewall being formed to provide gradually merging curves at the junctures between said sections to permit said vanes to pass from one section to the other during rotation of said rotor, an outlet port at the juncture between said first and second sections and an inlet port at the juncture between said second and third sections, the juncture at said discharge port consisting of a first portion adjacent said first section formed with its center of curvature located inwardly of said sidewall and a portion adjacent said second section formed with its center of curvature located outwardly of said sidewall, said discharge port extending circumferentially of said housing a relatively short distance along the extent of said second portion and also extending along the extent of said first portion.

2. In a rotary air compressor, a housing, a rotor supported for rotation about an axis within said housing and provided with a plurality of radially extending vanes slidably disposed in said rotor, said housing having end walls and an inner sidewall, means in said rotor for urging said vanes radially outwardly into sliding engagement with said sidewall, said rotor during its rotation successively engaging first, second and third sections of said sidewalls with said sections being arranged concentric relative to said axis, said first and third sections being formed to the same radius and said second section being of a lesser radius, said sidewall being formed to provide gradually merging curves at the junctures between said sections to permit said vanes to pass from one section to the other during rotation of said rotor, an outlet port at

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the juncture between said first and second sections and an inlet port at the juncture between said second and third sections, the juncture at said discharge port consisting of a first portion adjacent said first section formed with its center of curvature located inwardly of said sidewall and a portion adjacent said second section formed with its center of curvature located outwardly of said sidewall, said discharge port extending circumferentially of said housing a relatively short distance along the extent of said second portion and also extending along the extent of said first portion, a check valve in said discharge port positioned closely adjacent said sidewall for preventing flow of fluid into said housing.

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