Oct. 25, 1960 J. C. FISK 2,957,429

AXIALLY SHIFTABLE VANE PUMP

Filed June 1, 1956 3 Sheets-Sheet 1

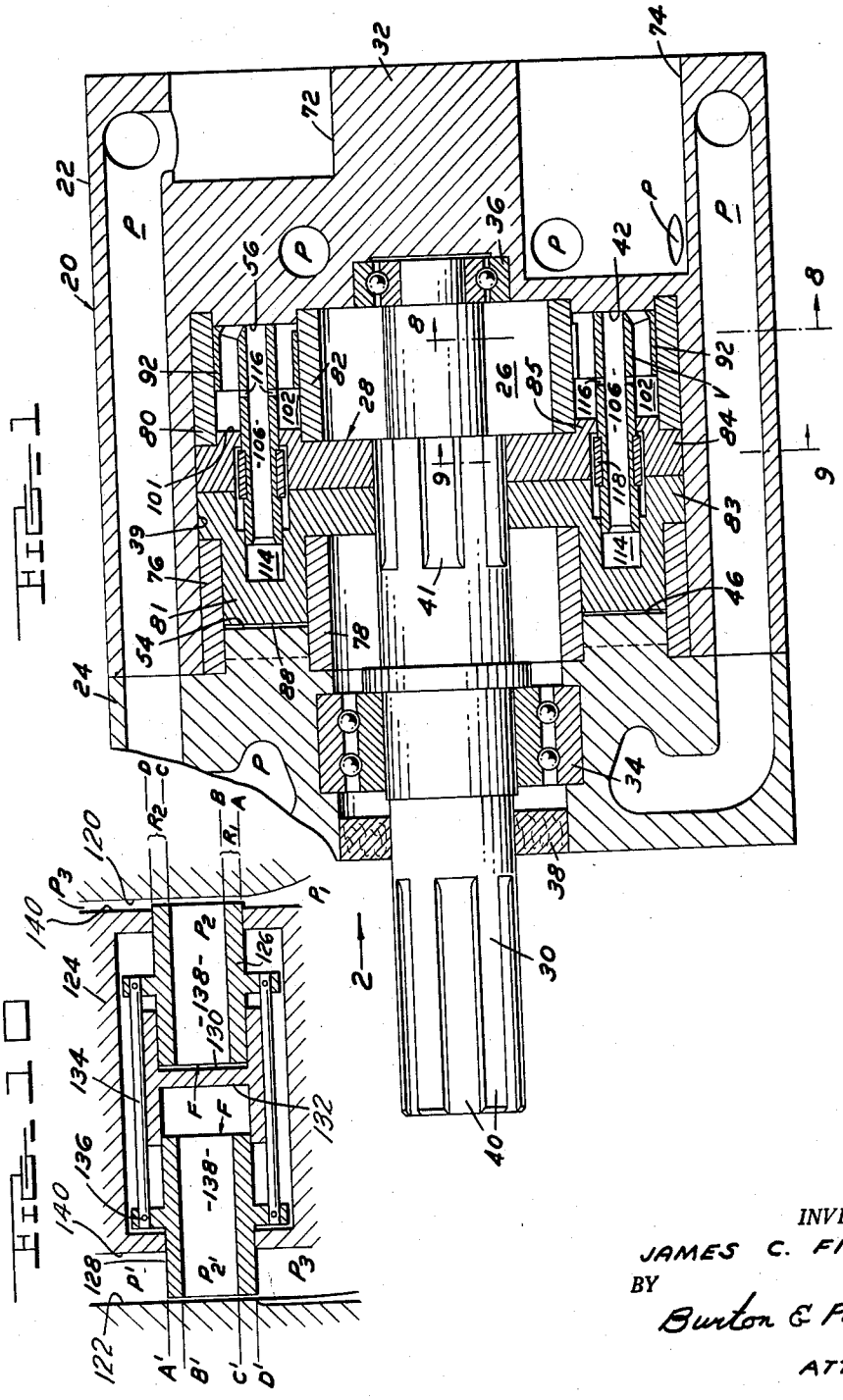


FIG. 1

FIG. 2

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AXIALLY SHIFTABLE VANE PUMP

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FIG. 2

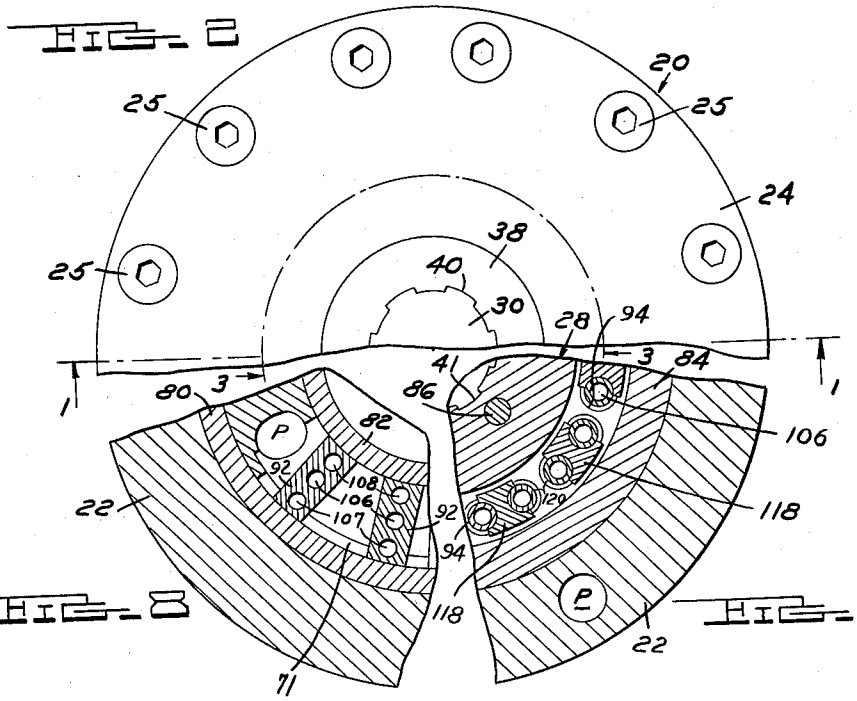
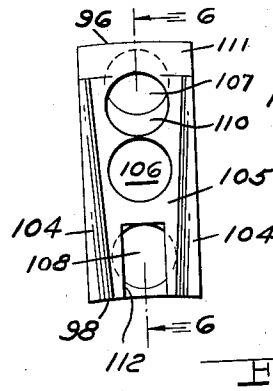


FIG. 8 FIG. 9

FIG. 5



7

FIG. 6

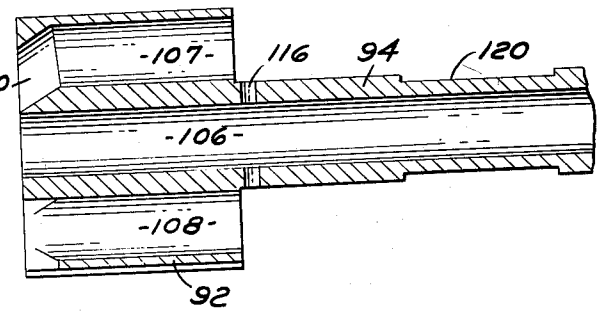
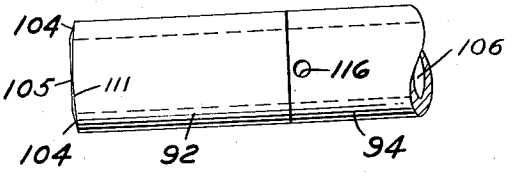


FIG. 7



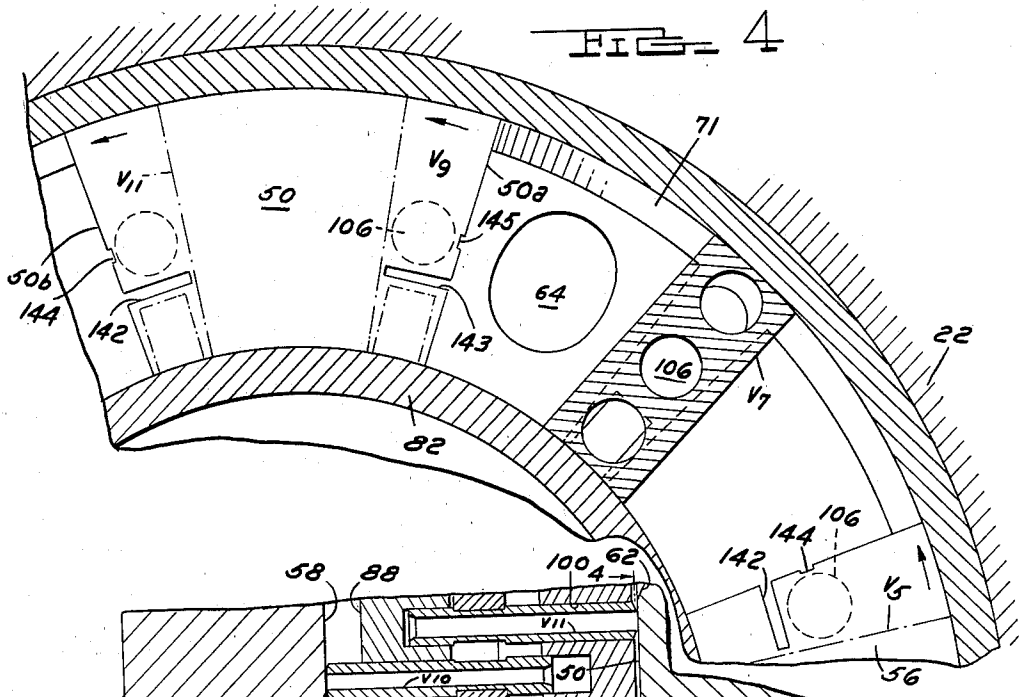
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FIG. 4



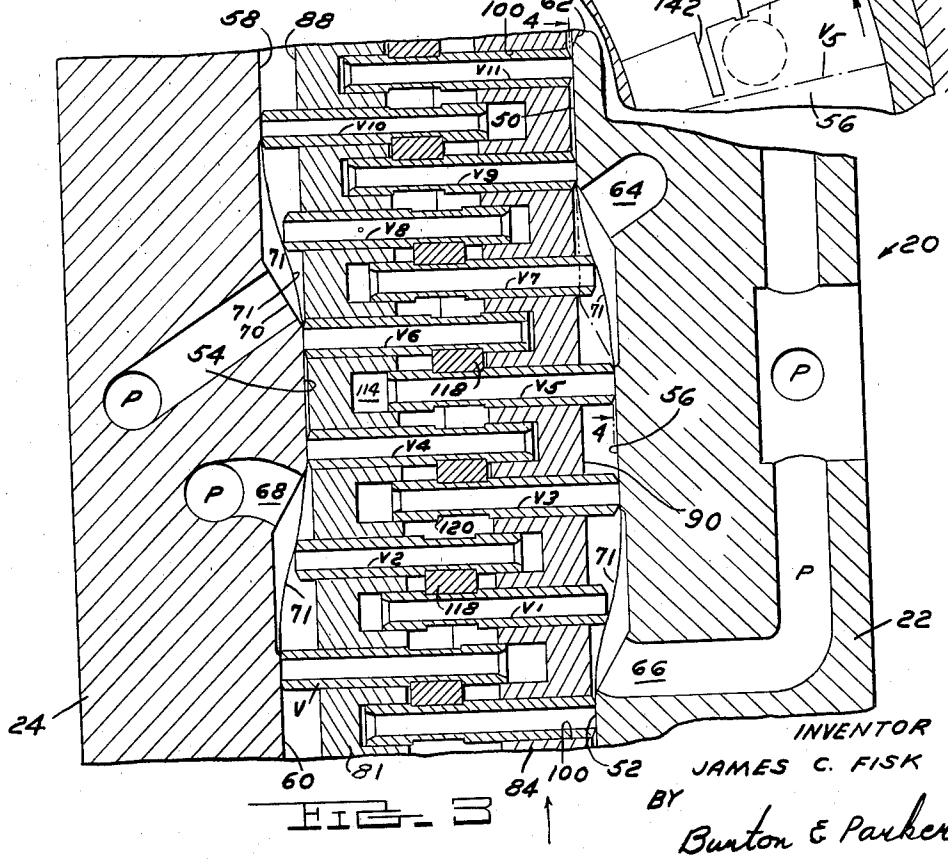


FIG. 3

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2,957,429

AXIALLY SHIFTABLE VANE PUMP

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16 Claims. (Cl. 103—139)

This invention relates to a liquid pump or motor and particularly to that type of pump or motor wherein a plurality of vanes carried by a rotor are rotated in a pumping chamber and during rotation are shifted axially of the rotor to effect the pumping action. For convenience I will term this type of device an axially shiftable vane pump. As will hereinafter appear, the subject invention is as readily suited for embodiment in a motor as in a pump.

The conventional pumps in widespread use wherein a plurality of vanes are carried by a rotor within a pumping chamber with the vanes shiftable in the rotor, universally effect the pumping action by shifting the vanes radially of the axis of rotation of the rotor rather than axially of the rotor. One of the primary reasons for the failure of axially shiftable vane pumps to attain any noticeable commercial success has been the heretofore unsatisfactory solution to the problem of vane wear caused by friction between the vanes and pumping chamber walls. In the invention herein disclosed this problem has been satisfactorily solved and a number of other positive advantages have, in addition, been obtained.

An object of the invention is the provision of an axially shiftable vane pump wherein the friction between the ends of the vanes and the end walls of the pumping chamber is minimized even though fluid pressure upwards of 2,000 pounds or more per square inch is involved. A concomitant object is the provision of a pump which may be constructed at a reasonably low cost, is readily assembled and disassembled, and which will have a long, useful life.

Another object of the invention is the provision of a pump fulfilling the above-mentioned objects and which also has theoretically perfect linearity of flow.

Another object of the invention is the provision of a pump which may also serve equally well as a fluid motor.

Still another object of the invention is the provision of a pump wherein the rotor assembly, which includes the axially shiftable vanes, is dynamically balanced during rotation.

Another object of the invention is the provision of a device of the character mentioned in which the rotor assembly is perfectly hydraulically balanced thereby eliminating wear on bearings and other parts to improve the efficiency of the device.

A further object of the invention is the provision of a pump wherein the vanes carried by the rotor are both hydraulically and mechanically balanced during rotation of the rotor whereby suitable lubricating films may be maintained between the various working parts of the pump.

A still further object of the invention is the provision of a pump or motor of highly efficient operation, of small size in relation to its volumetric and pressure capacities, and which is of smooth performance.

Still a further object of the invention is the provision in a pump of the character mentioned of a rotor assembly having connected pairs of oppositely extending, axially

shiftable vanes, whose end faces ride over staggered gates at opposite ends of the chamber in which such assembly rotates, with means cooperating with each vane to hydraulically bias the vane outwardly to counterbalance the inward thrust on the vane face, and with the mechanical connection between pairs of oppositely extending vanes in simultaneous pumping phases force balancing the outward thrust on each vane by said means, whereby vane and gate face wear is greatly minimized if not entirely eliminated.

A still further object of the invention is the provision of gate faces at opposite ends of the rotor chamber which are so constructed as to provide for equalization of high pressure fluid on opposite sides of a vane as the vane passes onto or off of the gate faces in a high pressure area, whereby the lubricating film between vane and gate faces may be controlled to prevent displacement thereof under the influence of high pressure outward thrust on the vane.

Another object of the invention is the provision of means which serve to pressurize the vane-biasing means simultaneously with the equalization of high pressure on opposite sides of vanes passing off of or onto a gate face at a high pressure area.

Other objects, advantages, and meritorious features in addition to those above enumerated will become more fully apparent by reference to the following description, claims, and drawings wherein:

Fig. 1 is a cross sectional view through a pump embodying my invention taken substantially on the longitudinal axis of the pump;

Fig. 2 is an end view of the upper half of the pump shown in Fig. 1 looking in the direction of arrow 2;

Fig. 3 is a flattened, substantially 180° cylindric section of revolution generated along the line 3—3 in Fig. 2;

Fig. 4 is a cross sectional view taken substantially on the line 4—4 of Fig. 3;

Fig. 5 is an end view of one of the vanes of the pump shown in Fig. 1;

Fig. 6 is a cross sectional view taken on the line 6—6 of Fig. 5;

Fig. 7 is a fragmentary top view looking in the direction of arrow 7 in Fig. 6;

Fig. 8 is a fragmentary cross sectional view taken substantially on the line 8—8 of Fig. 1;

Fig. 9 is a fragmentary cross sectional view taken substantially on the line 9—9 of Fig. 1; and

Fig. 10 is a fragmentary cross sectional view through a rotor assembly in a modified form of the invention.

The invention herein disclosed is embodied in either a pump or motor having a housing in the interior of which is a rotor chamber. The chamber has an encircling side wall and opposed end walls. A rotor assembly is received in the chamber for rotation about the axis of the chamber. The rotor assembly includes a rotor and a plurality of oppositely extending, axially shiftable vanes or fins mounted or received in the rotor and which vanes or fins cooperate with the opposed end walls of the chamber to effect pumping or motor functions. To effect these functions the opposed end walls of the chamber are shaped to define annularly arranged, axially alternately staggered, end wall surfaces. Those surfaces spaced farthest from the opposite ends of the rotor allow the vanes to project farthest from the rotor while the reverse of this is true with respect to the remaining surfaces. It is while the vanes are projecting farthest out of the rotor that effective work is performed by the device, because during this interval the vanes move fluid from an inlet port to an outlet port or, when the device is serving as a motor, the vanes are hydraulically urged from an inlet port toward an outlet port. When the vanes pass over those surfaces lying closest to the rotor, the vanes co-

operate with such surfaces to prevent the passage of any appreciable portion of the fluid between the ends of the rotor and such surfaces, and instead, compel the fluid to pass out of outlet ports communicating with the fluid pressure system in which the device is coupled. These surfaces, therefore, act as fluid barriers and cooperate with the vanes to force or resist fluid flow through the device. Hereinafter such surfaces will be referred to as gates.

As shown in Figs. 1 and 3, the illustrated embodiment of the invention includes a housing **20**. The housing may be formed of two elements **22** and **24** connected together in any convenient manner, as by bolts or the like **25** shown in Fig. 2. The housing is provided with a rotor chamber **26** within which is received a rotor assembly **28** for rotation on the axis of the chamber. Access to the rotor assembly is gained through removal of the element **24** from the element **22**. The element **24** forms an end wall portion for the housing, and a rotor-supporting shaft **30** is received through such end wall. The opposite end wall portion of the housing is indicated at **32** and may be integral, if desired, with the element **22**. The rotor shaft is carried by anti-friction bearing assemblies **34** and **36** mounted in said end walls. Each bearing assembly may include inner and outer races separated by balls in the conventional manner. A fluid seal **38** may encircle the shaft and prevent the escape of the fluid from the housing around the shaft. The outer end of the shaft may be splined as at **40** for connection with a source of power for driving the pump, or for taking power off of the shaft when the device is used as a motor. The inner end of the shaft may be splined as at **41** for connection to the rotor assembly.

The rotor chamber or cavity **26** within the housing **20** has a circular or cylindrical encircling side wall **39** and is provided with axially spaced apart end walls between which the rotor assembly rotates. Each end wall is shaped to define a plurality of annularly alternately arranged, equidistantly spaced blocking and working gates. Such gates are staggered axially of the chamber and are disposed in substantially reverse relation at opposite ends of the chamber. The faces of the gates lie in planes normal to the axis of the rotor chamber. As shown in Fig. 1, a pair of working gates 180° apart are indicated at **42** and **56**, while a pair of blocking gates 180° apart are indicated at **46** and **54**. In Fig. 3, six such gates are shown. Eight gates are provided in this embodiment of my invention, two working and two blocking gates at each end of the rotor chamber. Three blocking gates are indicated in Fig. 3 at **50**, **52**, and **54**, and three working gates at **56**, **58**, and **60**. Fluid pressure inlet and outlet ports open into the pumping chamber between adjacent gates, with five such ports being indicated at **62**, **64**, **66**, **68**, and **70**. Between adjacent gates are vane-shifting surfaces or camming surfaces **71**. Suitable passageways indicated at P extend through the pump housing interconnecting the inlet ports and interconnecting the outlet ports and terminating in primary inlet and outlet ports **72** and **74** opening outwardly of the housing, and to which fluid lines may be connected in any suitable manner for conveying fluid pressure to and from the pump.

Two pairs of rotor rings, **76** and **78**, and **80** and **82**, are rotatably received within the housing in concentric spaced apart relation and cooperate with the rotor assembly and gates to form annular pumping chambers or raceways at opposite ends of and forming portions of the rotor chamber, and within which the pumping or motor functions occur. The rotor assembly includes a two-part radially extending rotor proper formed of elements **83** and **84** which are placed back to back over the splines **41** of the rotor-supporting shaft. The rotor proper extends and rotates between and with the rotor rings with opposite ends of the rotor provided with annular shoulders **81** and **85** extending into the raceways. Suitable securing means such as bolts, the body of one of which is indicated at **86** in Fig. 9, serve to tie the rotor elements **83** and **84** together. As shown in Figs. 1 and 3, the axial end faces **88** and **90** of the shoulders of the rotor are spaced from the blocking gate faces at opposite ends of the rotor chamber. As shown in Fig. 1, the end face **88** of the rotor is spaced from the blocking gate faces **46** and **54**. As is shown in Fig. 3, the rotor end face **90** is spaced from the blocking gates **50** and **54**. The importance of this spacing will become apparent hereinafter.

The rotor assembly also includes a plurality of annularly arranged, equidistantly spaced apart, axially oppositely extending vanes, which are received in the rotor for shiftable movement axially thereof and project at their outer ends beyond the end faces of the rotor. In a pump having eight working and blocking gates, twenty-four vanes are provided so that at least one vane is always in contact with each gate. The vanes are indicated at V in the various figures of the drawings and are shown in detail in Figs. 5, 6, and 7. It will be noted that each vane does not extend completely through the rotor.

As shown in Figs. 5, 6, and 7, each vane includes a head portion or vane proper **92**, and a shank portion **94**. The head portion is generally rectangularly shaped in cross section as shown in Figs. 4 and 5, while the shank may be circular in cross section. The upper and lower edges **96** and **98** of the vane proper may be respectively convex and concave to fit the curvature of the rotor rings. A plurality of rectangularly shaped radially and axially outwardly opening slots **100** are cut in equidistantly spaced relation through each annular shoulder **81** and **83** of the rotor element, each to receive a vane head, such that the vane head is disposed between and abuts the rotor rings as shown in Figs. 1 and 4 and sweeps the bottom wall, viz., the gates, of the raceways. Each vane head shifts axially in its slot within the rotor. From a consideration of the slots **100** as shown in Figs. 1 and 3, it is noted that they are, in effect, blind cavities, the sides of the slots and the rotor rings serving to prevent the escape of fluid out of the slots except through the open end thereof at the end of the rotor. The vane shanks **94** extend through the bottom **101** of each slot so that the fluid cannot escape therethrough except by passing through the vane shank as hereinafter mentioned. Because the slots **100** form blind cavities within which the vane heads shift, when the vane heads are in the extended position outwardly of the slots, a small chamber **102** is formed behind each vane head and within the slot. The purpose of this chamber is considered in detail hereinafter.

As shown in Figs. 5 and 7, the end face of each vane is chamfered as at **104** to form an effective vane face **105** having the shape of a circular ring sector. Extending axially through each vane head are three passageways, **106**, **107**, and **108**. The outer end of passageway **107** is deflected toward the axis of the vane as at **110** to provide a vane face portion **111** which abuts the vane shift cams **71**. The opening of passageway **108** through the end face **105** of the vane head is rectangular in shape as at **112**. It will be noted that there is an effective vane face surface area around each opening and between the openings and the chambers **104**. The passageway **106** extends axially through the shank **94** of each vane to open through the rear end thereof into a blind cavity or cylinder **114** formed in the rotor; one such cavity being provided for each vane. Fluid communication with cavity or cylinder **114** is accomplished only through passageway **106**.

The vane shank **94** in combination with the cavity **114**, and the cavity **102** in combination with the vane head comprise a fluid pressure actuator which counteracts fluid pressure forces on the vane face **105** as hereinafter explained and urges the vane toward the blocking and working gates at the ends of the rotor chamber. These actuators also serve to provide a linear fluid flow characteristic for the pump. Small ports **116** open through the

vane shanks just behind the vane heads as shown in Figs. 1, 6, and 7, to equalize pressure in cavities **102** and **114**.

As best shown in Fig. 3, the vanes extend alternately in opposite directions out of opposite ends of the rotor. Means couple oppositely extending pairs of vanes together for joint shiftable movement. Such means for each pair of vanes includes a key **118** and the shanks of the vanes, and more particularly a relief **120** formed in the shanks and into which the key fits. The axial distance between opposite ends of the rotor chamber or, in other words, the axial distance between gate faces at opposite ends of the rotor chamber, is constant through the circular extent of the chamber. The axial extent of each connected pair of vanes is such that the vane faces bear lightly against the opposite ends of the rotor chamber throughout the rotation of the rotor, and each connected pair of vanes shifts axially as it passes between the faces of the blocking and working gates over the camming surfaces **71**.

To effect the pumping action upon rotation of the rotor, and assuming the rotation to be in the direction of the arrow in Fig. 3, fluid is drawn out of ports **66** and **70** and carried toward the next adjacent port in the direction of rotor movement. The angular width of the gate faces is such, in relation to the spacing of the vanes in the rotor, that a vane is always in contact with each gate face. In Fig. 4 gate face **50** is shown. Its angular extremities are indicated at **50***a* and **50***b*. Thus, the fluid paths at opposite ends of the rotor are intercepted by a vane at all times, and therefore fluid will be forced to move from inlet to outlet ports between adjacent gates. The fluid drawn from port **66** in Fig. 3 is carried across the working gate **56** to the discharge port **64** through which it is discharged from the pump. The blocking gate **50** prevents continued movement of any substantial portion of the fluid with the vanes. While vanes V_1, V_3, and V_5 are moving fluid from port **66** to port **64**, their companion vanes V_2, V_4, and V_6, respectively, are completing their discharge of fluid through port **68** and are moving across the blocking gate **54** to a position to draw fluid out of port **70** and pass it to the next outlet port. Thus it will be noted that opposite ends of each connected pair of vanes are in opposite pumping phases. Because the vanes of each connected pair are laterally, or more accurately, angularly offset, the blocking and working gates at opposite ends of the rotor chamber are similarly offset to accommodate the movement of the offset vanes.

It is now apparent that this pump has pumping facilities at both ends of the rotor and that the pumping action is substantially 90° out of phase as between the two ends of the rotor. With the number of blocking and working gates being equal to four at each end of the rotor chamber and with twenty-four vanes, the pump is of a two-cycle design in that each vane performs two pumping functions per revolution of the rotor, i.e., crosses two working gates and two blocking gates during each revolution of the rotor. With the pump operating at 1200 r.p.m., each vane would shift 80 times per second in carrying out the pumping function. If the pump develops 2200 pounds per square inch, it becomes apparent that not only are the vanes subjected to a very high speed of movement but also very high pressures in and against which they must be shifted. I have found that a lesser number of cycles than two per revolution upsets both hydraulic and dynamic balance while a greater number of cycles would reduce the angular travel allotted for shifting the vanes from maximum to minimum projection out of the rotor and vice versa. However, with slower speed pumps a greater number of cycles may be feasible.

In Fig. 10, an alternative construction is shown. A rotor **124** is disposed between blocking and working gates **120** and **122**, respectively. Vanes **126** and **128** are disposed in axial alignment in blind cavities **130** and **132** in the rotor **124**. The vanes are tied together for joint reciprocation. To schematically illustrate one manner of tying the vanes together, a pair of rods **134** extend through the rotor and are coupled at opposite ends to the vanes as by pins **136**. With the vanes disposed in axial alignment, the blocking and working gates at opposite ends of the rotor chamber may also be directly axially opposite each other. Each vane is provided with an axially extending passageway **138** opening through opposite ends thereof and placing the blind cavities **130** and **132** in communication with the fluid pressure areas at the outer ends of the vanes.

Opposite ends **140** of rotor **124** are spaced from the blocking gates forming small channels between the ends of the rotor and the blocking gates. While only one blocking gate **120** is shown adjacent one vane **126**, it is to be understood that this relative arrangement obtains between each blocking gate and the adjacent end of the rotor.

The passageways **106**, **107**, and **108** through the vanes of the Fig. 1 embodiment, and the passageway **138** through each of the vanes of the Fig. 10 embodiment, together with the blind cavities **102** and **114** in the rotors, perform two important functions heretofore briefly mentioned. The first of these functions is that the passageways and blind cavities cooperate with those portions of the vanes extending into the blind cavities to counterbalance the inward thrust on the vanes caused by the high fluid pressure at the outer end of the vanes thus minimizing the mechanical effort necessary to shift the vanes as they pass between blocking and working gates. This first function will be more fully discussed hereinafter. The second important function is that the passageways and cavities provide means for maintaining complete linear relationship between fluid flow discharge of the pump and rotational velocity of the rotor. Imagine, for purposes of explanation, that the liquid in the pump is a continuous strip of uniform cross section sliding along between the ends of the rotor and the end walls of the rotor chamber at a constant velocity. Then consider the effect of thrusting a solid vane transversely into this strip. The oncoming strip must either back up or slow down to make room for the thickness of the vane. Then imagine that the vane is withdrawn from the strip. The forward portion of the strip must hesitate or slow down to allow the oncoming portion of the strip to close the gap left by the vane. This speeding up and slowing down of the strip, when rapidly repeated, would create damaging and troublesome vibration in the pump and associated hydraulic system.

However, with the vane provided with passageways communicating with a blind cavity or cavities which, together, have a volumetric capacity equal to the fluid displacement of the extended vane, the fluid displaced when the vane is forced into the moving strip is absorbed by the cavities and passageways and the strip moves along uninterrupted. The reverse of this occurs when the vane is withdrawn from the strip. The result is that the vibration which would otherwise result is eliminated and complete linearity of flow is obtained. This analysis, together with the realization that the vanes of each connected pair are in opposite pumping phases explains the reason for hydraulically isolating the cavities and passageways of one vane from those of the companion vane of each pair of vanes.

A problem arising from the structures as thus far described in that when a vane face rides off of or onto a gate face in a high pressure area, i.e., when vane V_2 starts to ride onto blocking gate **54**, or vane V_5 begins to ride off of the working gate **56** onto the camming surface **71**, certain unbalances in the inward and outward thrusts on the vanes arise which must be corrected to avoid vane and gate face wear. Consider vane V_5 in Fig. 3. As this vane rides off of gate face **56**, a progressively increasing amount of the vane face is exposed to high pressure. Because the companion vane V_6 is in a low pressure area, the mechanical tie between vanes V_5 and V_6 will be ineffective to balance the inward thrust caused by the high

pressure at the outer end of vane V_5. If counterbalancing pressure is therefore not early admitted into the blind cavity behind vane V_5, a substantial force will be transmitted to the vane face of companion vane V_6, which is simultaneously passing into a low pressure area at an inlet port of the pump. However, if the counterbalancing pressure is admitted too early to the blind cavity, then the outward force on the vane would substantially exceed the inward force due to limited vane face exposure. In either event, at high operating pressures, the unbalanced forces would greatly exceed the lubricating strength of the hydraulic fluid film between the vane and gate faces, pushing such film aside and permitting the faces to come into direct abutting contact. This condition would result in inefficiency and wear of the faces.

The reverse of the above situation will occur when vane V_2 approaches closely the position of vane V_4 in Fig. 3. As shown, vane V_4 is just completing its passage onto blocking gate face **54**. As vane V_2 begins to ride over gate **54**, a progressively decreasing amount of the vane face is exposed to high pressure while passageway **106** is still open to the high pressure. Therefore the outward thrust on the vane by the fluid pressure actuator, which comprises the blind cavity **114** and the vane shank, and the blind cavity **102** and the rear end of the vane head, will increasingly exceed the inward force on the vane face. With high fluid pressures involved the lubricating film between the vane and the gate faces is displaced resulting in vane and gate face wear and inefficiency of the pump.

As mentioned above there is a space between the ends of the rotor and the blocking gate faces. Therefore, in order to seal this space and prevent the passage of any appreciable amount of fluid pressure therethrough, one vane must always be in abutment with the gate face, and one side of the vane will be subject to high pressure and the opposite side to low pressure.

A satisfactory solution to this problem involves two aspects of the gate face construction. First, the angular width or circumaxial width of each gate face is equal to the angular spacing of the vanes in the rotor plus substantially the angular width of one vane face. Second, circumaxially extending notches or channels **142**, **143**, **144**, and **145** of the character hereinafter particularly mentioned, are cut in opposite extremities of the blocking and working gate faces. The spacing of the vanes in relation to the angular width of gate **50** and the angular distance between such gate and gate **56** is shown in Fig. 4, where vanes V_5, V_7, V_9, and V_{11} are shown. The notches **142** and **143** are cut in opposite extremities of each gate face, as shown in Fig. 4 in blocking gate face **50**, and in one extremity of working gate face **56**. The notches **142** and **143** are located radially where they will not communicate with the passageways through the vanes. The angular length or what may be termed circumaxial length of such notches, as shown in Fig. 4, is slightly less than the width of the vane faces so as not to provide a leaking bypass past both vanes on the gate faces to the low pressure side of the gates when the vanes are in phase-shifting positions, such as at the position shown in Fig. 4. While the notches **142** and **143** are shown of the length above mentioned, it will be understood that they may be of other lengths depending upon the circumaxial width of the gates and the spacing of the vanes in the rotor. The notches **144** and **145** are cut in opposite extremities of each gate, such as shown in Fig. 4, in blocking gate **50**, and in one extremity of working gate **56**. These latter notches are at radial locations such that they will intercept and communicate with a passageway extending through the vanes, such as the passageway **105**.

The relationship in the circumaxial lengths of notches **142** and **144** at the high pressure side of a working gate is such that they simultaneously admit high pressure liquid to the low pressure side of a vane and to the vane blind cavities **102** and **114** immediately prior to or at the time that the vane begins to move off of the working gate

and while all or substantially all of the vane face is still covered by the working gate. The relationship of the circumaxial lengths of notches **143** and **145** at the high pressure or leading edge of each blocking gate is such that high pressure fluid is admitted to the leading side of a vane passing over such edge of the blocking gate throughout the same interval that high pressure fluid is admitted to the vane blind cavities through the passageway **106** in the vane, and the admission of fluid to the leading side of the vane and to the blind cavities is simultaneously discontinued after the vane has passed a predetermined distance over the leading edge of the blocking gate. As a result, the outward thrust on a vane at the high pressure side of the blocking and working gates caused by the admission of high pressure fluid to the vane blind cavities is effectively resisted by a pressurized oil film between the vane face and the gate face. Why this is true will now be explained.

It will be recalled that when a vane is in a working phase, one side is subjected to high pressure and the other side is exposed to low pressure. Thus, if an oil film between a vane end face and a gate face is pressurized, it has a means of escape to the low pressure side of the vane. However, if before or about the time the vane moves off the gate, it is completely surrounded by high pressure, then there will be no path of escape for the lubrication film between the vane and gate faces. In other words the lubrication film itself becomes pressurized to oppose or assist in opposing the force from the vane blind cavities. Thus in this phase lubrication is assured.

Conversely, as a vane rides onto the leading edge of a blocking gate, as vane V_9 in Fig. 3 has just done, from a high pressure area, the pressure at the leading edge of the gate is disassociated from the pressure ahead of the vane such that the lubricating film between the overlapping vane and gate faces may escape under the influence of the outward thrust on the vane by the high pressure in the blind cavities. However, if high pressure surrounds the vane face, the lubricating film cannot escape and will be maintained despite the outward thrust on the vane.

How the notches at the high pressure sides of the blocking and working gates function is best explained by reference to Fig. 4. In Fig. 4 the vanes are moving in a counterclockwise direction as shown by the arrows. As vane V_5 begins to ride off of gate **56**, an increasing amount of the vane face is exposed to high pressure liquid. However, just as this occurs, notch **144** is placed in communication with passageway **106** to admit this high pressure to the vane blind cavity to counteract the increasing inward thrust on the vane face. Simultaneously the blind end of notch **142** is uncovered at the trailing or low pressure side of vane V_5 to admit high pressure fluid to such low pressure side to equalize the liquid pressures on opposite sides of the vane. At this instant the following vane V_3 (see Fig. 3) then begins its working phase as the transition of the fluid pressure from vane V_5 to the leading face of vane V_3 occurs. With high pressure liquid on both sides of vane V_5, the liquid lubricating film between the vane face and gate face cannot be displaced. In addition, the fluid pressure actuator of the vane is functioning to offset the inward thrust on the vane face. As a result of these effects, the lubricating film of liquid is maintained between the vane face and the face of the working gate **56**, and wear on these faces is eliminated at this crucial instant of load transition.

Just the reverse of the above operation of the notches occurs as vane V_9 rides onto blocking gate **50**. As the vane face is exposed a decreasing amount to the high pressure liquid at outlet port **64**, the vane actuator still being in communication with the high pressure, would thrust the vane outwardly with sufficient force to overcome the lubricating liquid film between the vane and gate

faces if it were not for notch **143**. This notch passes high pressure liquid across the gate face beneath the vane face to the leading edge or low pressure side of the vane to maintain high fluid pressure at opposite sides of the vane. With the high pressure at opposite sides of the vane, the lubricating film between the vane and gate faces cannot escape and therefore the lubricating film is maintained between the faces despite the outward thrust on the vane caused by the high pressure fluid in the blind cavities.

While the only useful functions of these gate notches occur at the high pressure sides of the working and blocking gates, the gates are symmetrically notched for purposes of reversibility either in the direction of rotor rotation of change from pumping to motor functions, since either such variation of usage transforms a low pressure to a high pressure chamber.

The notch **142** in gate **50** (see Fig. 4) has the effect of premature transition of the working load from the working vane V_{11} leaving the gate in a counterclockwise direction, to the following vane V_9. Thus, in order for the following vane V_9 to be properly seated on gate **50** in preparation to receive the pressure load existing at the left side of the gate, it is necessary that the gate face be wider than would otherwise be necessary by the circumaxial width of the channel **142**, or essentially the width of one vane face.

One accomplishment of mechanically tying the oppositely extending vanes together in pairs is that as one vane is moved axially inwardly of the rotor in crossing a camming surface from a working to a blocking gate, it will effect an outward movement of its companion vane. In this way the vanes are held in lightly abutting contact with the opposite end walls of the rotor chamber.

In considering another advantage obtained by mechanically connecting oppositely extending vanes, reference is directed at the outset to Fig. 10 because the operational theory of both this embodiment and that of the other figure is the same. If the vanes **126** and **128** were solid instead of being provided with a passageway **138** therethrough, the inward force of fluid pressure on the end faces **105** of the vanes would urge the vanes inwardly and away from the gates. The vanes would would also prevent linearity of flow as above described. To counteract the force tending to urge the vanes inward, a spring might be installed in the blind cavity of each vane, but such spring would have to be quite strong, and when the vanes passed through low pressure areas during rotation of the rotor, the wear on the vane faces against the gates would be excessive.

The U.S. patent to von Pittler 750,219 suggested placing the blind cavity in which each vane shifted in communication with the fluid pressure at the outer end of the vane to effect a counterbalancing of the inward force on the vane face, and proposed the use of a light spring to bias the vane outwardly toward the gates. To accomplish this von Pittler provided a small passageway communicating with the fluid pressure at the outer end of the rotor and communicating with the blind cavity behind the vane.

While one of the problems arising from connecting the blind cavity with the pressure at the outer end of the vane has been hereinabove mentioned together with the solution thereto of the notches **142—145** at the extremities of the gate faces, another problem also arises which I have discovered could be solved by tying pairs of oppositely extending vanes together as above described. This problem was not solved by von Pittler, nor, so far as I am aware, by any of the other prior art inventors. The problem is that during the working phase of each vane, i.e., when a vane separates a high pressure area from a low pressure area, the fluid pressure actuator of the vane tends to thrust the vane outwardly toward the gates with such force that the pressure at the vane face tending to urge the vane inwardly is surpassed, and, in fact, the outward thrust is sufficiently greater than the inward thrust that the lubricating film of liquid between the vane and gate faces is pushed aside permitting the vane face to bear directly against the gate face.

That the fluid pressure actuator could cause an unbalance of forces on a vane was first mentioned in the discussion of notches **142** and **143**. There, however, any problem arising was overcome by the notches. The problem arises again, however, as soon as a vane has moved sufficiently far across a blocking gate face such that the vane face prevents the further passage of liquid through the notch **143** to the leading side of the vane, and a low pressure area exists at the leading side of the vane while a high pressure area exists at the trailing side of the vane. Whenever one side of a vane is exposed to high pressure and the opposite side is exposed to low pressure, the vane may be considered as in a working phase. In addition to the working phase of a vane when it crosses a blocking gate, it is also in a working phase when crossing a working gate. This latter working phase begins as soon as the load on the immediately preceding vane is transferred to the vane in question. For example, vane V_3 in Fig. 3 is in the act of receiving the load from the vane V_5 because the notch **142** at the edge of working gate **56** is being uncovered by the face of vane V_5 such that high pressure liquid can pass thorugh the notch to the trailing side of vane V_5. The working phase of vane V_3 will end when it transfers its load to vane V_1 in the same manner as did vane V_5.

That the outward force on a vane in the working phase may substantially exceed the force against the vane face, tending to urge the vane inwardly, may best be explained by considering first vanes V_2 and V_7 in Fig. 3. The outer ends of these vanes are in the act of passing between working and blocking gates and the fluid pressure at the outer ends of the vanes communicates with the blind cavity behind each vane to hydraulically balance each vane and prevent vane and cam surface wear.

Now consider, however, vanes V_3 and V_4, and V_9 and V_{10} and assume that they are in their working phases such that high pressure areas exist on one side with low pressure on the other side of each. The outer ends of these vanes appear to abut gates. In fact, however, due to working clearances the ends of the vanes do not entirely prevent leakage of liquid between their faces and the respective gates. Consider Fig. 10 where, while the vanes are in axial alignment, the principles under discussion apply equally well. The vane **126**, which we may compare with vane V_9, is shown as spaced slightly from the blocking gate face **120** forming a small channel through which fluid may leak. The spacing has been exaggerated for purposes of clarity of operational description. The length of this channel may be considered the distance A—D, and leakage of high pressure liquid from the high pressure side P_1 to the low pressure side P_3 occurs through this channel. P_3 corresponds to the pressure in the area between the vanes V_9 and V_{11} and the end of rotor **88** and the blocking gate face **50** in Fig. 3. With respect to vane **128**, a channel extends between A^1—D^1 and corresponds to the leakage space between the end of the vane V_{10} and the working gate face **58** in Fig. 3.

With the provision of the passageway **138** through vanes **126** and **128**, each of channels A—D and A^1—D^1 are broken into two channels A—B, C—D, and A^1—B^1 and C^1—D^1, respectively. P_2 represents the pressure in the passageway **138** in vane **126** and P^1_2 represents the pressure in passageway **138** in vane **128**. R_1 and R_2 represent the frictional resistance to flow caused by the walls of the channels.

If the resistance or impedance to fluid flow in channel A—D was all due to friction between the fluid and the channel walls, then a steady gradient drop in pressure could be expected from one end of the channel to the other. By simple hydrodynamic physics, the velocity of flow through a restriction, such as the restriction of the

aforementioned channels, is determined by the differential in fluid pressure at opposite ends of the restriction divided by the resistance to flow through the restriction or, in other words

$$\frac{P_1-P_2}{R}$$

where:

P_1 is the fluid pressure on one side of the restriction, P_2 is the pressure on the other side of the restirction, and R is the resistance to fluid flow through the restriction. Since R_1 and R_2 are substantially equal, by simple algebra it may then be determined that the pressure P_2 in the blind cavity is equal to one-half the sum of the pressures between opposite sides of the vane or

$$\frac{P_1+P_3}{2}$$

The average pressure on the vane face would also equal one-half the pressure differential between opposite sides of the vane, and therefore the vane would be in hydraulic balance.

However, such is actually not the case and this is one reason the von Pittler disclosure does not solve the problem of vane wear. Actually the principal impedance to leakage flow between the vane face and the gate face is due to dynamic reaction of the liquid. The velocity of leakage flow is more affected by the inertia of the liquid. Because the pressure is least where the velocity between the faces is greatest, approximately the total drop in pressure in channels A—B and C—D occurs substantially at points A and C, respectively. Because this is where the greatest pressure drop occurs, instead of being distributed evenly across the vane face, the inward thrust on the vane would not be as great as where the pressure gradient was evenly distributed across the vane face. The pressure in passage **138** will still be approximately one-half P_1+P_3 since the impedances of A—B and C—D are substantially equal. The outward thrust on the vane, however, is the result of an even pressure for all practical purposes across the rear end of the vane. It is elementary that the thrust is equal to the pressure multiplied by the area subjected to the pressure. Therefore the pressure in the blind cavity will actually urge the vane outwardly with a greater force than it is urged inwardly.

The above is true only where the vane is in the working phase. During such time, if the outward thrust represented in Fig. 10 by the arrow F exceeds the inward thrust on the vane end face by a sufficient amount, the liquid film lubricating the vane face and the gate face may be displaced so that the vane rides directly upon the gate face. This, in fact, occurs when substantial fluid pressures are involved. Even though as the channel A—D becomes smaller as the thrust F urges the vane toward the gate face, and therefore the leakage approaches zero, the lubrication seems to be excessively reduced before an equilibrium is established. This is probably due to a slight compressibility of the liquid together with slight mechanical deflections sustaining pressure after leakage has been essentially stopped.

Due to fabricational imperfections and strains in the pump housing resulting from the great fluid pressure forces within the housing, certain areas of the vane faces permit more leakage clearance with the associated gate faces than do other areas. In other words, during the working phase of a vane, the mating vane and gate faces are not perfectly parallel. As a result, the more open areas continue to permit leakage and thus pressurize cavities as the closer fitting areas have forced lubrication out from between the confronting vane and gate faces.

I have discovered that by mechanically connecting pairs of oppositely extending vanes which are both in simultaneous working phases, I can force balance the pair of vanes, because the outward thrust on each vane of a connected pair of vanes counterbalances the outward thrust on the other vane and thereby prevents an outward movement of either vane to reduce working clearance containing lubrication. Such mechanical connection of the vanes has been heretofore described. As a result of this mechanical connection of the vanes, force F in Fig. 10 is balanced by force F^1 such that the vanes of each connected pair are in balance and no vane or gate face wear will result.

It should also be noted that the outward thrust on each vane caused by fluid pressure in the blind cavities serves to hydraulically balance the vane as it rides over the camming surfaces **71**. This is particularly important in the tied vane construction herein disclosed because if the inward thrust on a vane riding over a cam in a high pressure area is not counterbalanced, the load would be transmitted directly through the mechanical tie to the corresponding vane to urge such latter vane against the cam it is riding down in a low pressure, thereby causing cam and gate face wear in the low pressure area.

In a pump of the type herein disclosed where it is desirable to have a maximum fluid capacity both with respect to volume and pressure with minimum leakage across the vane faces, it is desirable and important to have the vanes project outwardly from the rotor as far as possible during their passage over the working gates, subject to certain limiting factors. However, the greater the projection, the greater the shear stresses on the vanes and a necessarily increased thickness of the vanes is required, particularly where the vanes are formed of light weight, low strength materials.

Because of fabricational simplicity, it is desirable to make the vanes with parallel sides as shown in Figs. 5 and 7. But with parallel sides, as the thickness of a vane is increased to gain strength, the angular travel allotted to axial shifting is correspondingly reduced. To compensate for this, the vanes are chamfered as at **104**, as above described, with the lines of intersection of the chamfers of each vane face with the planar end face portion **105**, lying on radii of the rotational axis of the rotor. While these chamfers raise a problem of undesirable end thrust because the chamfer at the high pressure side of a working vane is exposed to the high pressure liquid, the problem is overcome by the mechanical coupling together of pairs of oppositely extending vanes. The companion vane of each pair is subjected to the same undesirable end thrust oppositely directed, and the two thrusts nullify through the mechanical connection between the vanes.

What I claim is:

1. In an axially shiftable vane fluid pressure power converter: a housing defining the rotor chamber, a plurality of annularly alternately arranged blocking and working gates at each end of said chamber with the gates at one end of the chamber having faces opposed to the gates at the opposite end of the chamber, the blocking gates at each end of the chamber being substantially axially aligned with the working gates at the opposite end of the chamber, a rotor disposed in the chamber for rotation about the axis thereof, a plurality of oppositely axially extending hydraulically isolated mechanically connected pairs of vanes mounted in the rotor to sweep said gates, hydraulic biasing means in the rotor for each vane to hydraulically bias the vane outwardly, each vane provided with a passageway opening at the outer end through the end face of the vane and communicating at the inner end with said means, and the marginal trailing edge portion of the face of each working gate and the marginal leading edge portion of the face of each blocking gate provided with a pair of circumaxially extending notches, one of said notches in the working gate disposed to intercept the passageway opening through the end of a vane passing over the margin of the gate and the other said notches of a length sufficient to equalize the fluid pressure on opposite sides of the vane, and one of the pair of notches in the blocking gate disposed to intercept the

passageway opening through the end of a vane passing over such marginal portion of the gate and the other notch of a length sufficient to equalize the fluid pressure on opposite sides of the vane.

2. The invention as defined in claim numbered 1 characterized in that the said pair of notches in each gate are of unequal length with the longer notches being radially located offset the passageway in each vane and the shorter notches being radially located to intercept and establish fluid communication with the passageway in each vane.

3. The invention as defined in claim numbered 1 characterized in that the notches are provided at both the leading and trailing edges of the faces of each gate.

4. An axially shiftable vane pump or motor comprising: a housing defining a cylindrical rotor chamber, a plurality of annularly alternately arranged axially staggered blocking and working gates at opposite ends of the chamber with the blocking and working gates substantially axially opposite each other as between opposite ends of the chamber and each having a flat face, fluid conducting means opening at one end into opposite ends of said chamber and adapted for coupling at the opposite end with a fluid pressure system in which the pump or motor is to be connected, a rotor in said chamber for rotation therein, a plurality of structurally connected pairs of axially shiftable oppositely extending vanes mounted in the rotor with the outer end of each vane having a flat face parallel with and opposed to the flat gate faces and adapted to sweep the flat faces of the gates to effect pumping or motor functions at opposite ends of the rotor, means cooperable with the vanes to shift them axially as they pass between adjacent gates, individual fluid pressure operated balancing means in the rotor operatively connected with each vane and fluid pressure balancing the vane during angular travel of the vane between adjacent gates, and fluid conducting means communicating with said balancing means and the fluid pressure at the outer end of each vane to establish fluid communication therebetween.

5. In an axially shiftable vane fluid pressure power converter: a housing defining the rotor chamber, a plurality of annularly alternately arranged blocking and working gates at opposite ends of the chamber, each gate having a flat face, the blocking gates at each end of the chamber being substantially axially aligned with the working gates at the opposite ends of the chamber, a rotor assembly disposed in the chamber for rotation about the axis thereof with the ends of the rotor opposing the flat faces of the gates, fluid passageways opening at one end into opposite ends of said chamber and opening at the other end outwardly of the housing for communication with a fluid pressure system in which the converter is to be connected, a plurality of oppositely axially extending hydraulically isolated mechanically connected pairs of vanes received in and comprising a portion of said assembly, each vane having a flat end face disposed parallel with and juxtaposing the flat faces of the gates during rotation of said assembly, individual fluid pressure operated balancing means in said assembly operatively connected to each vane to individually bias each vane outwardly toward the opposed end of the chamber, means cooperable with the vanes to shift them axially as they pass between adjacent gates, said assembly provided with first passageway means establishing fluid communication between the biasing means of each vane and the fluid pressure at the outer end face of the vane, and the marginal trailing edge portions of each working gate face and the marginal leading edge portions of each blocking gate face provided with second fluid pressure passageway means at such marginal portions establishing fluid communication between fluid pressures on opposite sides of a vane passing over such portions, and said second fluid passageway means communicating with said first fluid passageway means during angular travel of a vane across said marginal portion of the gates to establish communication with the biasing means.

6. The invention as defined in claim 4 characterized in that the last-mentioned fluid-conducting means comprises a passageway in each vane establishing fluid pressure communication between the outer end of the vane and the balancing means associated with the vane.

7. The invention as defined in claim 4 characterized in that the last-mentioned fluid-conducting means comprises a passageway in each vane opening at one end through the flat end face of the vane and communicating at the opposite end with the balancing means associated with the vane and establishing fluid pressure communication between the fluid pressure acting against the flat end face of the vane and said balancing means.

8. The invention as defined in claim 4 characterized in said gates are provided with means adjacent the trailing edge of each working gate and the leading edge of each blocking gate to bypass fluid pressure from the high pressure side of a vane at such edges to the low pressure side while substantially all of the vane end face confronts the gate, and said means at the trailing edge of the working gates and the leading edge of the blocking gates communicating with the last-mentioned fluid-conducting means while substantially all of the vane end face confronts the gate to place the hydraulic balancing means associated with such vane in communication with the high fluid pressure adjacent the vane end face.

9. The invention as defined in claim 8 further characterized in that said means at the trailing edge of the working gates and the leading edge of the blocking gates is also provided at the leading edge of the working gates and the trailing edge of the blocking gates.

10. The invention as defined in claim 4 characterized in that each connected pair of vanes comprises two individual vanes, coupling means mechanically connecting together the individual vanes of each connected pair for joint shiftable movement axially of the rotor, said fluid pressure operated balancing means for each vane comprises a blind cavity in the rotor within which a portion of the vane is received with such portion and the cavity forming a fluid pressure actuator, and the last-mentioned fluid-conducting means communicates with the balancing means of each vane by opening into the blind cavity associated with the vane.

11. The invention as defined in claim 4 characterized in that said means for shifting the vanes between adjacent gates comprises camming surfaces at opposite ends of the rotor chamber and extending between adjacent gates.

12. An axially shiftable vane pump or motor comprising: a housing defining a cylindrical rotor chamber, a plurality of annularly alternately arranged axially staggered blocking and working gates at opposite ends of the chamber with the blocking and working gates substantially axially opposite each other as between opposite ends of the chamber and each having a flat face disposed perpendicular to the axis of the cylindrical rotor chamber, fluid-conducting means opening at one end into opposite ends of said chamber and adapted for coupling at the opposite end with a fluid pressure system in which the pump or motor is to be connected, a rotor in said chamber for rotation therein, a plurality of vane-receiving cavities in opposite ends of the rotor and each opening at one end axially outwardly of the rotor, a vane received within each cavity for shiftable movement therein axially of the rotor, means within the rotor mechanically connecting together pairs of oppositely extending vanes for joint axial movement, the outer end of each vane having a flat face paralleling and confronting the flat gate faces and adapted to sweep the same during rotor rotation to effect motor or pumping functions at opposite ends of the rotor, means cooperable with the

vanes to shift them axially as they pass between adjacent gates, said vanes of each mechanically connected pair having portions cooperating with the cavities within which the vanes are received to provide fluid pressure operated counterbalancing actuators for each vane to balance the vane during angular travel between adjacent gates, and fluid-conducting means communicating with each actuator and the fluid pressure at the flat vane face to establish fluid pressure communication therebetween.

13. The invention as defined in claim 4 characterized in that said balancing means comprises a cavity in the rotor for each vane and each vane includes a portion received in the associated cavity for shiftable movement therein and cooperating with the cavity to form therewith a fluid pressure operated counterbalancing actuator for the vane.

14. The invention as defined in claim 13 characterized in that said portions of each pair of oppositely extending vanes are mechanically connected together, with such portions cooperating with the cavities to prevent fluid pressure intercommunication between the counterbalancing actuators of connected pairs of vanes.

15. In an axially shiftable vane pump or motor: a housing defining a cylindrical chamber, end walls at opposite ends of the chamber provided with annularly arranged axially staggered flat faces providing a plurality of gates lying substantially perpendicular the axis of the chamber with the faces at one end of the chamber being staggered substantially reverse to those at the other end of the chamber, said end walls provided with fluid pressure inlet and outlet ports between adjacent staggered faces, a shaft extending through an end wall and into the chamber, a rotor mounted on the shaft for rotation between the end walls of the chamber, said rotor provided with a plurality of pairs of oppositely axially extending blind cavities, a pair of oppositely extending vanes received in each pair of cavities for shiftable movement axially of the rotor, means extending through the rotor mechanically coupling each pair of vanes together for joint shiftable movement, the outer end of each vane having a flat face juxtaposing the flat faces of said gates during rotor rotation, means in the rotor chamber cooperating with the vanes for axially shifting the vanes

between adjacent gates upon rotation of the rotor, each vane provided with a passageway opening through the outer end face thereof and extending along the vane and communicating with the blind cavity within which the vane is received to permit pressurization of the cavity with fluid pressure from the outer end face of the vane, and each cavity with its vane received therein communicating only with the fluid pressure at that end of the rotor through which its vane projects.

16. The invention as defined in claim 4 characterized in that the balancing means for each vane comprises a cavity in the rotor in which is received a portion of the associated vane to form a fluid pressure operated actuator, the volumetric capacity of each actuator being equal to the volumetric displacement of that portion of the associated vane extending beyond the end of the rotor, and the last-mentioned fluid-conducting means comprises a passageway in each vane opening at one end into the cavity of the associated actuator and opening at the opposite end through the outer end of the vane to establish fluid communication between the actuator and the fluid surrounding the outer end of the vane.

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