

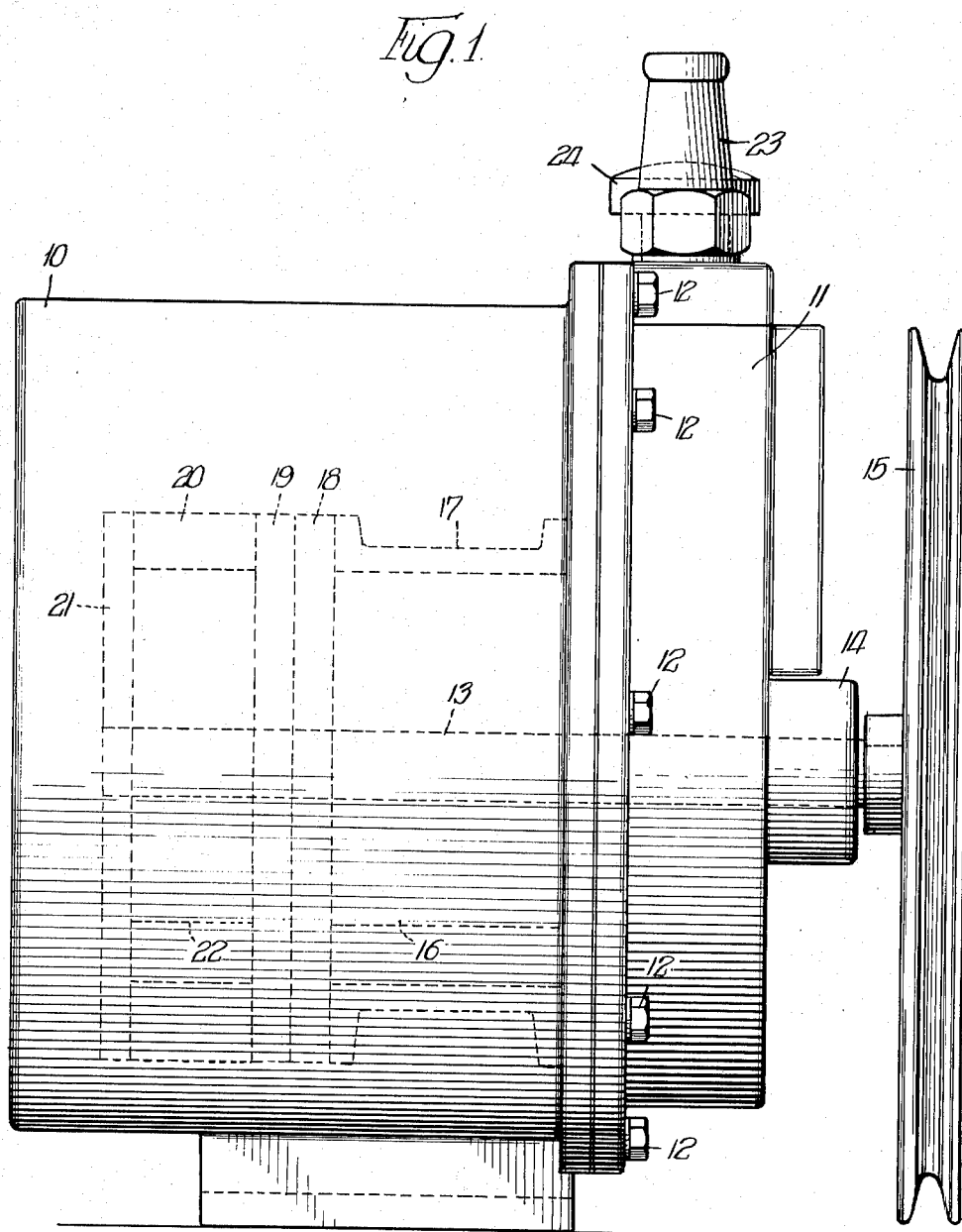
Feb. 15, 1966

J. DUBROVIN
VACUUM PUMP

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3 Sheets-Sheet 1



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Fig. 3

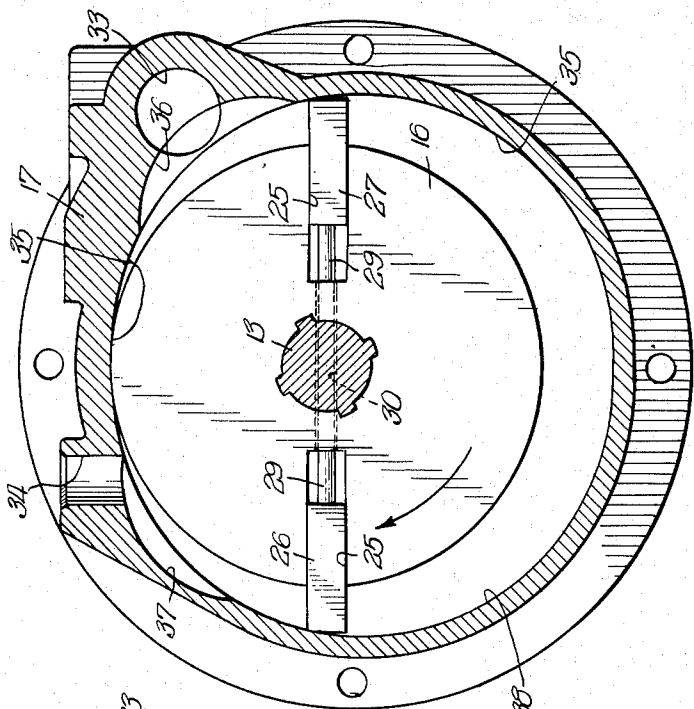
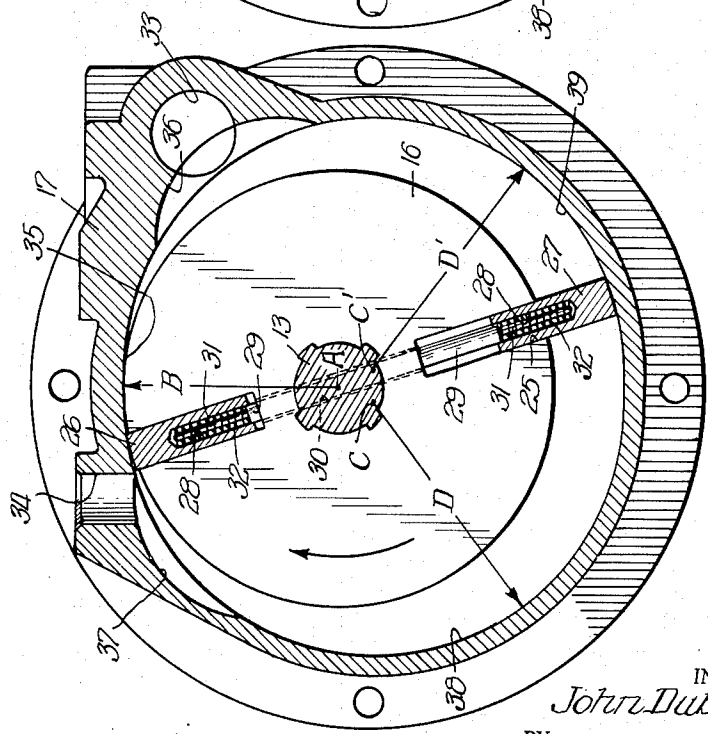


Fig. 2



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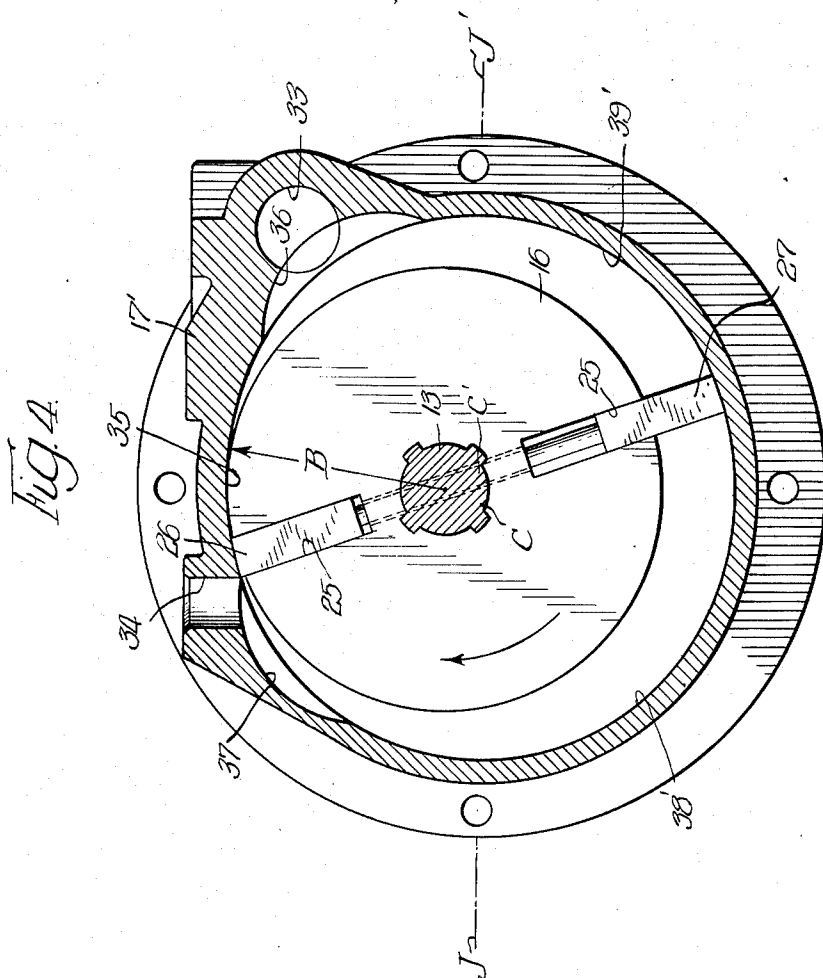
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3 Sheets-Sheet 3



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3,235,172

VACUUM PUMP

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1 Claim. (Cl. 230-153)

The present invention relates to improvements in vacuum pumps, and by way of example only and not as a limitation thereto, the invention is useful in rotary oil sealed vacuum pumps for the production of extremely high vacuums. For purposes of description only, the invention will be described in conjunction with the improvement of a pump of the general type disclosed in my earlier Patents Nos. 2,337,849 and 2,885,143 and my copending application, Serial No. 189,369, filed April 23, 1962, now abandoned, of which this application is a continuation-in-part, wherein the rotor is provided with outwardly spring-pressed vanes in sliding engagement with the surrounding wall of the stator and wherein the rotor is eccentrically mounted in the stator but concentric on the shaft, and turns in an arcuate seat in the stator, which seat has the same radius of curvature as the rotor. It will be understood that the invention is useful in other types of pumps as well.

It is the general object of the invention to increase the efficiency of a pump by incorporating in the same new and improved design features to at least materially reduce vapor formation by frictional volatilization and achieve improved pump starting conditions.

A more specific object of the invention is to provide a new and improved pump or stator cavity design establishing controlled reduction or operational friction and heat generation in the area of contact between the rotor and stator wall.

Still another object is to provide for improved efficiency in vacuum pump operation including starting thereof by use of a specially designed pump cavity which permits higher gas conductance and improved new gas acceptance while controllably reducing vane sealing force applied against the stator wall away from the seat area.

Another object taken in conjunction with the foregoing is to increase the efficiency and capacity of a vacuum pump not only with regard to pumping speeds but also with regard to new gas intake and gas and volatilized vapor discharge.

Other objects not specifically set forth will become apparent from the following detailed description of the invention made in conjunction with the accompanying drawings wherein:

FIG. 1 illustrates a standard type of vacuum pump in elevation, this pump being a two stage unit including a finishing stage and roughing stage generally illustrated in broken lines;

FIG. 2 is a transverse section of one of the stages of the pump of FIG. 1 illustrating in detail the design principles of the present invention;

FIG. 3 is a view similar to FIG. 2 showing the rotor of the stage in a different position to better illustrate certain operational advantages of the present invention; and

FIG. 4 is a view similar to FIGS. 2 and 3 showing a pump with an elliptical cavity.

In the vacuum pumps disclosed in my earlier patents, a Gaede type movement is utilized. Such a movement involves the provision of a seat or seal surface portion of the stator located in the area between the intake and the exhaust and having the radius to the rotor set closely to it with the rotor being mounted on a shaft that is concentric to the arc of the seat. Thus the seat is eccentric to the basic curvature of the cavity surface of the stator,

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the arc of the main curvature being centered a greater distance away from the seat than the radius of the rotor. A pair of oppositely positioned and oppositely acting vanes are spring mounted in the rotor and their outer edges engage the stator cavity surface for pumping action in the known manner. When each vane moves across the seat surface in engagement therewith, the spring acting on the vane is under a sufficient compressive load to exert an adequate force on the vane to force the same outwardly into relatively tight running engagement with the seat surface as well as the opposite vane against the opposite stator wall. As the particular vane moves out of the seat surface area, it moves across the intake opening of the stator cavity which has a radius of curvature falling between the dimension of the major and minor axes, thus resulting in the maintaining of vane compression to a degree substantially equal to that established in the seat area.

In considering the foregoing, it has been found that a vane moving past the intake opening of a stator cavity is still under a relatively great spring load as it is still substantially in a retracted position in its associated rotor guide slot. The spring load directly affects the amount of friction developed between the end of the vane and the engaging stator cavity surface area. It has been found that the frictional forces developed in this area are sufficient to materially increase the temperature of the operating parts with the result that volatilization occurs. The thin film of lubricant applied to the engaging surfaces when subjected to adequate temperature rise will volatilize to form a gas or vapor contributing to raise the internal pressure of the pump. The efficiency limitation, i.e. ultimate lowest pressure, of oil sealed mechanical high vacuum pumps is defined by that point at which the vapor pressure created under operating conditions in the stator cavity at the intake balances against the amount of pressure in the system undergoing evacuation. Obviously, no mechanical pump can pump lower than its own internal pressure. Thus, the extent to which the lubricant vaporizes in the pump cavity constitutes a direct limitation on the efficiency or capacity of the pump.

The present invention is directed to a new and improved design of pump cavity which provides for reduced operational friction and temperature development and improved gas and vapor conductance during operation of the pump. The design concepts of the invention are applicable to a one or two stage vacuum pump primarily of the type described, but not necessarily limited thereto, and for purposes of general identification, FIG. 1 illustrates somewhat schematically a two stage pump. This pump includes a housing formed from an oil case portion 10 sealed with an end plate portion 11. The housing portions 10 and 11 are suitably flanged and interconnected through the flanges by a plurality of fasteners 12.

A rotor shaft 13 extends through the pump housing having its outer end portion suitably journaled in a bearing housing 14 and its inner end suitably journaled in the oil case portion 10 in a manner not shown. The outer end of the shaft 13 carries a pulley 15 for use in operating the pump in the known manner.

Within the oil casing portion 10 of the housing, the shaft 13 has mounted thereon a relatively wide rotor 16 received within the finishing stage cavity of a stator 17. The rotor 16 is mounted concentrically on the shaft 13 with the common centers thereof being offset relative to the center of the stator 17 in the known manner to provide for eccentric operation of the rotor 16 within the stator 17. The shaft 13 extends through a pair of center plates 18 and 19 and through the cavity of the roughing stator 20 of the pump. The innermost end of the shaft

13 is suitably journaled in a pump end plate 21. A roughing stage rotor 22 is mounted on the shaft 13 within the roughing stage stator 20 and operates in the same manner as the finishing stage rotor 16 previously described.

The end plate portion 11 of the housing has mounted thereon an intake nipple assembly 23 which is in communication through suitable passages (not shown) with the finishing stage cavity defined by the stator 17. A pump exhaust valve unit 24 is also carried by the end plate portion 11 and is in communication with the discharge port of the roughing stage cavity defined by the stator 20. A suitable quantity of lubricating oil is carried in the oil case portion 10 of the housing to maintain proper lubrication of the moving parts of the pump and seal the same.

FIGS. 2 and 3 illustrate the special cavity design features of the present invention as well as the advantageous operational aspects of these features. In each view a stage of a vacuum pump is illustrated in cross section, it being understood that while the finishing stage rotor 16 and stator 17 are shown, the same principles of design may be readily utilized in the roughing stage of a pump. The rotor 16 is of known design being mounted on the shaft 13 and being suitably keyed thereto for rotation therewith. Oppositely opening slots 25 are provided in the rotor 16 and receive therein slidable vanes 26 and 27. Each of these vanes is formed with a bore 28 dimensioned to receive therein one end of a pin-like spring holder 29 which slidably extends through a transverse bore 30 in the shaft 13. The spring holder 29 at opposite ends thereof is formed with spring centering pins 31 having mounted thereon the coil springs 32. The centering pins are of a length to terminate short of the inner ends of the vane bores 28 and the springs 32 are seated at opposite ends between the ends of the bores 28 and spring holders 29 and are under compression. The spring mounting of the vanes 26 and 27 is conventional and provides for continuous urging of the vanes outwardly of their respective grooves 25 into outer end engagement with the stator cavity surface.

The stator 17 is provided with an intake opening 33 and an outlet or discharge opening 34 with the latter being arranged for communication with the second stage of the pump. Located between the intake and discharge openings is an arcuate surface portion 35 which is the seat or seal for the rotor 16, and which may be of any suitable length and configuration. The center of the rotor 16 coincides with the center of the shaft 13, this common center being identified by the character A in FIG. 2. The radius B of the seat 35 substantially corresponds to the radius of the rotor 16. During movement of either vane 26 and 27 along the seat 35 the vane is forced into its groove 25 with the actuating spring thereof being compressed. At this stage of the operation the vanes exert a substantial amount of force outwardly against the surface of the stator cavity to form a complete seal therewith. To each side of the rotor sealing seat 35 the stator cavity will preferably include a pair of milled slots or recesses 36 and 37 of known type which are in association with the intake and discharge openings 33 and 34 respectively. These slots establish gas conductance areas to improve gas introduction and discharge into and from the cavity.

The essence of the invention involves the enlarging of the stator cavity immediately to each side of the seat 35 and throughout the remaining internal surface area thereof. The enlarging of the cavity is represented by the surface portions 38 and 39 having radii greater than the radius of the seat 35 as measured from centers which are offset relative to one another and to the center A. A factor in limiting pump efficiency concerns the ability of the pump to perform at a given pumping speed at all pressures from atmospheric to the ultimate degree of vacuum desired. A pump should be provided with ade-

quate conductance space through which the gas being evacuated can move so that efficient evacuation can be obtained even where the starting pressure is rather substantial. In order to materially improve over-all conductance, the cavity of the stator 17 to both sides of the seat 35 including the ports 33 and 34 is enlarged in oval-like shape. The enlarged cavity as viewed in FIG. 2 may be divided into the two surface portions 38 and 39 for purposes of explanation.

The surface portion 38 has a center of curvature C thus providing a radius of curvature D. Similarly, the surface portion 39 is provided with a center of curvature C' thus defining a radius of curvature D'. The center of curvature C is below and to one of the center A of the rotor 16. The center of curvature C' is below and to the other side of the center A as viewed in FIG. 2. The radii of curvature D and D' will preferably be equal. The surface portions 38 and 39 adjacent their intersection will be honed or machined so that they smoothly join one another at the bottom of the cavity. Very little material is required to be removed as the centers are closely spaced, which preferably results in the stator cavity being of true oval shape with the exception of the seat 35. In providing for improved conductance by use of a selectively enlarged cavity, the radii of curvature D and D' are greater than the radius of curvature B of the seat 35. The enlarged surface portions 38 and 39 extend across the ports 33 and 34 into juncture with the seat 35 thus further increasing conductance at the ports.

With the provision of the enlarged cavity portions 38 and 39 to each side of the seat 35 in junction therewith, the vane 36 from its substantially compressed and retracted position as it approaches the intake port 33 along the seat 35 starts to project or extend itself from the groove 25 in response to action of its spring mounting as it enters the enlarged portion 39 as shown in FIG. 3. The radii D and D' of the enlarged portions 38 and 39 are preferably selected to provide for substantially full projection of either vane as it passes therethrough. Under such conditions the compression force exerted by the springs mounting the vanes is at a minimum and the force applied by the vanes against the cavity surface areas of the portions 38 and 39 is normal. Thus there is a substantial reduction in the degree of heat created by friction in the cavity and this reduction, as related to spring force, commences in the immediate area of the intake port 33 and extends beyond the exhaust port 34. With the resultant decrease in heat, there is substantially less lubricant volatilization and the internal pressure of the pump is materially reduced. Accordingly, the efficiency of the pump in drawing a vacuum is materially increased.

With a substantial decrease in volatilization especially at the intake or exhaust ports of the pump, the volumetric efficiency of the pump will be substantially increased as the conductance is materially improved. No substantial quantity of volatilized gas is present adjacent the intake and exhaust and the pump has greater capacity for gas drawn into the same through the intake port 33 from the system undergoing evacuation. It will be noted that the intake port is so located immediately adjacent the seat 35 to provide for progressively increasing projection of each vane immediately following the passing of the intake port. Thus as the vane moves beyond the intake port 33, the friction developed by the vane 26 as shown in FIG. 3 is continually reducing and little, if any, volatilization occurs. Following complete passing of each vane beyond the intake port 33, the intake area will be freer of volatilized vapor for the acceptance of new gases from the system undergoing evacuation.

While a substantial improvement in conductance and pumping efficiency is made available by use of the improved oval-shaped stator cavity design, it has also been found that substantially improved pump starting conditions result. In all oil sealed mechanical vacuum pumps, the most adverse starting conditions exist when the pump,

at rest, has a large amount of liquid lubricating oil present in the stator cavity. This arises most frequently when the pump is allowed to stand with the intake exposed to a comparatively large volume of evacuated space. The oil in the reservoir above the exhaust port being subject to full atmospheric pressure is forced back through the exhaust port by a combination of hydrostatic and atmospheric pressures into the stator cavity. After the stator cavity has been filled with liquid oil, rotor turning requires the maximum starting torque due to the presence of the liquid oil in the stator cavity instead of gas. Until the oil is removed from the stator cavity by pump operation, much of the starting torque applied to the rotor creates hydraulic back pressure against the outer end of the oncoming vane. The hydraulic back pressure forces the oncoming vane radially inwardly against the spring thereof thus allowing the vane to skim over some of the liquid oil that cannot be cleared through the small exhaust port. On succeeding revolutions, as the hydraulic back pressure against the end of the vane exceeds the spring force, each vane gradually reduces the amount of liquid oil in the stator cavity until only the normal lubricating volume maintained by the oil feed system remains in the stator cavity.

With the improved stator cavity design of the present invention, the minimum or reduced vane pressure exerted against the stator cavity outside of the seat results in a material reduction in starting torque requirements in the presence of a liquid oil filled stator cavity. The enlarged stator cavity of the specified oval-shape as stated above, results in the reduction in vane sealing pressure outside of the seat area to a degree that the vane pressure against the stator wall does not exceed that which is basically required to maintain an adequate seal. In the absence of excessive vane pressure against the stator wall outside of the seat area, there is a substantial reduction in vane spring compression resistance to hydrostatic back pressure upon pump starting. The required starting torque is reduced and upon succeeding revolutions of the rotor, the vanes function in the manner described above to ultimately remove the excessive oil from the stator cavity to an extent that only the requisite oil film is retained therein for efficient lubrication. Thus, the improved pump is still capable of operating in the known manner to force excessive accumulated oil from the stator cavity upon starting of the pump, and yet the starting torque required is substantially reduced.

Expressed in terms of general design, the wall of the cavity connecting the opposite sides of the seat 36 including the inlet and outlet ports and the portions 38 and 39, is of oval curvature. The major diameter of the special oval shape can be measured at approximately right angles to the perpendicular radius of curvature B of the seat 35 as viewed in FIG. 2. This diameter is at least equal to, and preferably greater than, the sum of B plus the distance from the center A of the rotor 16 to the opposite wall portion of the cavity such as the juncture of the portions 38 and 39. With this dimensional relation, the spring compression force against each vane 27 and in turn against the stator wall will approach equality generally throughout the oval-shaped portion of the stator between opposite sides of the seat 35.

As an example of the various radii of curvature used in designing the pump cavity, given a circular stator 17 prior to machining into oval shape and having an internal surface radius of 2.187 of an inch, the center A would be established at 0.265 of an inch above the initial given center and seat 35 would be formed with a radius of curvature B of 1.938 of an inch. Thus the radius of the rotor 16 would be set with this radius being slightly under that of the seat for minimum operational clearance. The curvatures of the surface portions 38 and 39 would be defined from centers C and C' located at 0.048 of an inch below the initial given center as the case may be. The radii of curvature D and D' would each be 2.187 of an

inch. The foregoing figures are approximate figures and are given merely by way of example and are not intended to be limiting to the scope of the present invention.

From the foregoing specific example of a new and improved pump design of the present invention, it will be noted that in the position of the vanes at right angles to the center of the seat as shown in FIG. 3, the distance between the sealing tips of the vanes or the length of the major axis of the stator cavity is substantially greater than the distance between the center of the seat and the directly opposite point on the stator cavity.

An embodiment constructed in accordance with this principle is illustrated in FIG. 4. In this embodiment, like reference characters have been used to indicate like parts, and similar or corresponding parts have been indicated by identical reference numerals primed. The wall of the cavity connecting the opposite sides of the seat 35 including the inlet and outlet ports, together with the arcuate portions 38' and 39' are of oval or elliptical curvature.

The major diameter of the specially constructed oval or elliptical cavity can be measured along a line at approximately right angles to the perpendicular bisector or the seat 35 generally along the radius B as was described above in conjunction with FIG. 2. As is fully illustrated in FIG. 4, the major diameters are identified by the lines J and J' extending through the centers C and C'. Hence, the minor diameter will also bisect the seat portion 35. The sum of the locus of points formed by the cavity as measured from the two fixed centers or foci C and C' is constant with the exception of the arc formed by the seat 35. Expressed another way, the cavity assumes an elevational configuration which is of the shape of a figure formed by the intersection of a plane and cone when the plane passes obliquely through the opposite side of the cone. This essentially defines an oval or elliptical form similar to the embodiment shown in FIGS. 2 and 3 which obtains the benefits above in conjunction with the previous embodiment.

In conventional designs, it has been the practice to provide the greatest distance along the lines extending from the center of the seat to the opposite wall or surface portion of the stator cavity as compared with the dimension of the stator cavity measured at right angles thereto. By providing the stator cavity with the special elliptical shape of my invention, material improvements in pump operational characteristics are made possible, such improvements dealing not only with improved conductance and capacity but also permitting a substantial reduction in starting torque requirements.

Obviously, certain modifications and variations of the invention as hereinbefore set forth may be made without departing from the spirit and scope thereof, and therefore only such limitations should be imposed as are indicated in the appended claim.

I claim:

In a rotary oil sealed vacuum pump for producing high vacuums, a stator having a cavity which is provided with an arcuate seat and with inlet and outlet ports at opposite sides of said seat in close proximity thereto, and a rotor including projecting vanes mounted for rotation in said cavity with the center of rotation of said rotor being nearer the seat than the opposite wall of the cavity and coincident with the center of curvature of said seat, the improvement comprising said rotor being provided with a pair of oppositely directed outwardly spring pressed vanes each of which is independently radially movable to maintain continuous sealed engagement with said seat and cavity wall during rotation of said rotor, said cavity connecting the opposite sides of said seat being enlarged co-extensive with and beyond said inlet and outlet ports to provide a general oval shape for increased extension of said vanes during engaging movement at least adjacent said inlet port to reduce the frictional forces created by such engagement, the enlargement of said cavity being

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based on two centers spaced transversely of a line bisecting said seat, when said seat is centrally located between said inlet and outlet ports, said cavity being so shaped that the sum of the locus of points forming the cavity wall as measured from the two fixed centers, is constant from one side of said seat to an opposite side of said seat thereby to create said general oval shape for increased extension of said vanes during rotation movement of said rotor, and the spaced centers being located beyond and oppositely and radially offset relative to the center of radius of said seat and relative to said seat.

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