

June 17, 1969

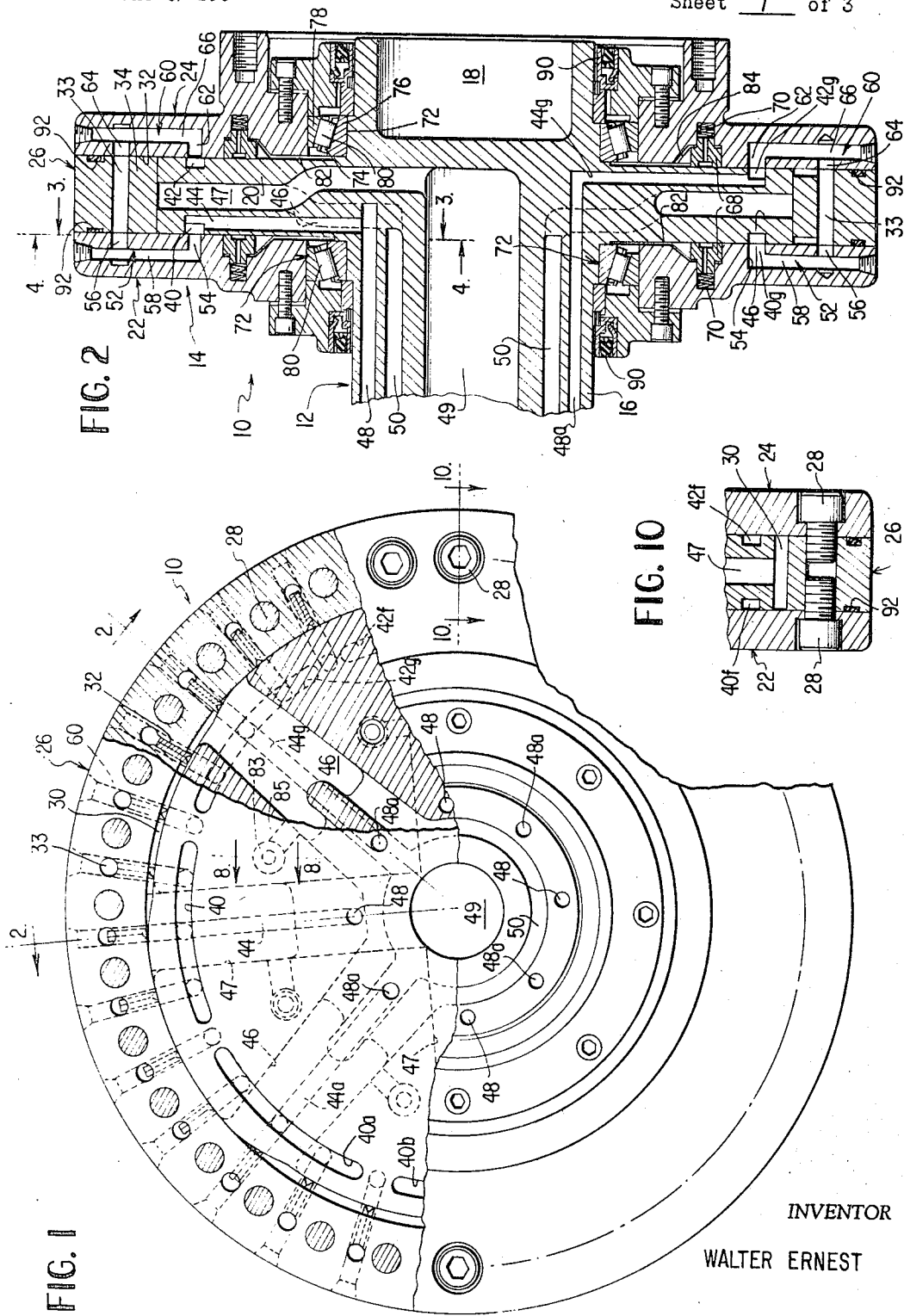
W. ERNST

3,450,004

AUTO-KINETIC WHEEL OR FLUID MOTOR

Filed June 6, 1967

Sheet 1 of 3



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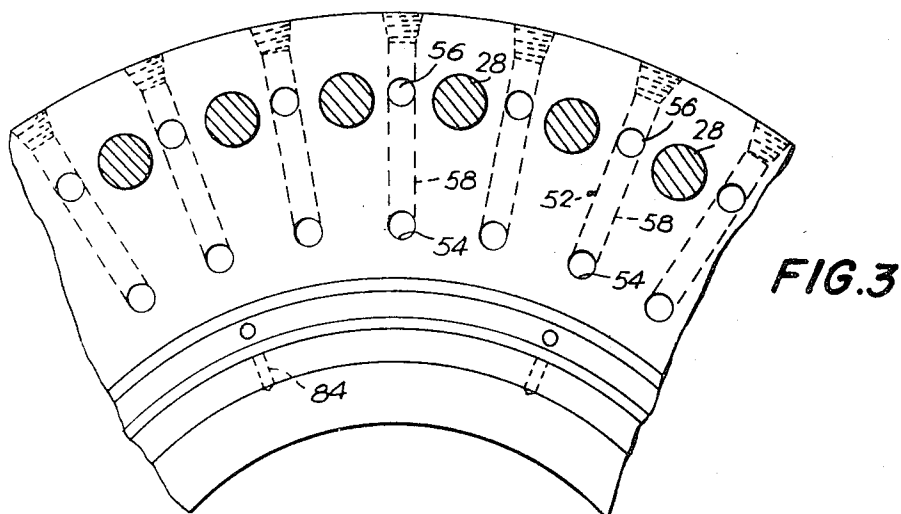
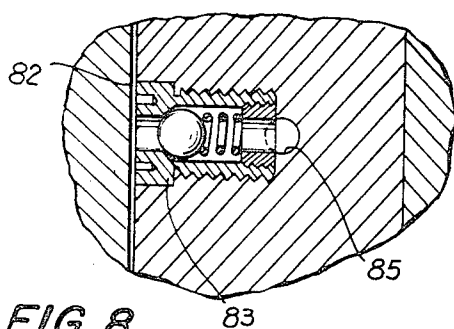
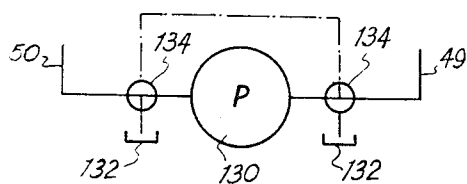
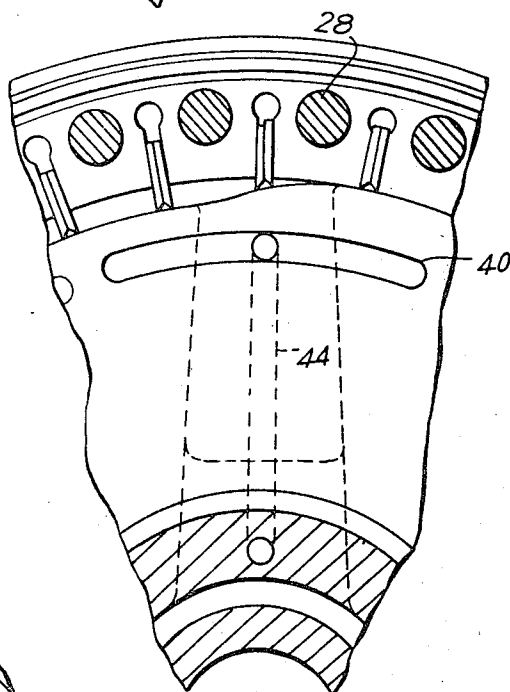


FIG. 4



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FIG. 5

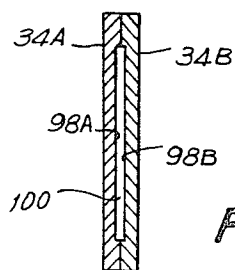
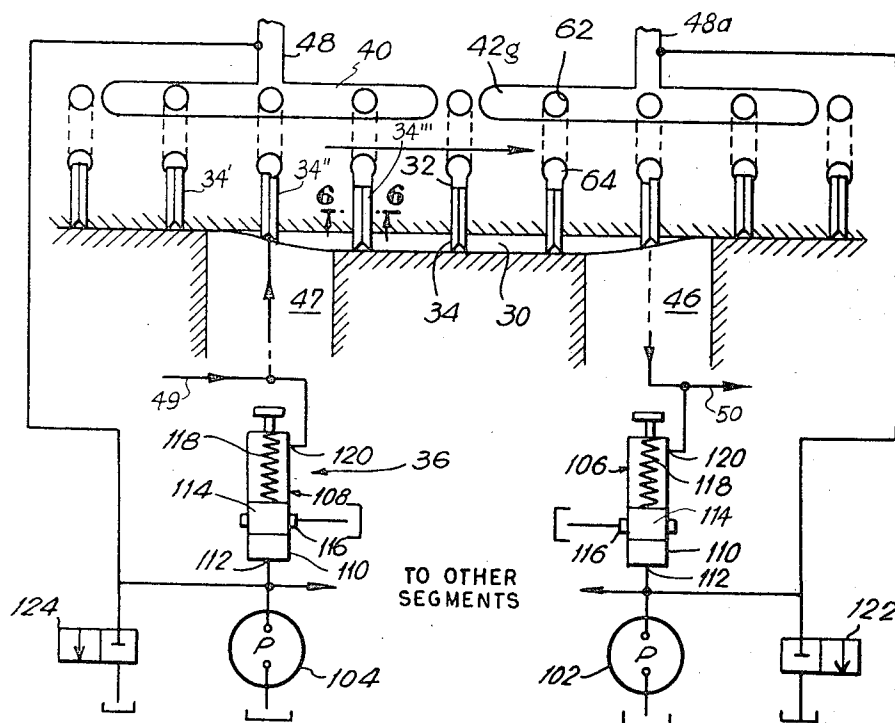


FIG. 6

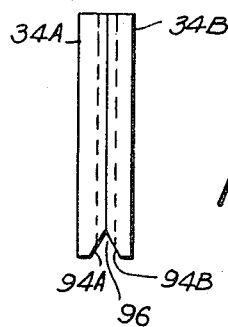


FIG. 7

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AUTO-KINETIC WHEEL OR FLUID MOTOR

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14 Claims

ABSTRACT OF THE DISCLOSURE

The device disclosed herein comprises an auto-kinetic wheel or fluid motor in which hydraulic fluid is fed to a plurality of working chambers along paths generally parallel to the axis of rotation of the wheel. A plurality of retractable vanes are positioned to work in each of the fluid chambers and are positioned in working relation by the use of fluid pressure. When the fluid pressure working against the vanes is relaxed, the vanes are retracted because of centrifugal force operating thereon.

Background of the invention

This invention relates to an auto kinetic wheel or fluid motor for applying torque to a driving wheel. It is conventional in fluid motors to provide retractable vanes working in the fluid chamber. This is customarily done by the use of springs although hydraulic means to force the vanes into the working chamber are known. Exemplary prior art disclosures showing the fluid biasing of vanes are United States Patents 624,462, 2,663,995, and 3,254,570.

The disclosure of Patent 624,462 is believed to be the most pertinent concerning this invention since the device of this patent discloses a driven wheel carrying the sliding vanes. One of the difficulties of a device wherein the sliding vanes are carried by an external rotating member is that centrifugal force tends to move the vanes outwardly away from the central stationary member. It appears that the device of Patent 624,462 is a relatively slow speed device since the wheel is used to drive an endless belt.

In the construction of a relatively high speed wheel, it has been found that the centrifugal acceleration imparted to the movable vanes may amount to many hundreds of G's. It has accordingly been found that relatively high pressures are required to move the vanes inwardly against the stationary member. Because of the relatively high pressure required to move the vanes and the relatively lower pressures needed to rotate the outer wheel about the inner stationary member, it has been found necessary to utilize separate fluid systems to move the vanes and to rotate the wheel. This is in contrast to the devices of the aforementioned patents wherein the same fluid system is used to drive the wheel and to bias the vanes. It will be evident, however, that the devices of the prior art have not appreciated the difficulties accompanying the construction of a high speed auto kinetic wheel or fluid motor wherein the outer rotating member carries the slidable vanes.

Summary of the invention

The present invention provides for the construction of an auto kinetic wheel or fluid motor capable of high speeds by the use of an internal stationary member and an outer rotating wheel. The outer rotating wheel carries a plurality of slidable vanes capable of being positioned adjacent the inner stationary member. A first hydraulic system provides pressurized fluid for rotating the outer wheel about the inner stationary member. A second hydraulic system provides fluid at a higher pressure for

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biasing the slidable vanes toward the inner stationary member.

The present invention relates, generally, to an auto kinetic wheel or fluid motor that is particularly adapted to supply torque to a driving wheel, such as a pneumatic tire, a driving sprocket, an endless track, a rim particularly adapted for use upon a rail, and the like, and even a combination thereof.

More particularly, this invention relates to a fluid motor comprising a stationary portion and a rotatably movable portion, said motor being so constructed and arranged as to form a component part of a driving wheel that is particularly adapted to have any one of a number of diverse types of wheel elements structurally operatively associated therewith, such as a pneumatic tire, a driving sprocket, an endless track, a rim particularly adapted for use upon rails, and the like, and even a combination thereof.

A major problem arising in propelling a vehicle is that the vehicle requires high torque during acceleration and a lower torque, coupled with high speed during high speed travel. When fluid power is utilized to propel the vehicle, using a positive displacement fluid motor or auto-kinetic wheel, the selection of proper torque and power may be accomplished by means of varying the displacement of the fluid motor or auto kinetic wheel. In essence, then, during acceleration, a large displacement is desirable to enable the production of high torque and during high speed travel, a small displacement is desirable to reduce the fluid flow rate while providing high speed.

One previously suggested solution has been to use a variable displacement motor, utilize the full displacement thereof during starting and reduce the displacement of the motor as the vehicle picks up speed. The major objection to this solution is poor efficiency. This is easy to understand, if consideration is given to the fact that the losses in a conventional hydraulic motor are practically constant, regardless of displacement; and at full displacement the efficiency may be quite good, but will deteriorate rapidly as the displacement is decreased.

Accordingly, with a conventional motor, high efficiency may be accomplished during the brief starting period, when it is really not required, but low efficiency may be present during the extended running period, when high efficiency is especially desirable.

It has also been previously suggested to use a plurality of essentially constant displacement motor chambers and to supply hydraulic fluid under pressure selectively to some or all of the chambers by means of appropriate valving to produce the effect of a variable displacement motor means. With this scheme, all of the driving wheels of a vehicle may be operatively connected with fluid motors, and the number of such motors supplied with fluid under pressure be varied to produce effective displacement changes, or single motors having multiple, separately fed chambers may be used. Even this solution, however, under high speed drive produces a low efficiency due to winding losses and turbulent or flow losses within the motor units due to the oil being continuously pumped or circulated within the motor housing.

Accordingly, it is a primary object of the present invention to provide a fluid motor for supplying torque to a driving wheel, said motor being so constructed and arranged as to enable free-wheeling operation thereof without creating any windage, turbulent or flow losses, further enabling all of the supporting wheels of a vehicle to function as driving wheels during high acceleration and to function efficiently as free-wheeling supporting wheels during high speed rotation.

Still other objects and important features of the present invention will be apparent from a study of the specification following taken with the drawing, which together

shown, illustrate, describe and disclose a preferred embodiment or modification of the invention and what is now considered to be the best mode of practicing the principles thereof. Other embodiments or modifications may be suggested to those having the benefit of the teachings herein, and such other embodiments or modifications are intended to be reserved especially as they fall within the scope and spirit of the subjoined claims.

In the drawings:

FIGURE 1 is a side elevation view, partially in cross-section of a fluid motor constructed in accordance with the present invention;

FIGURE 2 is a cross-sectional view taken along the line 2—2 of FIGURE 1;

FIGURE 3 is a partial cross-sectional view taken along the line 3—3 of FIGURE 2;

FIGURE 4 is a partial cross-sectional view taken along the line 4—4 of FIGURE 4;

FIGURE 5 is a schematic illustration of the vane arrangement and the control system therefor of the fluid motor of the previous figures;

FIGURE 6 is an enlarged cross-section of a vane taken along the line 6—6 of FIGURE 5;

FIGURE 7 is an enlarged side elevational view of a vane according to the invention;

FIGURE 8 is an enlarged cross-sectional view taken along the line 8—8 of FIGURE 1;

FIGURE 9 is a schematic view of the primary hydraulic system for driving the wheel of FIGURES 1 and 2; and

FIGURE 10 is an enlarged cross-sectional view taken substantially along line 10—10 of FIGURE 1.

With reference now to the drawing, and particularly FIGURES 1—4 thereof, there is illustrated therein a fluid motor or motorized wheel 10, constructed in accordance with the principles of the present invention, comprising a fixed or stationary generally cylindrical shaft member 12, and a generally cylindrical housing member 14 rotatably mounted upon and about the shaft member 12.

The shaft member 12 comprises a generally cylindrical base portion 16 having a generally centrally disposed blind opening or bore 18 extending axially therein, and a radial flange portion 20 extending radially outwardly from the base portion 16 generally intermediate the ends thereof. The base portion 16 is particularly adapted to be fixedly structurally operatively associated with any suitable fixed support of a vehicle (not shown). This may be accomplished in any suitable manner, and, for example, the portion 16 may be provided with threaded bores (not shown) with which any suitable fastener (also not shown), such as cap screws, bolts, and the like, are particularly adapted to be threadably engageable.

The housing member 14 comprises a first completely annular section or plate 22 which is rotatably mounted upon the base portion 16 of the shaft member 12 adjacent one side or surface of the flange portion 20, thereof, and a second annular section or plate 24, which is rotatably mounted on the base portion 16 adjacent the opposite side or surface of the flange portion. An annular ring or cam ring 26 is disposed or positioned radially outwardly of and in rotatably slidable engagement with the exterior or peripheral surface of the flange portion 20, and is secured to the first and second annular housing plates 22 and 24 in any suitable manner, as through the medium of suitable fasteners 28.

The flange portion 20 is provided on the periphery thereof with a plurality of substantially equi-spaced generally arcuately or circumferentially extending chambers or recesses 30 therein for a purpose to be more fully described hereinafter. While four recesses 30 are illustrated by way of example, it will be understood that the number or quantity thereof may be varied, depending upon desired characteristics of the motor, such as torque output, rotational velocity, and the like.

The ring or cam ring 26 is provided with a plurality of

radially outwardly extending slots or grooves 32 equally spaced about the periphery thereof. The grooves 32 terminate in an enlarged passage 33. A vane 34 is particularly adapted to be radially reciprocally or slidably mounted in each of the grooves 32, and is urged or biased inwardly into mutual cooperative engagement with the peripheral surface of the flange portion 20 by an auxiliary hydraulic system 36 (see FIGURE 5) to be described hereinafter. The vanes 34 may be fabricated of any suitable material, and may be of any suitable construction. In addition, the number or quantity of vanes 34 and corresponding grooves 32 may be varied, as may the number of chambers 30, depending upon the aforesaid desired characteristics of the motor 10. For exemplary purposes only, therefore, there is illustrated thirty-two vanes 34, five of such vanes being particularly adapted to extend to within a corresponding one of the chamber 30 at any one time.

The flange portion 20 is provided with a plurality of first separate annular groove portions 40, 40a, 40b, 40c, 40d, 40e, 40f, 40g in the aforesaid one side or input surface facing the annular housing plate 22. There is also provided in the flange portion 20 a plurality of second separate annular groove portions 42, 42a, 42b, 42c, 42d, 42e, 42f, 42g in the opposite side or the output surface of the flange portion 20 facing the second annular housing plate 24. The shaft member 12 is provided with radially extending bores 44, 44a, 44b, 44c, 44d, 44e, 44f, 44g. The radially outer ends of the bores 44, 44b, 44d, 44f are in communication with the groove portions 40, 40b, 40d, 40f as shown in FIGURES 1 and 2. The radially outer ends of the bores 44a, 44c, 44e, 44g are in communication with the groove portions 42a, 42c, 42e, 42g as also shown in FIGURES 1 and 2. As will be explained more fully hereinafter, fluid is supplied to the bores 44, 44b, 44d, 44f to bias the vanes 34 into the chamber 30 while the vane biasing fluid is allowed to escape through the bores 44a, 44c, 44e, 44g.

Also provided in the shaft member 12 are a plurality of radially extending bores 46 having the radially outer ends thereof in communication with one end of the chambers 30 and similar radially extending bores 47 in communication with the opposite end of the chambers 30. As will be explained more fully hereinafter, fluid is supplied to and exhausted from the bores 47, 46 to propel the annular ring 26 about the stationary shaft member 12.

The shaft member 12 is particularly adapted to be integrally formed with fluid conduits 48, 48a, 49, 50. The inner ends of the conduits 48 are in communication with the bores 44, 44b, 44d, 44f while the inner ends of the conduits 48a are in communication with the bores 44a, 44c, 44e, 44g. The inner ends of the conduits 49, 50 are in communication with the bores 47, 46. It will accordingly be seen that there is provided a shaft member 12 comprising a flange portion 20 that is so constructed and arranged to present integrally fabricated fluid conduits thereby eliminating the need for separate and distinct connections. Each of the conduits 48, 48a, 49 and 50 may be provided with threaded portions for connection with suitable fluid lines (not shown). The separate conduits 48 may be manifolded together as may the separate conduits 48a.

The fluid conduits 49 and 50 are adapted to be utilized to convey fluid under pressure from a supply source, such as a pump 130 and reservoir 132, to a shaft member 12 and to convey or exhaust fluid from the shaft member bores 47, 46 to the reservoir 132 in a manner to be described more fully hereinafter. The fluid conduits 48, 48a are adapted to be utilized to convey auxiliary fluid under pressure from or to a suitable supply source, such as an auxiliary pump, to or from a shaft member 12, and to convey auxiliary fluid to or from the annular groove portions 40, 40a, etc., 42, 42a, etc., through the bores 44, 44a etc., for biasing the vanes 34 as hereinafter described.

Since the fluid conduits 49 and 50 carry the main

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operating hydraulic fluid to the motor chambers 30, it is desirable that the conduits 49 and 50 be of maximum cross-section, to reduce pressure losses therein. Accordingly, and as shown, the fluid conduit 49 is of large generally circular cross-section extending generally axially of the shaft member 12. The fluid conduit 50 is of annular cross-section, extending generally concentric with the fluid conduit 49. This construction enables the conduits 49 and 50 to be maximized consistent with the cross-section of the shaft member 12.

Since the conduits 48 and 48a carry only auxiliary hydraulic fluid, the flow rate therethrough is substantially less than the flow rate through the conduits 49 and 50, and hence, the conduits 48 and 48a are of smaller cross-section.

With continued reference now to FIGURES 1 through 4, the first annular plate 22 of the housing member 14 has formed therein a plurality of radially extending channels 52 of generally U-shaped configuration having the radially inner end thereof in communication with the annular groove portions 40, 40a, etc., in the shaft member flange portion 20. The radially outer end of each of the channels 52 is in communication with one of the grooves 32, to supply and exhaust auxiliary hydraulic fluid under pressure thereto radially outwardly of the vanes 34. As pointed out above, the radial inner end of each channel, as defined by a leg 54 thereof, is disposed in communication with the annular groove portions 40, 40a, etc. The radial outer end of each channel, as defined by a leg 56, is disposed in communication with the enlarged passageway 33 of the grooves 42. The bight 58 of each channel 52 extends between each of the legs 54 and 56. It may now be said that each leg 54 and 56 defines an opening or aperture, one providing communication with the groove portions 40, 40a, etc., and the other providing communication with the respective grooves 32. The bight 58 defines a fluid passage providing communication with and between the apertures or legs 54 and 56. Similarly, the second annular plate 24 of the housing member 14 has formed therein a plurality of radially extending channels 60 having the inner ends thereof in communication with the groove portions 42, 42a, etc. To this end, the plates 22 and 24 may be fabricated substantially identical with one another.

As pointed out above, the plates 22 and 24 are substantially identical. Accordingly, the channels 60 may be of substantially the same configuration as the channels 52, and therefore, are of generally U-shaped configuration equi-spaced about the plate 24, with thirty-two such channels being provided in the illustrated example. The channels 60 comprise a plurality of legs 62 and 64 which extend in a direction generally transversely of the plate, with the leg 62 defining an aperture providing communication with the groove portions 42, 42a, and the leg 64 defining an aperture in communication with the respective grooves 32. The bight 66 of the channels 60, like the bight 58, defines a fluid passage providing communication with and between the apertures 62 and 64.

Referring briefly to FIGURE 2, it will be noted that when biasing fluid is delivered through the conduit 48, the bore 44, the groove 40, the channel 52 and the passageway 33 to bias the vanes 34 downwardly, pressurized fluid is also delivered in the channel 60 and the groove 42. It will be noted that no fluid is exhausted from the groove 42. There accordingly results a counterbalancing force on the right side of the flange portion 20 that offsets the hydraulic force imparted to the left side of the flange portion 20. Therefore an axial pressure balance occurs across the flange portion 20 resulting in an axial pressure balance of the rotating plates 22, 24 and the annular ring 26.

The apertures 56 in the housing plate 22 are located, disposed or positioned adjacent corresponding ends of the grooves 32 in the ring or cam ring 26, and the apertures

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64 in the housing plate 24 are disposed adjacent the opposite corresponding ends of the cam ring grooves 32. Thus, it will be readily seen that, since the plates 22 and 24 the structurally operatively associated with the ring 26, and are, therefore, particularly adapted to be rotatably moved therewith, the apertures 56 and 64 are particularly adapted to be disposed in constant communication with the ends of the grooves 32, and may, therefore, be utilized to supply auxiliary fluid to or to drain or exhaust fluid therefrom.

The groove portions 40, 40a, etc., and 42, 42a, etc., are located in a corresponding one of each face of the flange portion 20 of the shaft member 12, immediately adjacent the inner faces of the plate members 22 and 24. In accordance with the hereinbefore set forth construction, it will be understood that the plate members are rotatably movable relative to the shaft member 12, and, thus, the flange portion 20 thereof. Accordingly, it is desirable that leakage between these relatively movable members be substantially completely precluded, particularly since a further criterion is that substantially all of the pressure fluid be directed into the chambers 30 on the supply side, enabling the presentation of a fluid motor, such as the motor 10, operable with maximum efficiency and substantially nonexistent loss, leakage and wastage of fluid pressure.

To this end, each of the cylindrical plate members 22 and 24 are provided with completely annular seal rings 68, which may be fabricated of any suitable material, such as metal, and may be of any suitable construction. These seal rings are positioned or located in the interior faces of the plates 22 and 24, enabling them to be disposed in sealing engagement with the oppositely disposed exterior faces of the flange portion 20 of the shaft member 12. To insure that the sealing rings are maintained in this relationship with the flange portion, biasing means, such as coil springs 70, are particularly adapted to be positioned axially outwardly of and behind the rings to urge them in a direction towards the flange 20. In addition, the rings 68 are particularly adapted to be positioned or located radially inwardly of and substantially adjacent to the groove 40 to preclude leakage in a direction generally radially inwardly thereof. Any leakage in a direction generally outwardly of the grooves 40 and 42 will eventually find its way into the chambers 30, though such leakage is substantially precluded, as will be discussed hereinafter. Accordingly, it can be seen that the need for manufacturing the various component parts of the fluid motor 10, as hereinbefore described, at very close manufacturing tolerances has been substantially decreased. In fact, only the surfaces that affect the efficiency of the motor 10 must be accurately machined or finished to close tolerances.

As hereinbefore pointed out, the housing member 14 is rotatably structurally operatively associated with, upon and about the shaft member 12. Accordingly, each of the plates 22 and 24 are provided with annular bearing housings 72 within which antifriction bearings 74 are particularly adapted to be disposed, the bearings 74 being particularly adapted to carry or bear the friction introduced by and between the relatively movable members 12 and 14. The bearings 74 may be of any suitable construction, and, in the exemplary form illustrated, the use of a tapered roller bearing comprising inner and outer races 76 and 78, and generally frustoconical rollers 80 disposed therebetween is invoked. It is, of course, both necessary and desirable that the bearings 74 be lubricated. This is accomplished with facility through the medium of the seal ring 68. Thus, fluid under pressure in the chambers 30, and the groove portions 40 and 42 will tend to flow between the plates 22 and 24 and the flange portion 20 of the shaft member 12. The seal rings 68 will tend to preclude leakage therepast, but, under the influence of the springs 70, and by choosing the spring accordingly, will enable a minimal portion of fluid pres-

sure to flow therepast. Hereinbefore, it was pointed out that manufacturing tolerances need not be close. In fact, at the location radially inwardly of the rings 68, the plates 22 and 24 can be so configured and arranged as to present a gap 82. This gap will provide communication between the location of the seal ring 68 and the bearings 74. A fluid pressure that flows past the seal ring 68 into the gap 82 adjacent the input side or surface of the flange portion, will, therefore, flow towards the bearing housing 72 in the plate 22 to lubricate the bearing 74.

It is desirable that all of the leakage of lubricating fluid past the seal ring 68 be directed to the bearings 74 and that the differential area in back be vented and to this end, each of the plates 22 and 24 are provided with vents 84 providing communication between the gaps 82 and the rear of the seal rings 68. Furthermore, single acting or one-way check valves 83 are positioned between the gaps 82 and the conduits 47 and 46, enabling flow to take place therepast in one direction only to the exhaust ones of the conductors 47 and 46 through vents 85. Accordingly, should the pressure of lubricating fluid in the area of the gap 82 be greater than exhaust pressure, the fluid will be directed to the exhaust ones of the conduits 47 and 46, or to those of such conduits connected to the aforesaid reservoir 132.

It will therefore be understood that a fluid motor constructed in accordance with the present invention such as the motor 10, is so constructed and arranged as to make use of substantially all of the supplied fluid pressure, resulting in maximum efficiency, even though close manufacturing tolerances are not required, and yet is self-lubricating. In addition, the aforesaid construction presents an internally case drained system, enabling fluid pressures to be returned to exhaust. And, since the fluid motor 10 is internally case drained, the need for separate, independent, and external conduits from the motor to exhaust is eliminated. Still further, and as hereinbefore pointed out, the percentage of area of the motor that must be accurately machined or finished is reduced.

A seal ring 90 is positioned axially outwardly of each of the bearing housings 72 to preclude the leakage of lubricating fluid outwardly of and from the bearings 74. The rings 90 may be of any suitable construction, and may be fabricated of any suitable material, keeping in mind only that they are subject to wear since they are disposed between relatively movable parts, namely, the base portion 16 of the shaft member 12, and to the plates 22 and 24 of the housing member 14.

In the operation of the fluid motor 10, when a fluid under pressure is supplied to the integral conduit 49, for example by a pump or pumps 130, the fluid flows into the shaft member radial bores 47. The fluid in the bores 47 passes into the chambers 30 adjacent one end thereof (FIGURE 5) and is trapped between the end thereof and the adjacent vane 34. The fluid under pressure thus exerts a force against the end of each chamber 30 adjacent the terminus of the bore 47 and against the adjacent vane members 34 of the ring 26. Since the vane members 34 are mounted only for radial movement on the ring 26 and thus cannot move in circumferential direction relative thereto, the fluid under pressure in each chamber 30 causes the ring 26 to rotate about the shaft member 12 in a generally clockwise direction. Since the housing plates 22 and 24 are secured to the ring 26, they rotate therewith about the base portion 16 of the shaft member 12. As each vane 34 in each chamber 30 passes the bores 47, therefore, a force is applied to the cam ring 26 by the fluid pressure between the vane member 34 and the rearward chamber end thereby resulting in a smooth, positive and continuous rotation of the wheel 14 about the shaft member 12.

In order to drain the fluid in the chambers 30 from the opposite ends thereof, the integral bores 46 and conduits 50 lead to the fluid reservoir 132.

Contemporaneously with the delivery of pressurized fluid through the conduit 49 and the bore 47, pressurized fluid is delivered from pump 104 through the conduits 48 to the bores 44, 44b, 44d, 44f. Accordingly pressurized fluid flows through the grooves 40, 40b, 40d, 40f and the channels 52 into the enlarged passageways 33 of the grooves 32 to bias the vane members 34 into the chambers 30. Referring now to FIGURE 5, it will be seen that the vane members 34', 34'', 34''' are biased toward the stationary flange portion 20 with the vane 34''' constituting the sealing member between the wheel 14 and the stationary flange portion 20. As rotation of the wheel 14 continues, the vane 34''' and its associated channels 52, 60 move out of communication with the groove 40 to a position not in communication with the groove 42g. Since the fluid trapped in the channels 52, 60 and the enlarged passageway 33 associated with the vane member 34 which is not in communication with either of the grooves 40, 42g is at a higher pressure than the fluid in the chamber 30, some drainage of the fluid therefrom occurs through the vane member 34. It will accordingly be seen that the vane member between the grooves 40, 42g is no longer a sealing member and the fluid pressure in the chamber 30 adjacent thereto is substantially the same as the pressure in the exhaust bore 46.

Continued rotation of the wheel 14 about the flange 20 places the channels 52, 60 and the enlarged passageway 33 of the vane member 34 in communication with the groove 42g permitting escape of fluid from the vanes through relief valve 106, auxiliary pump 102. Release of the pressurized fluid from the passageway 33 allows the vanes 34 to retract by the action of the rising cam surface.

It should be noted that an axial balance on the flange portion 20 is maintained throughout the rotation of the wheel 14 thereabout. This is the result of pressurizing and depressurizing oppositely disposed grooves substantially simultaneously. In other words, the groove 40 substantially simultaneously pressurized with the groove 42 with the grooves 40g and 42g being depressurized substantially simultaneously.

In order to insure against leakage of the pressure fluid outwardly of and from the fluid motor 10, a pair of O-rings 92 is provided between the annual housing plates 22 and 24 and the ring or cam ring 26 of the housing member 14. It is noted, however, that any other suitable or conventional type of sealing means could be utilized.

In order to reverse the direction of the instant motor 10, that is, rotate the housing member 14 in a clockwise direction, all that is required is to connect the conduit 50 to the source of fluid pressure or pump 130, and to connect the conduit 49 to the reservoir 132. This will result in fluid being supplied through the radial bores 46 to the opposite end of the chambers 30 and being drained from the aforesaid end of the chambers by the radial bores 47. This reversal of flow in the conduits 49 and 50 can easily be accomplished by connecting them to a conventional reversing valve or valves 134 which are operable to selectively connect the conduits to the fluid pressure source or to the reservoir.

It is obvious that any suitable or desired number of chambers 30 may be provided in the flange portion 20 and that any desired number of vanes, conduits and arcuate groove portions may likewise be provided without departing from the spirit and scope of the present invention, as hereinbefore pointed out. The motor 10 will, of course, still present a smooth, positive continuous and controlled output. Also, any suitable means may be utilized to brake the motor 10 of this invention, including a self-braking system if the fluid circuit be a closed one.

The stationary shaft 12 can be designed as a stub shaft to be bolted to a vehicle frame, or to a vehicle truck, or may be made as a through-axle. Provision may be made, in a conventional manner, to convey the pressure oil to

the shaft member 12 through suitable connections which may be of simple design, because the shaft 12 remains stationary.

With particular reference now to FIGURES 5, 6 and 7, the vanes 34 are co-extensive in length with the axial length of the flange portion 20, and their width is such as to extend nearly the depth of the slots 32.

Each of the vanes 34 consists of two identical members 34a and 34b provided with diverging bevelled edges 94a and 94b to define a small chamber 96 in the inner edge of the vane. Each of the vane members 34a and 34b is provided with a longitudinally extending recess 98a and 98b, respectively, to define therebetween a capillary passage 100 enabling a flow of hydraulic fluid from the bottom of the groove 32 to the chamber 96.

In the standard vane motor, wherein the vanes are mounted on the periphery of the inner member, centrifugal force assists in biasing the vanes outwardly against the cam ring during rotation of the motor. In the subject motor, however, the opposite is true. The centrifugal force tends to move the vanes away from the cam ring, and particularly so at high speeds. The commonly used expedient of pressurizing the vanes with driving fluid is not practical in the subject motor, because the hydraulic pressure required to force the vanes inwardly has to be quite substantial. Accordingly, the auxiliary hydraulic system 36 is utilized to provide a higher fluid pressure in order to force the vanes 34 inwardly into the chambers 30.

The vanes 34, valved as shown, serve as small motor pistons which assist the driving torque of the wheel and, in fact, make up for the torque lost due to the displacement of the vanes, so that the wheel may develop the full theoretical torque.

Referring now more particularly to FIGURE 5, the vane pressurizing system or auxiliary hydraulic system 36 comprises two auxiliary pumps, a pump 104 for providing hydraulic fluid to the pressure segments and a pump 102 for providing pressure fluid to the exhaust segments. Pressure and exhaust arrangements are identical, so that the motor may be reversed.

The auxiliary pumps 102 and 104 apply hydraulic pressure to the radially outward bottom portions of the grooves 32 through pressure relief valves 106 and 108, respectively, which preferably are of the adjustably type. The relief valves 106 and 108, which are illustrated schematically in FIGURE 5, may be of any conventional construction and may, for example, comprise a cylinder 110 provided with an inlet port 112 at one end thereof and a piston 114 slidably mounted therein. An exhaust port 116 is provided in spaced relationship to the inlet port 112 and adjustable biasing means, such as a spring 118 biases the piston within the chamber 110 towards the inlet port 112.

Accordingly, it will be noted that the relief valves 106 and 108 are subjected to two forces. One of such forces is an adjustable spring load produced by the spring 118. This spring load is precisely so adjusted that the pressure produced by the pumps 104 and 102 balances the centrifugal force of the vanes 34 at a preselected maximum speed. Such an adjustment prevents overspeeding of the motor. If an attempt is made to exceed the specified speed, the added centrifugal force of the vanes 34 will cause additional pressure to open the relief valves 106 and 108 and tend to let the vanes 34 retract to let oil leak therepast, which slows down the rotational speed of the motor. The relief valve 106 and 108 therefore, serve as a simple and efficient speed governor.

The chamber region between the inlet port 112 and the piston 114 is connected to the line pressure produced by respective auxiliary pump 102 or 104. On the pressure side, that is, the valve 108, the driving pressure exactly balances the hold-down pressure of the vanes, and the same is true on the exhaust side, so that the vanes will always be in hydrostatic balance, except at the sealing point between pressure and exhaust, where the vanes

would be subjected to excessive sealing pressure. To alleviate this condition, pressure is continuously bled through the capillary slot 100 provided in each of the vanes 34 to produce almost perfect hydrostatic balance.

The other force to which the relief valves 106, 108 are subjected results from the provision of a pair of conduits 120 respectively connecting the upper ends of the valves 106, 108 with the conduits 49, 50. Assuming that the conduit 49 is delivering driving fluid from the pump 130 to the bore 47, the pressure therein is transmitted through the conduit 120 to the relief valve 108 serving the pump 104. The pressure transmitted to the relief valve 108 acts against the piston 114 to increase the pressure delivered by the pump 104 in an amount equal to the pressure in the chambers 30. It will accordingly be seen that the pressure generated by the pump 104 is equal to that pressure to bias the vanes 34 against the centrifugal force produced by rotation and the pressure in the chambers 30. It is evident, therefore, that the vanes 34 on the pressure side of the chambers 30 are in hydrostatic balance.

Assuming that the conduit 50 is exhausting driving fluid from the chamber 30, the pressure therein is transmitted to the relief valve 106 through the conduit 120. Since the pressure in the conduit 50 is nearly atmospheric, the pump 102 generates pressure sufficient only to overcome the centrifugal force imparted to the vanes 34. It will accordingly be seen that the vanes 34 in the exhaust side of the chamber are in hydrostatic balance.

The vanes 34 between the inlet and exhaust sides of the chambers 30 is also in hydrostatic balance because the fluid trapped in the channels 52, 60 and the passage-way 33 associated therewith is at a pressure substantially equal to the pressure necessary to overcome the centrifugal force and the pressure in that segment of the chambers 30.

As hereinbefore pointed out, the vanes 34 act as auxiliary motors, so that the full capability of the motor can be realized. The vane balancing arrangement of FIGURES 5-7 has another very remarkable feature. A two-way vent valve 122 and 124, respectively, are provided connected with each of the vane pressurizing lines. With the vent valves 122 and 124 closed, the vanes 34 are pressurized and will cause the motoring action of the wheel. But, if the vent valves 122 and 124 are opened, the vanes 34 become de-pressurized and centrifugal force causes them to retract into the slots 32 producing an essentially smooth contoured brim rotating about the flange portion 20 with no oil being churned or circulated and accordingly no windage or mechanical losses, other than the losses of the bearings. What was, then, a driving wheel, becomes a mere supporting wheel of high efficiency, provided little drag. Flow through channels 50 and 49 is shut off during that period to avoid loss of pressure fluid.

Accordingly, a plurality of fluid motors or auto-kinetic wheels constructed in accordance with the principles of this invention may be provided on a vehicle, and all of the wheels pressurized to produce high starting torque. Upon achieving a high speed, one or more of the wheels may be disconnected from the hydraulic system, and by merely opening the vent valves 122 and 124, such disconnected motors become highly efficient, low loss supporting wheels permitting efficient, low displacement and high speed operation of the vehicle.

It will now be understood that a fluid motor constructed in accordance with the principles of the present invention, such as the fluid motor 10, is so constructed and arranged as to supply a substantial tractive thrust or torque without necessitating the use of additional mechanisms, such as gears, torque multipliers, and the like. In addition, such a fluid motor is further so constructed and arranged as to apply the aforementioned substantial tractive thrust or torque tangential to its periphery, and substantially completely thereabout, in a direction corresponding to the desired direction of rotatable movement. Furthermore,

such a fluid motor is reversible, and is still further so constructed and arranged as to enable the supply and exhaust of pressure fluid thereto and therefrom through the medium of fluid pressure conduits integral with a stationary portion thereof, eliminating the need for one or more direct connections to a rotatably movable portion thereof. Still further, a fluid motor constructed in accordance with the present invention, such as the fluid motor 10, is so constructed and arranged as to direct substantially all of the pressure fluid from a supply to the area or vicinity of a plurality of pressure chambers, substantially eliminating loss or wastage of said pressure fluid, and a resulting decrease in efficiency, and particularly in torque output, said fluid motor being so constructed and arranged as to enable a certain portion of said pressure fluid to be directed to locations requiring lubrication, and further enabling said pressure fluid to satisfy all of the requirements of propulsion and lubrication without necessitating the use of independent sources of supply and lubrication. Moreover, a fluid motor constructed in accordance with principles of the present invention, such as the fluid motor 10 is so constructed and arranged as to enable efficient free wheeling operation thereof without necessitating the use of additional mechanisms, such as clutches, and the like, and to provide simple and effective automatic governor or speed controlling action. Additionally, the driving wheel itself is structurally operatively associated with the fluid motor in an extremely simple manner, thus facilitating the assembly and disassembly thereof with respect thereto.

Certain directional terminology, such as "inwardly," "outwardly," "adjacent," and the like, has been used in the foregoing description to facilitate an understanding of the present invention. This relative terminology is intended in its normal and accepted sense, and, therefore, is to be given the broadest possible interpretation, and construction, particularly if and when used in the ensuing claims. It is, however, not in any way to be considered limiting.

While the invention has been shown, illustrated, described and disclosed in terms of embodiments or modifications which it has assumed in practice, the scope of the invention should not be deemed limited by the precise embodiment or modifications herein shown, illustrated, described or disclosed, such other embodiments or modifications intended to be reserved especially as they fall within the scope of the claims here appended.

What is claimed is:

1. A fluid motor comprising:
 - a stationary flange portion having drive fluid conduit means and first auxiliary fluid conduit means including supply means and exhaust means;
 - a housing positioned on the flange portion for rotation thereabout, the flange portion and the housing cooperating to provide a plurality of working chambers in communication with the drive fluid conduit means;
 - a plurality of vanes slidably carried by the housing and extending generally radially inwardly therefrom for movement into the working chamber;
 - the housing providing second auxiliary fluid conduit means communicating with the slidable vanes and alternately communicating with the first auxiliary supply means and the first auxiliary exhaust means for cyclically advancing and enabling retraction of the vanes during rotary movement of the housing about the flange portion;
 - a drive fluid supply system for delivering pressurized fluid at a first pressure to the drive fluid conduit means; and
 - an auxiliary fluid supply system for delivering pressurized fluid at a second pressure higher than the first pressure to the first auxiliary fluid conduit means.
2. The fluid motor of claim 1 wherein the secondary auxiliary fluid conduit means extends

from one side of the stationary flange portion to the other side thereof.

3. The fluid motor of claim 1 wherein the first auxiliary fluid conduit means comprises
 - a plurality of first grooves in one side of the stationary flange portion and spaced generally circularly about the axis of rotation;
 - a plurality of second grooves in the opposite side of the stationary flange portion and spaced generally circularly about the axis of rotation, the second grooves being generally aligned with the first grooves;
 - auxiliary fluid delivery passageway means in communication with alternate ones of the first grooves; and
 - auxiliary fluid exhaust passageway means in communication with alternate ones of the second grooves, the alternate ones of the second grooves being circumferentially offset from the alternate ones of the first grooves; and
- the second auxiliary fluid conduit means comprises flow path means providing communication between the generally aligned first and second grooves.
4. The fluid motor of claim 1 further comprising means for diverting pressurized auxiliary fluid away from the first auxiliary fluid conduit means.
5. A fluid motor comprising:
 - a stationary flange portion,
 - a housing portion on the flange portion for rotation thereabout, the flange portion and the housing portion cooperating to provide a plurality of working chambers;
 - a plurality of vanes slidably carried by the housing and extending radially inwardly therefrom for slidably engaging the flange portion;
 - a primary system for delivering pressurized fluid to the working chambers having a first source of pressurized fluid at a first pressure;
 - an auxiliary system for delivering pressurized fluid to the vanes for biasing the vanes toward the stationary flange portion comprising
 - a separate second source of pressurized fluid at a higher pressure than the first pressure.
6. The fluid motor of claim 5 wherein the auxiliary system is arranged to substantially simultaneously deliver pressurized fluid to opposite sides of the stationary flange portion.
7. The fluid motor of claim 5 wherein the auxiliary fluid system comprises
 - a pressure relief valve operable upon the increase in pressure above a predetermined value, to bypass at least a portion of the auxiliary fluid away from the vanes.
8. The fluid motor of claim 7 wherein the pressure relief valve comprises means for adjusting the predetermined value.
9. The fluid motor of claim 7 wherein the auxiliary fluid system comprises
 - conduit means leading to the vanes; and
 - a conduit connecting the conduit means and the pressure relief valve to prevent bypassing of the auxiliary fluid until the pressure in the auxiliary fluid system exceeds the sum of the predetermined pressure value and the pressure in the conduit means.
10. A fluid motor comprising:
 - a stationary flange portion having drive fluid conduit means and first auxiliary fluid conduit means;
 - a housing positioned on the flange portion for rotation thereabout, the flange portion and the housing portion cooperating to provide
 - a plurality of working chambers in communication with the drive fluid conduit means;
 - a plurality of first grooves on one side of the stationary flange portion and spaced generally circularly about the axis of rotation;
 - a plurality of second grooves on the opposite side of

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the stationary flange portion and spaced generally circularly about the axis of rotation;
 the first auxiliary fluid conduit means having
 auxiliary fluid delivery passageway means in communication with alternate ones of the first grooves, and
 auxiliary fluid exhaust passageway means in communication with alternate ones of the second grooves, the alternate ones of the second grooves being circumferentially offset from the alternate ones of the first grooves;
 the housing providing second auxiliary fluid conduit means providing communication between the generally aligned first and second grooves;
 vane means, carried by the housing for movement into and out of the working chambers, in communication with the second auxiliary fluid conduit means; and means for delivering pressurized fluid to the drive fluid conduit means to rotate the housing and to the first auxiliary fluid conduit means to position the vanes.
 11. The fluid motor of claim 5 wherein the auxiliary system comprises means for cyclically delivering and exhausting pressurized fluid from the separate second source to and from the vanes in response to the rotation of the housing.
 12. A rotary fluid motor comprising:
 a stationary hub;
 a housing rotatably mounted on the hub and defining therewith a working chamber;
 vane means slidably carried by the housing for generally radial movement into and out of the working chamber;
 fluid pressure means in communication with the outer end portion of the vane means for advancing the same inwardly into the working chamber and in communication with the working chamber to drive

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the vane means and housing about the hub; the fluid pressure means comprising
 means for cyclically increasing and decreasing the fluid pressure acting against the vane during rotation of the housing to biasingly advance and centrifugally retract the vane means.
 13. The rotary fluid motor of claim 12 wherein the working chamber comprises a rotationally rearward portion; and
 the pressure decreasing means includes means for decreasing fluid pressure acting radially against the vane means as the vane means approaches the rearward portion of the working chamber.
 14. The rotary fluid motor of claim 12 wherein the working chamber comprises a rotationally forward portion; and
 the fluid pressure means includes means for increasing the fluid pressure acting radially against the vane means as the vane means approaches the forward portion of the working chamber.

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