Jan. 9, 1962 A. E. RINEER 3,016,020

FLUID POWER CONVERTER

Filed Aug. 23, 1957 2 Sheets-Sheet 1

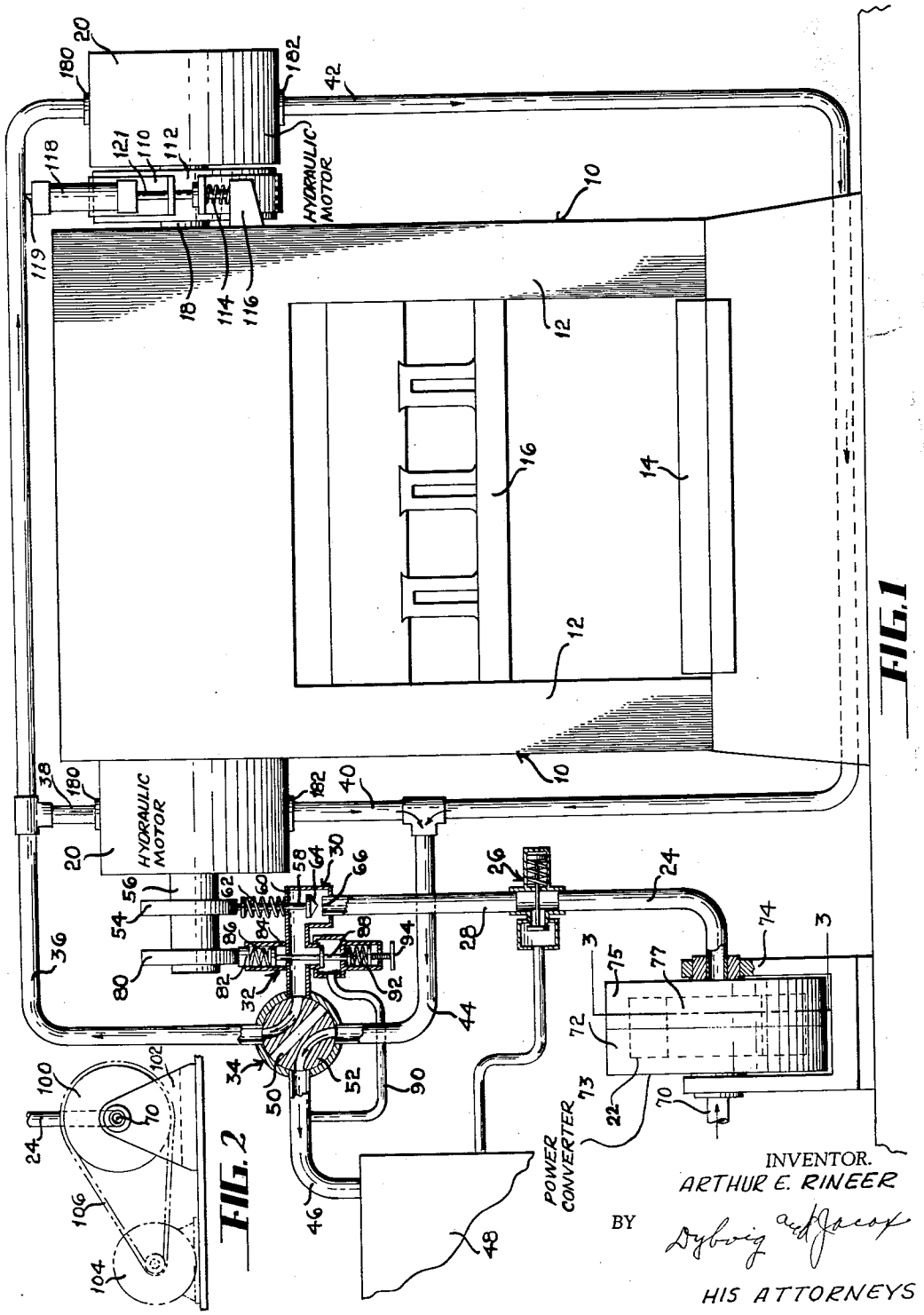


FIG. 1

FIG. 2

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FLUID POWER CONVERTER

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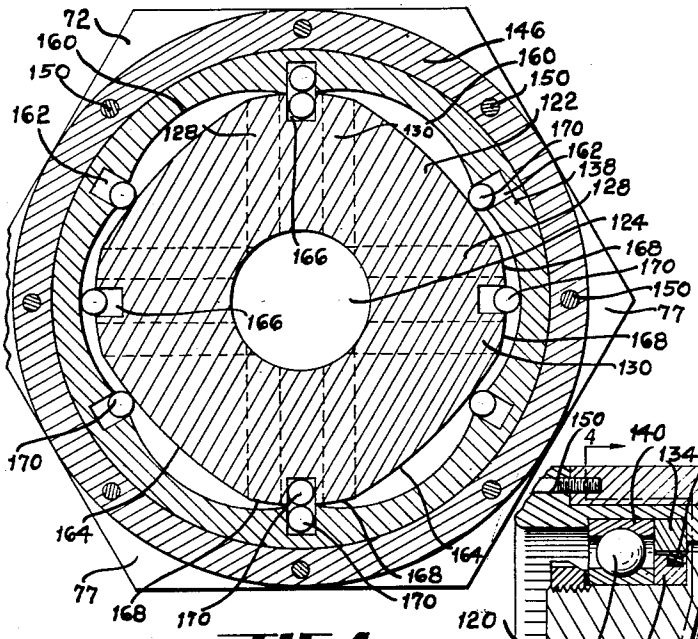


FIG. 4

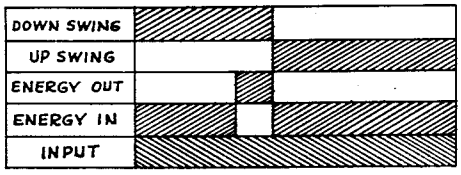


FIG. 6

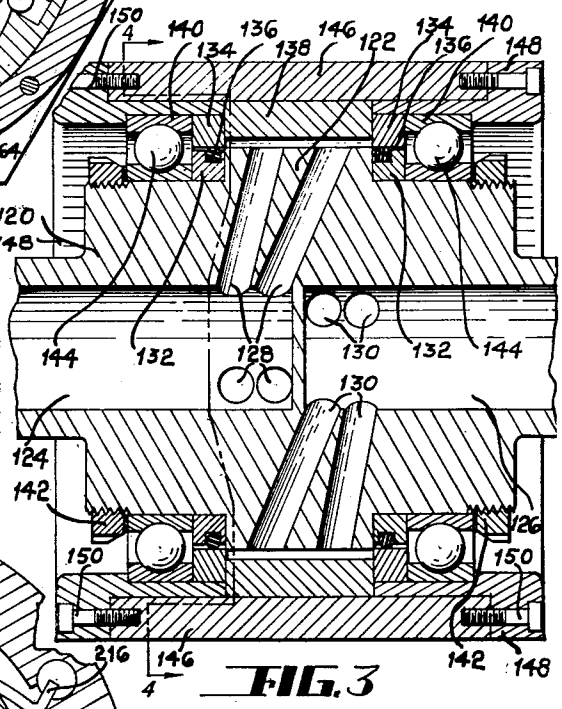


FIG. 3

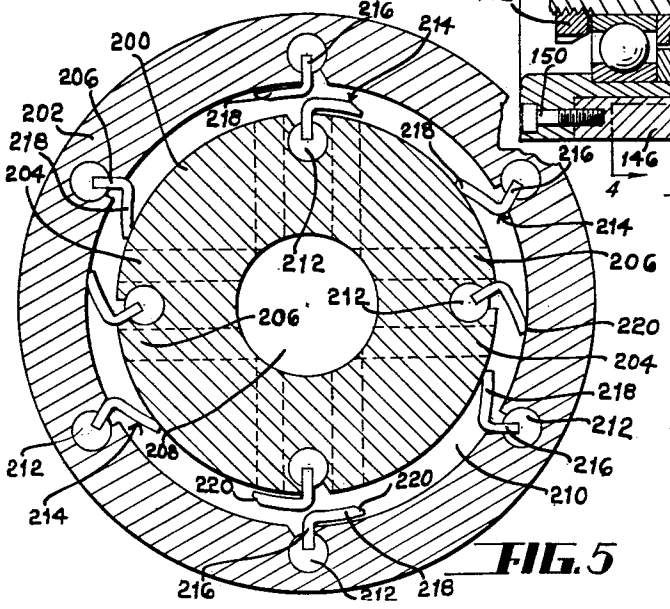


FIG. 5

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3,016,020

FLUID POWER CONVERTER

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Filed Aug. 23, 1957, Ser. No. 679,958
14 Claims. (Cl. 103—118)

This invention relates to a fluid power converter and more particularly to the combination of a novel hydraulic power source with associated mechanisms for controlling the operation of a punch press or the like. However, this invention is not necessarily so limited.

An object of the present invention is to provide a hydraulic drive for punch presses and the like including means for storing mechanical energy in a form readily convertible to hydraulic power as required by the driven apparatus.

Another object of this invention is to provide, in association with a fluid power converter for converting mechanical power to fluid power, a mechanical energy storage device for supplying bursts of mechanical energy to the fluid power converter, the assembly operating inherently to satisfy the demands of a driven apparatus.

Another object of this invention is to provide, in combination, an inertia device and a fluid pump for driving punch presses and the like, the construction and arrangement being such that the power supplied to the combination may be, selectively, mechanical, or hydraulic.

Still another object of this invention is to provide an improve fluid power converter.

A further object of this invention is to provide a hydraulic drive for a punch press or the like, including control means for regulating the speed of operation of the driven apparatus, the control means operating in synchronism with the present position of the driven apparatus.

Still a further object of this invention is to provide a hydraulic drive for punch presses and the like including control means for regulating the hydraulic fluid pressure delivered to the driven apparatus, the control means operating in synchronism with the present position of the driven apparatus.

Still another object of this invention is to provide a hydraulic drive for punch presses and the like, wherein the direction of drive is reversible.

Still another object of this invention is to provide, in a hydraulic drive assembly for a punch press or the like, a braking mechanism operating in synchronism with the hydraulic drive.

Other objects and advantages reside in the construction of parts, the combination thereof, the method of manufacture and the mode of operation, as will become more apparent from the following description.

In the drawings, FIGURE 1 is a front elevational view of the hydraulic drive and operating controls of this invention in association with a conventional punch press, portions of the assembly being broken away, and other portions being shown in section.

FIGURE 2 is an elevational view of a modification of the hydraulic drive illustrated in FIGURE 1.

FIGURE 3 is a sectional view taken substantially along the line 3—3 of FIGURE 1.

FIGURE 4 is a sectional view taken substantially along the line 4—4 of FIGURE 3.

FIGURE 5 is a sectional view analogous to that of FIGURE 4 illustrating a modification.

FIGURE 6 is a chart illustrating energy relationships encountered in the operation of the hydraulic press drive.

Referring to the drawings in detail, FIGURE 1 illustrates a conventional punch press **10** provided with the hydraulic drive of this invention. The punch press **10** comprises a frame **12** supporting a bed **14** and a ram

16 adapted to move reciprocally above the bed **14**. The ram **16** is driven reciprocally through a crankshaft **18** rotated by hydraulic motors **20**, there being one hydraulic motor situated on either side of the frame of the press **12**. It is to be understood that the two hydraulic motors **20** can be replaced by one hydraulic motor having twice the capacity of the motors **20**.

These hydraulic motors **20** will be given further description in the following. For the present, it suffices to note that these motors are reversible positive displacement rotary fluid power converters.

Hydraulic fluid under pressure is supplied to the hydraulic motors **20** through a fluid power converter **22**, through a first conduit **24**, a pressure relief valve **26**, a second conduit **28**, a flow control valve **30**, a second pressure relief valve **32**, and a reversing valve **34**. The reversing valve **34** in the position illustrated connects the high pressure side of the hydraulic fluid inlet conduits with conduits **36** and **38** for conveying hydraulic fluid to the motors **20**. Simultaneously, the valve **34** connects conduits **40** and **42** leading from the motors **20** to conduits **44** and **46** leading to a fluid reservoir **48**.

By rotating the valve **34** a quarter turn in either direction of rotation, the inlet and outlet connections for the motors **20** are reversed, such that the direction of rotation of the motors **20** is reversed. The valve **34** is provided with an open center conduit **50**, at an intermediate position between the extreme reverse positions of the valve, for connecting the high pressure hydraulic conduits **24** and **28** directly to the reservoir **48** through the conduit **46**, bypassing the drive motors **20**. This position of the valve **34** is employed for idling the press **10**.

The valve **34** illustrated is a manually operable valve having a rotatable inner manifold **52**. However, it is not desired to limit the scope of the present invention to the particular form of reversing valve illustrated, it being recognized that other types of valves, such as spool valves, may be employed for this purpose. It is further deemed to be within the scope of this invention to employ a semi-automatic reversing valve for reversing the direction of hydraulic fluid flow through the motors **20**, the valve being actuated by solenoids, for example.

The construction and operation of the fluid power converter **22** is illustrated in FIGURES 3 and 4. The power converter is assembled upon a substantially cylindrical shaft portion **120**. The portion **120** includes an enlarged contoured portion **122** designated a stator. Axially aligned manifolds **124** and **126** within the shaft portion serve as fluid inlet and outlet manifolds, these manifolds connecting to fluid ports **128** and **130**, respectively, which conduct fluid to the periphery of the stator **122**.

On either side of the stator **122** are placed a pair of concentric annular wear rings **132** and **134** of rectangular cross-section. The interface between the concentric rings coincides substantially with the maximum radius of the stator **122**. This radius is designated the pitch radius. An annular O-ring seal **136** recessed in the inner wear ring **132** obstructs fluid flow through the interface between the wear rings **132** and **134**.

A contoured annular element **138** designated a rotor encircles the stator **122** between the annular rings **134**. The rotor **138** is positioned axially relative to the stator by annular ball bearings **140** placed on the shaft portion **120** outside the spaced pairs of concentric wear rings **132** and **134**.

Nuts **142** threadedly engaging opposite ends of the shaft portion **120** compress the inner races of the bearings **140** against the inner wear rings **132**. This firmly secures the inner races of the bearings in fixed spaced relation to the stator **122**. As illustrated, the balls **144** of the bearings **140** are recessed in the inner and outer races such that an axial thrust upon one race is trans-

mitted through the balls to the opposite race. Accordingly, as the nuts **142** are tightened upon the inner races of the bearings **140**, the outer races are compressed against the wear rings **134** which in turn grip the rotor **138** and align same axially with the stator **122**. Lock washers **145** are used to lock the nuts **142** in position.

A cylindrical housing **146** is projected over the rotor **138** and keyed thereto by means not shown. The housing **146** is centered concentrically upon the shaft portion **120** by end rings **148** seated upon the outer races of the bearings **140** and bolted to the housing **146** by suitable bolts **150**.

This novel construction affords several distinct advantages. As will be described hereinafter, the rotor **138** may be rotated upon the stator **122** by delivery of a mechanical torque to the rotor, or by the introduction of fluid under pressure to one of the manifolds **124** and **126**. It is apparent from the construction that the rotor **138** is centered for rotation by the bearings **140**, these bearings absorbing all radial and axial loads to which the rotor may be subjected. Since these bearings may be constructed with great precision precise centering of the rotor about the stator is readily obtained.

Additionally, it will be noted that the only rotary fluid seals required are at the interfaces between the wear rings **132** and **134**. These seals are effected by the O-ring seals **136**. Essentially, then, fluid leakage in the power converter is controlled by only two ring seals.

Other advantages derived from this construction reside in the ease of assembly and disassembly, low tolerances resulting in interchangeability of parts, et cetera.

The oppsing faces of the rotor **138** and the stator **122** are contoured as illustrated in FIGURE 4. The inner surface of the rotor **138** is divided into six sectors **160** by six equispaced axially extending channels **162** opening substantially to the pitch radius as defined hereinbefore. The sectors **160** intermediate the channels **162** are cut on a radius of curvature less than the pitch radius so as to be recessed from the pitch radius.

The peripheral surface of the stator **122** is divided into four sectors **164** by four equispaced axially extending channels **166** opening substantially to the pitch radius. A land **168** occupying substantially the pitch radius extends circumferentially from either side of each channel **166**. As illustrated there are two fluid ports **128**, connecting with the manifold **124**, opening through each land **168**, on the counterclockwise side of each channel **166**. Similarly, a pair of ports **130**, connecting with the manifold **126**, open through lands **168** on the clockwise side of each channel **166** as viewed in FIGURE 4.

The sectors **164** between each pair of lands **168** are cut on a radius of curvature exceeding that of the pitch radius so as to be recessed from the pitch radius. The arrangement is such that the rotor **138** may rotate relative to the contoured surface of the stator without abrasion therebetween.

Seated loosely within each of the channels **162** and **166** is an elongate cylindrical valve **170** which is preferably made of light weight plastic material. The radius of these valves **170** is made slightly in excess of the radial departure from the pitch radius of the sectors **160** and **164**, respectively. Correspondingly, the width and depth dimensions of the channels **162** and **166** exceed slightly the diameter of the valves **170**. The length of these valves **170** equals substantially the distance between the wear rings **132** and **134** such that the ends of these valves will make a fluid seal against these annular rings. No appreciable leakage results at the ends of the valves **170** due to the gap between the wear rings **132** and **134** if this gap is made small.

The contouring dimensions of the rotor and stator surfaces are such that once the power unit is assembled with the valves in place in their respective channels, the valves are trapped in their respective channels and can move only in a radial direction. While the valves **170** have been described as all having the same diameter, it is apparent that depending upon the radius of curvature of the sectors of the rotor and stator, the valves **170** of the rotor need not necessarily have the same diameter as the valves of the stator. Correspondingly, the channels **162** and **166** need not necessarily have the same dimensions.

The construction herein set forth, in so far as it relates to the fluid power converter **22**, constitutes an improvement upon the fluid power converters disclosed in my copending applications, Serial Nos. 595,372 and 640,918, filed July 2, 1956 and February 18, 1957, respectively, and entitled Fluid Power Converter. As described in detail in these pending applications, the operation of the power converter is as follows. Upon the introduction of a fluid under pressure into the manifold **124**, fluid flows through the several ports **128** to the annular space between the rotor and stator. There the fluid flows circumferentially toward the several ports **130** which function as fluid outlet ports. The flow of fluid across the channels **162** and **166** draws the valves **170** therein outwardly toward the opposing contoured surface. As the valves seat upon the opposing surfaces, the annular cavity between the opposing surfaces is partitioned. This condition is illustrated statically in FIGURE 4, where it will be noted that the fluid inlet ports **128** are sealed off from the fluid outlet ports **130**.

As the valves move into position to separate the fluid inlet and outlet ports, the fluid pressure in the portions of the annular cavity communicating with the fluid inlet ports rises, tending to drive the rotor in the counterclockwise direction as viewed in FIGURE 4. That is, the rotor is driven such that the fluid can move in the counterclockwise direction across the stator surfaces from inlet to outlet. As the rotor rotates, the valves **170** are cammed one over the other due to the contouring of the rotor and stator surfaces.

Short circuiting between the fluid inlets and the fluid outlets is prevented by making the central angle between the fluid inlet ports and the fluid outlet ports, that is the central angle of the stator sectors **164**, equal to 60°. This is the central angle between adjacent channels **162** of the rotor.

Since there is no short circuit path of leakage between the fluid inlets and fluid outlets, there is a fixed relationship between the angular velocity of the rotor and the velocity of the incompressible fluid flowing through the power converter. In other words, a given volume of fluid flowing through the device will displace the rotor a fixed distance. The power converter is thus a positive displacement device.

It is apparent that the power converter of FIGURES 3 and 4 can be operated in either the clockwise or counterclockwise direction depending on which manifold **124** or **126** is exposed to the greater fluid pressure.

The fluid power converter is further operable either as a motor or a pump. Operation as a motor has been described. If the fluid power converter is initially primed with a fluid, then the rotor is driven mechanically, there will be an agitation of the fluid in the annular space between the rotor and stator. This fluid movement seats the valves **170**. Subsequently, fluid will be propelled through the power converter as a result of the delivery of mechanical power to the rotor. In practice, the power converter when continuously rotating in a given direction makes a smooth transition from operation as a motor to operation as a pump, and vice versa, depending on whether the rotor is delivering or receiving mechanical energy.

The present power converter, while designed with four stator valves and six rotor valves, will operate satisfactorily with other numbers of valves. For example, there may be five rotor valves and four stator valves as shown in my aforementioned copending applications, or there may be three rotor valves and two stator valves. In

general, the number of valves in the rotor must exceed the number of valves and associated fluid inlets and outlets in the stator.

The advantage in using the six-four valve arrangement, as illustrated herein, is that the radial forces to which the rotor element is subjected are balanced at all times. That is, diametrically opposed rotor surfaces are always subjected to equal fluid pressures. Accordingly, there is no tendency for the rotor to rotate eccentrically upon the stator. This construction, therefore, eliminates any tendency the rotor may have to wobble as it rotates.

The rotor and stator of this power converter are essentially coacting torque transmitting members which are set into relative rotation by the movement of fluid therebetween. In practice, either may be mounted stationary. In constructing the fluid motors **20** of the hydraulic press assembly, the structure of the rotor and stator, as illustrated in FIGURES 3 and 4, is reversed. That is, the outer contoured member of these motors, corresponding to the rotor **138** of FIGURE 4, is mounted stationary and is provided with four channels and valves, and with flanking inlet and outlet ports and lands. The inner member, corresponding to the stator **122** of FIGURE 4 is provided with six channels and valves. The inlet and outlet ports for the stator are manifolded in the housings **178** with inlet and outlet conduits **180** and **182** emerging from the housings. These motors **20** operate identically with the power converter **22** except that the inner members thereof rotate, rather than the outer members.

The operation of the press drive of FIGURE 1 will now be described. Fluid under pressure is supplied to the manifold **124** of the power converter **22** from a suitable fluid source through a conduit **70**. The fluid under pressure drives the power converter **22** as a motor, exhausting into the conduit **24**. The power converter **22** is mounted within a comparatively large flywheel **72** journalled for rotation in a suitable cradle **74**. The flywheel **72** comprises two cylindrical halves **73** and **75** each provided with a hexagonal cavity. A complementary hexagonal key **77** is fixedly secured about the housing **146** of the power converter, as illustrated in FIGURE 4. In assembling the flywheel, the halves **73** and **75** are bolted about the flywheel, the flywheel being independently supported for rotation in the cradle **74**, but keyed to the housing **146** of the power converter so as to rotate therewith.

If the valve **34** is in the position illustrated in FIGURE 1, the fluid exhaust from the power converter **22** will pass to the motors **20**, which rotate to reciprocate the ram **16**, then to the reservoir **48**. If the ram **16** is permitted to reciprocate in a work-free cycle, the flow rate of the hydraulic fluid from the power converter **22** to the drive motors **20**, then to the reservoir **48** remains substantially constant, except as may be modified by the control valve **30** to be discussed subsequently. If the system is permitted to come to equilibrium, the flywheel **72** will rotate at an angular velocity determined by the ratio of the fluid flow incident through the conduit **70** to the volume of fluid displaced in the power converter **22** in each revolution. Similarly, the average angular velocity of the motors **20** will be determined by the ratio of fluid flow incident through the conduit **70** to the volume of fluid displaced in each motor in each revolution.

In this equilibrium condition, the horse power consumed by the system in work-free operation will be proportional to the product of the average fluid flow incident in the conduit **70** and the fluid pressure drop from the conduit **70** to the conduit **46**. The factors creating this pressure drop are, among other things, fluid friction losses in the conduits, fluid friction losses across the motors **20** and the power converter **22**, mechanical friction associated with the flywheel **72** and the crankshaft **18**, and the work required in reciprocating the ram **16**.

It is to be noted that in this equilibrium state, the power converter **22** is operating neither as a motor nor a pump, but is coasting without any substantial drain on the hydraulic system other than fluid friction losses across the power converter and the consumption of power from the hydraulic system to maintain the speed of the flywheel **72** against mechanical friction. The rotating flywheel represents a considerable amount of stored energy in this equilibrium state.

When metal is introduced into the punch press such that the press, in forming the metal undergoes a work producing cycle, this equilibrium is disturbed. As the ram **16** travels downwardly to bring metal shaping dies into contact with the metal to be shaped, the ram **16** is braked by the metal, slowing the motors **20** and creating thereby a back pressure in the fluid conduits **24**, **28**, **36** and **38**. This back pressure tends to reduce the flow of fluid through the power converter **22**. However, the inertia in the flywheel **72** opposes any such reduction. The result is that flywheel **72**, now tending to rotate more rapidly than the reduced fluid flow through the power converter **22** permits commences to drive the power converter **22** as a pump in direct opposition to the back pressure in the conduits **24**, **28**, **36** and **38**. The operation of the combined flywheel **72** and power converter **22** is such that the combination strives, due to the inertia of the flywheel **72**, to rotate at constant speed, thus striving to deliver a constant volume of hydraulic fluid to the motors **20**. This operation continues until sufficient pressure has been developed in the motors **20** to drive the ram **16** through the metal forming cycle.

After the work producing cycle, on the upstroke of the ram **16**, the hydraulic power incident through the conduit **70** is partitioned, part of the power being consumed in recharging the flywheel **72** to its equilibrium speed, and the remainder being consumed in elevating the ram **16**.

The operating cycle is illustrated graphically in FIGURE 6 by means of a bar graph. The horizontal axis of this graph is a time axis and extends through one cycle of continuous operation of the press. This is illustrated by the first two lines of the graph, which show that the first part of the operating cycle comprises a downswing of the ram of the press and the second part of the cycle comprises an upswing of the ram of the press. The last line of the graph shows a constant fluid energy input throughout the entire press operating cycle.

The third and fourth lines of the graph refer to the energy consumption of the flywheel. These show that the flywheel receives energy throughout the first part of the downswing, then discharges energy to the press at the end of the downswing. Thereafter, the flywheel absorbs energy from the fluid input throughout the upswing of the ram.

In the light of the foregoing discussion, a substantially constant input of hydraulic power may be regarded as divided into two parts, one part for carrying the ram **16** through a work-free cycle, and the other part for intermittently charging the flywheel **72** with mechanical energy which is released to move the ram **16** through the work producing portion of its operation cycle.

The capacity of the power converter **22** in relation to the moment of inertia of the flywheel **72**, is adjusted so that it is possible to restore fully the mechanical energy in the flywheel **72** during each cycle of operation when the press is under maximum load during continuous operation. Proper balance is achieved when the total hydraulic power entering the fluid power converter **22** from the conduit **70** approximates the average power consumption of the press **10** in continuous operation.

The construction and operation of the speed control valve **30** will now be described in detail. Associated with the speed control valve is a cam **54** fixedly secured to a shaft **56** splined to or integral with the rotating inner element of one of the hydraulic motors **20**. The cam **54** rotates in synchronism with the crankshaft **18** of the punch press, and, hence, in synchronism with the recipro-

cal movement of the ram 16. A valve element 58 reciprocally mounted in the housing 60 of the valve 30 is biased into abutment with the cam 54 by a compression spring 62. In high fluid pressure assemblies, the spring 62 may be omitted in the present arrangement, the biasing force being supplied solely by the fluid pressure. The valve element 58 has a tapered head 64 coacting with a valve seat 66 to meter the fluid flow through the conduit 28. With this arrangement of the valve 30, the flow of fluid through the conduit 28 to the motors 20, and hence the speed of the positive displacement motors 20, may be regulated in a predetermined manner by the cam 54, the fluid flow being synchronized with the movement of the ram 16 driven by the crankshaft 18. As an example, the shape of the cam 54 may be such that the ram 16 initially is moved rapidly on the downstroke, then decelerated during the work producing portion of the downstroke, then accelerated on the upstroke preparatory to the next operating cycle. The shape of the cam 54 and consequently the variation in the speed of movement of the ram 16 will, of course, depend upon the particular operation being performed by the press 10. It will be apparent to those skilled in the art that the speed control valve 30 will operate effectively anywhere in the hydraulic circuit provided the valve is in series relation with the motors 20.

In order to prevent press failure, the pressure relief valve 32 is operated in synchronism with the ram 16 by the following mechanism. Fixedly secured to the shaft 56 supporting the cam 54 is a second cam 80. Engaging the periphery of the cam 80 is a plunger 82 biased outwardly of a sleeve 84 mounted on the valve housing 60 by a compression spring 86. The compression spring 86 operates against a valve element 88 reciprocally mounted in the housing 60 and adapted to obstruct the flow of fluid out of the housing 60 to a conduit 90 connecting through the conduit 46 to the fluid reservoir 48. The valve element 88 is biased to fully obstruct the flow of fluid to the conduit 90 by a compression spring 92, the tension of which may be regulated by a control wheel 94.

The force with which the valve element 88 opposes a flow of fluid through the conduit 90 is determined by the difference in tension between the springs 92 and 86. This difference is in turn dependent on the angular position of the cam 80, and, as a consequence, is dependent upon the present position of the ram 16 in the press operating cycle. Accordingly, the valve 32 functions as a pressure relief valve, wherein the pressure at which the valve opens depends upon the present position of the ram 16.

Preferably, the shape of the cam 80 complements the mechanical advantage associated with the crankshaft 18 in moving the ram 16. Ordinarily, the mechanical advantage associated with the crankshaft 18 is at maximum when the ram 16 is near the top and bottom of its reciprocal movement and at a minimum when the ram is in the middle of its downstroke and upstroke. It is desirable that the torque deliverable to the crankshaft 18 vary, such that the product of the maximum torque deliverable to the crankshaft and the mechanical advantage between the crankshaft and the ram is a constant. Accordingly, the maximum force deliverable by the ram at any position will be a constant.

Thus, the torque deliverable by the motors 20, which is proportional to the fluid pressure deliverable to the motors 20, is regulated by the cam 80 so as to complement the mechanical advantage associated with the crankshaft 18. The valve 32 is adjusted to yield when the fluid pressure coupled with the mechanical advantage in the crankshaft 18, at any position of the crankshaft 18, will produce a ram pressure sufficient to fracture the press framework. It will be apparent to those skilled in the art that the valve 32 will operate effectively only if in parallel relation with the motors 20.

Due to the presence of the cam operated control valves 30 and 32 in the hydraulic circuit, it is apparent that the operation of the press 10 will not be equal in both directions of operation, as determined by the reversing valve 34, unless the cams 54 and 80 are symmetrical for both the upstroke and downstroke of the ram 16. This will not ordinarily be the case with the cam 54 and therefore separate cams must be employed for each direction of operation. In conventional press operation, however, the reversing feature is utilized only during set-up of the metal forming dies and the need for special reversing cams does not exist.

An exception arises in light press operations where it is sometimes the practice to operate the press in one direction, carrying the ram 16 on the downstroke through a metal forming operation, then to reverse the press on the upstroke to return the ram from the opposite direction through another metal forming operation, then, to reverse the press again on the upstroke for another metal forming operation, and so on. This type of operation is sometimes termed "rocking the press," the objective being to increase the press output by driving the crankshaft of the press reciprocally through a short stroke, rather than rotating the crankshaft a full 360° for each metal forming stroke. Where such operation is desired, a special symmetric control cam may be designed to replace the cam 54.

When it is desired to stop the press 10 in an idling position, the valve 34 is adjusted to the intermediate position venting the hydraulic power supplied through the conduit 70 to the reservoir 48, bypassing the driven motors 20. With the valve 34 in the idling position, the fluid pressure on either side of the motors 20 neutralizes to an equilibrium value, not necessarily atmospheric pressure. In the idling position, the crankshaft 18 is braked by a brake shoe 110 engaging a brake drum 112 fixedly secured to the crankshaft 18. The brake shoe 110 is pressed against the drum 112 with a pressure supplied by a spring 114 coacting with one end of the brake shoe 110 and a fixed support 116. When the press is next placed in operation by actuating the valve 34, the fluid pressure in the conduit 36 rises, actuating a hydraulic cylinder 118 through a conduit 119. The hydraulic cylinder 118 is provided with a plunger 121 for actuating the brake shoe 110 in opposition to the spring 114 to release the brake.

In the press drive system illustrated in FIGURE 1, the brake release operates effectively only when the conduit 36 is supplied with fluid under pressure. When the direction of operation of the press is reversed by actuating the control valve 34, the conduit 36 is not exposed to a fluid under pressure and the brake tightens, requiring the drive motors 20 to operate against the brake. Since reversal of the direction of the operation of the press is ordinarily desirable only for setting-up purposes, operation of the driving motors against the brake in the reverse direction is not objectionable. It is within the purview of this invention, however, to connect the conduit 119 directly between the cylinder 118 and the conduit 28, rather than the conduit 36, such that the brake for the crankshaft 18 is released whenever the motors 20 receive hydraulic power regardless of the direction in which the crankshaft 18 is driven.

The pressure relief valve 26 positioned between the conduits 24 and 28 functions as a safety relief valve in the event of failure or improper operation of the control valves 30 and 32. It is apparent that, should either of these control valves become jammed, or otherwise defective inoperation, an excessive pressure could develop in the conduits 24 and 28. With the pre-set spring loaded safety valve 26 in the conduit, such a pressure build-up would be prevented.

FIGURE 2 illustrates a modification of the power supply for the hydraulic press. In this modification is flywheel 100 is mounted for rotation within a suitable cradle

102 and adapted to be driven by an electric motor 104 through a suitable V-belt drive 106. The fluid power converter 22 is preferably mounted within the flywheel 100 in the manner illustrated in connection with the preferred embodiment. However, the assembly will work equally well if the power converter 22 is mounted outside the flywheel 100 and mechanically connected to the flywheel.

In the operation of this modification, the conduit 70 on the inlet side of the power converter 22 is connected to the fluid reservoir 48, and the outlet conduit 24 is connected through suitable valves to the motors 20, as in the preferred embodiment. Mechanical power is delivered directly to the flywheel 100 through the belt 106, thereby driving the power converter 22 as a pump at a speed sufficient to deliver the proper hydraulic pressure for work-free operation of the ram 16. When the ram 16 enters the metal forming portion of its cycle, the mechanical energy stored in the flywheel 100 provides a sufficient burst of power to carry the ram 16 through the metal forming portion of the cycle. Thus, in the modification of FIGURE 2, a substantially constant amount of mechanical power supplied to the flywheel 100 by the motor 104 may be employed to supply a variable hydraulic power for operation of the press 10.

It is worthy of note here that the present invention has distinct advantages over conventional presses, which are ordinarily operated with a flywheel mechanically connected to the crankshaft of the press through a clutch apparatus. In these conventional presses, the stored energy of the flywheel is released directly to the crankshaft whenever the ram of the press becomes jammed. The resultant stresses established in the frame and the working parts of the press are sometimes destructive to the whole press, or at least a part of it. Elaborate means are required to prevent failure of the press in the event the ram becomes jammed due to faulty operation of the metal forming dies.

In the present invention, however, the flywheel which supplies the necessary energy for carrying the ram through the metal shaping operation is mechanically separate from the crankshaft of the press and connected therewith only by the hydraulic system. With this arrangement, it is a relatively simple matter, as demonstrated herein, to provide the necessary pressure relief valves for limiting the mechanical torque deliverable to the crankshaft of the press such that stresses exceeding the structural capacity of the press cannot be created.

Another benefit derived from the use of the mechanically remote flywheel is the ease with which the press may be rocked for rapid production operations, due to the fact that the direction of movement of the crankshaft may be reversed without reversing the direction of movement of the flywheel 72. In such operation, braking of the press on reverse movement can be eliminated by connecting the brake release cylinder 118 directly to the conduit 28, as discussed hereinbefore.

Further advantages reside in the ease with which the speed of the ram and the pressure deliverable by the ram may be regulated through the control valves 30 and 32.

FIGURE 5 illustrates a modified fluid motor comprising an inner stator member 200 and an outer rotor member 202 assembled as illustrated in FIGURE 3 in connection with the preferred embodiment. The stator is provided with fluid inlet ports 204 and outlet ports 206 manifolded as in the preferred embodiment. The inlet manifold 208 is illustrated in FIGURE 5.

In this modification, the opposing rotor and stator surfaces are each substantially cylindrical and cooperate to form an annular cavity 210 therebetween. The body of the rotor is bored axially at six equispaced points to receive axially extending shafts 212. An axially extending V-shaped slot in the periphery of the rotor adjacent each boring therein exposes an axial segment of each shaft 212. The shafts 212 are each provided with axial slots which receive a short flange 216 of a right angular axially extending vane member 214. Arcuate long flanges 218 of these vane members 214 are thus supported for pivtoal motion in the annular cavity 210 by the shafts 212. Preferably, the flanges 218 have a radius of curvature corresponding to that of the adjacent rotor surface.

The stator 200 is similarly bored and slotted to receive four equispaced vane members 214 and shafts 212. The short flanges 216 of the rotor and stator vane members are of such a length that the long arcuate flanges 218, when oriented parallel to the rotor and stator surfaces will pass one over the other. The long flanges 218 are of such a length that, when pivoted into the annular cavity, they will ride upon the opposing surface. The ends of these long flanges 218 are beveled at 220 to establish an area of contact, as opposed to a line contact with the opposing surface. As illustrated, the vanes of the rotor and stator are oriented oppositely. The porting arrangement of the stator is such that the long flanges 218 in the vanes of the stator substantially overlie the fluid inlet ports 204 when oriented parallel to the surface of the stator.

The motor operates as follows. Fluid under pressure is introduced in the manifold 208, this fluid flowing radially outwardly to the periphery of the stator. There, the vane members 214 of the stator are driven outwardly, the beveled ends 220 engaging the surface of the rotor. A clockwise flow of fluid in the annular cavity 210 follows. This flow of fluid creates a low pressure zone on the outer surfaces of the flanges 218 of the vane members 214 of the rotor, causing these vane members to pivot into the annular cavity 210. As a result, these vane members partition the annular cavity obstructing the clockwise fluid flow.

This imparts a torque to the rotor 202, such that the rotor tends to rotate in the clockwise direction, as viewed in FIGURE 5. As the rotor rotates, the fluid flows from fluid inlets 204 to fluid outlets 206. The vane members of the rotor cam over the vane members of the stator without materially inhibiting the rotary movement of the rotor. As was the case with the preferred embodiment of FIGURE 4, the motor works with greatest efficiency when the central angle between the vane members 214 of the stator equals substantially the central angle between consecutive fluid inlet and outlet ports in each quadrant of the stator. This central angle is 60°.

This particular motor is designed for rotation only in the clockwise direction as view in FIGURE 5. It is apparent, however, that by reversing the orientation of the vane members of the rotor and stator, the motor can be designed to operate in the opposite direction. It is also apparent that, as was the case with the preferred embodiment, that the motor can be designed with a rotating inner, rather than outer member. This is accomplised by placing four vane members and associated fluid inlets and outlets in the outer member, and six vane members in the inner member. In contradistinction to the preferred embodiment, this motor is operable only as a fluid motor, and not as a fluid pump.

This motor may be employed as a substitute for the motors 20 of the preferred embodiment omitting the press reversing feature. The advantage of such substitution lies in the increased output torque obtainable in the motor for a given input fluid pressure. In the preferred embodiment of FIGURES 3 and 4, the output torque is limited through the use of circular, radially movable valves. In the present modification, wherein pivotally mounted vane members are employed, a considerably greater vane area may be presented to the fluid pressure.

The present modification may also be constructed more economically for the reason that the tooling of the rotor and stator surfaces is markedly simplified.

Although the preferred embodiment and various modi-

fications of the device have been described, it will be understood that within the purview of this invention various changes may be made in the form, details, proportion and arrangement of parts, the combination thereof and mode of operation, which generally stated consist in a device capable of carrying out the objects set forth, as disclosed and defined in the appended claims.

Having thus described my invention, I claim:

1. In a fluid power converter comprising concentric torque transmitting members mounted one within the other for relative rotation about a fixed axis, said members having opposed peripheries encircling said axis and defining an annular space therebetween, fluid inlet and fluid outlet means permitting a fluid flow circumferentially in said annular cavity, and valve means coacting with said fluid to impart a rotational torque to one of said torque transmitting members, the improvement wherein the inner of said torque transmitting members is formed by an enlargement of an inner cylindrical shaft and the outer of said torque transmitting members is a tubular member fixedly mounted within a tubular housing, said power converter including an annular wear ring of rectangular cross-section seated against each end of each torque transmitting member, the construction and arrangement being such that the wear rings on each end of said power converter are concentric and aligned axially, means providing a fluid seal between each pair of axially aligned wear rings, said wear rings and fluid seals cooperating to seal the ends of said annular cavity, a pair of ball bearing rings having inner and outer races mounted upon said inner shaft, there being one bearing ring disposed on each end of said inner shaft adjacent said wear rings, said bearing rings supporting said tubular housing on said shaft for rotation one relative to the other, and means for wedging the inner and outer races of said bearing rings aganist the adjacent wear rings, whereby said torque transmitting members and said wear rings are sandwiched and aligned axially by said bearing rings.

2. The improvement according to claim 1, wherein said wedging means includes a nut threadedly engaging each end of said inner shaft.

3. The improvement according to claim 1, wherein one of said torque transmitting members is provided with a plurality of equi-spaced axially extending first channels opening to said annular cavity and dividing the surface thereof contiguous with said annular cavity into a plurality of sectors, the sectors intermediate adjacent channels being recessed so as to enlarge said annular cavity, there being one fluid inlet means adjacent one side of each said first channel and one fluid outlet means adjacent the other side of each said first channel, the other of said torque transmitting members being provided with a plurality of axially extending equi-spaced second channels opening to said annular cavity and dividing the surface thereof contiguous with said annular cavity into a plurality of sectors, the number of second channels exceeding the number of first channels, the sectors of said other torque transmitting member being recessed so as to enlarge said annular cavity, said valve means comprising a plurality of elongate cylindrical valve members, there being one valve member seated in each of said first and second channels and extending between the wear rings on either end of said power converter, the diameter of said valve members being substantially equal to the radial depth of said channels, said valve members moving radially into said annular cavity in response to a fluid flow in said annular cavtiy, the construction and arrangement being such that the radius of said valve members exceeds the maximum radial movement thereof into said annular cavity whereby said valve members are permanently seated in their respective channels.

4. The improvement according to claim 1 wherein said valve means comprises a plurality of vane members extending axially between said wear rings and means pivotally securing said vane members to said torque transmitting members in equi-spaced relation, said vane members each being pivotable between a position substantially parallel to the surface of the supporting torque transmitting member to a position in contact with the surface of the opposing torque transmitting member partitioning the annular cavity therebetween, one of said torque transmitting members having more vane members distributed in the surface thereof than the other, there being one fluid inlet means adjacent one side and one fluid outlet means adjacent the other side of each vane member in said other torque transmitting member, the vane members of said other member each being pivotal to positions overlying said fluid inlet means, the vane members in the opposite torque transmitting member being oppositely oriented, the construction and arrangement being such that, upon a fluid under pressure being introduced in said fluid inlets, said vane members will be drawn by the fluid flow into said annular cavity to partition same whereby a rotational torque will be established tending to rotate said torque transmitting members one relative to the other, said vane members camming one over the other in said annular cavity as said torque transmitting members rotate one relative to the other.

5. A fluid power converter comprising inner and outer cylindrical torque transmitting members mounted one within the other for relative rotation about a fixed axis, said members having opposed peripheries encircling said axis and defining an annular space therebetween, a plurality of elongate vanes, means supporting a portion of said vanes for pivotal motion in equi-spaced relation at the periphery of one of said members, means supporting the remainder of said vanes for pivotal motion in the opposite sense in equi-spaced relation at the periphery of the other of said members, there being a greater number of vanes carried by one member than by the other member, means providing a fluid inlet adjacent one side and a fluid outlet adjacent the other side of each vane in said other member, the vanes of said other member each being pivotable between a position substantially parallel with the periphery thereof overlying the fluid inlets therein and a position inclined from said periphery for partitioning said annular cavity, the vanes in said one member each being likewise pivotable between a position parallel to the periphery thereof to a position inclined from said periphery for partitioning said annular cavity, the construction and arrangement being such that upon the introduction of a fluid under pressure into said fluid inlets, said vanes will be drawn into said annular cavity to partition same whereby said torque transmitting members will receive a rotational torque, the vanes upon rotation of said members one relative to the other camming one over the other so as not to obstruct the rotation.

6. The power converter according to claim 5, wherein said vanes are provided with arcuate pivotal flanges having a curvature equal substantially to that of the surface of the supporting torque transmitting member and are beveled on one margin to provide an area of contact with the opposing surface.

7. The power converter according to claim 5, wherein said vanes each comprise substantially right angularly disposed flanges and where in the means pivotally supporting each said vane comprises an axially extending shaft rotatably journalled within the body of the supporting torque transmitting member, said member having a slot in the surface thereof, exposing said shaft to said annular cavity, one flange of said vane fixedly engaging said shaft so as to be pivotal therewith, the other flange of said vane being thereby pivotally supported within said annular cavity.

8. A fluid power converter comprising, in combination, a rotor member and a stator member adapted to rotate one with respect to the other, one of said members being an inner member substantially cylindrical in shape and having a contoured surface at the periphery thereof, the other of said members being an outer member encircling

said inner member, said outer member having a contoured surface at the inner periphery thereof opposing the contoured surface of said inner member and having a length equal substantially to that of said inner member, the maximum radius of said inner member being substantially equal to the minimum radius of said outer member, this radius being designated the pitch radius, sealing means abutting the ends of said inner and outer members enclosing the interspace between said opposing contoured surfaces, said inner and outer members each having a plurality of axially extending equi-spaced channels in the contoured surface thereof opening to the interspace therebetween substantially at the pitch radius thereof, there being six channels in said rotor member and four channels in said stator member, a plurality of elongate cylindrical valve members, there being one valve member extending axially and loosely seated in each said channel, the ends of said valve members lying in close proximity to said sealing means, said channels dividing the contoured surfaces of said inner and outer members into sectors, the sectors of said inner member having a radius of curvature greater than the pitch radius and the sectors of said outer member having a radius of curvature less than the pitch radius whereby said sectors cooperate to provide a plurality of chambers between said inner and outer members, the maximum departure of the surface of each of said sectors from the pitch radius being less than the radius of each valve member seated in the channels of the opposing surface, and means providing a plurality of fluid inlet openings and fluid outlet openings in the contoured surface of said stator member, there being one fluid inlet opening adjacent one side of each channel in said stator member and one fluid outlet opening adjacent the other side of each channel of said stator member such that there is one fluid inlet opening and one fluid outlet opening in each sector of said stator member, the portions of the contoured surface of said stator member bearing said fluid inlet openings and said fluid outlet openings having a radius of curvature equal substantially to the pitch radius.

9. In a fluid drive including, in combination, a rotary fluid power converter and a flywheel mechanically connected to the power converter, the improvement wherein said power converter comprises inner and outer members disposed concentrically for relative rotation about a common axis, said members having opposing peripheries encircling said axis of relative rotation and related so as to provide contiguous chambers therebetween, said chambers each having a plurality of equally spaced axially extending channels in the opposing periphery thereof, there being more channels in the periphery of one of said members than in the other member, fluid inlet means opening to said chambers adjacent one side and fluid outlet means opening to said chambers adjacent the other side of each channel in said other member, and a valve disposed in each channel of both of said members, the valves in each member seating against the opposing periphery of the other member to partition and seal said chambers in response to fluid pressure in said chambers, said valves being adapted to move into said channels away from the opposing periphery to enable the valves of one member to cross over the valves of the other member as one member rotates relative to the other, said flywheel being connected to one of said inner and outer members and the other of said members being supported stationary relative to said flywheel, the operation being such that delivery of fluid under pressure to said fluid inlets induces said one member to rotate relative to the other member thereby rotating said flywheel, said flywheel, upon being rotated so as to store mechanical energy, being capable of returning this energy to the power converter by rotating said one member to drive the power converter as a pump.

10. In a fluid drive according to claim 9, the improvement wherein the outer member of said power converter is connected to said flywheel and said flywheel encases said power converter.

11. A fluid power converter comprising a hollow outer member and an inner member mounted within said outer member for relative rotation about a fixed axis, said members having opposed peripheries encircling said axis and defining an annular space therebetween, each said periphery including a plurality of spaced axially extending channels and intermediate surface portions, the channels of said inner and outer members opening to said annular space at positions substantially equidistant from said axis, such distance being designated the pitch radius for said power converter, the intermediate surface portions of said inner and outer members being recessed away from said pitch radius so as to define chambers in said annular space, a plurality of axially extending valves, there being one valve disposed in each said channel, said valves being freely movable radially in said channels between positions recessed therein and positions engaging the opposing surface portions of said members, one of said members having six channels and valves therein and the other of said members having four channels and valves therein, said other member having a plurality of fluid inlets and outlets therein, there being one fluid outlet positioned to one side and one fluid outlet positioned to the other side of each channel thereof, the fluid inlet and outlet located between each pair of adjacent channels of said other member defining a central angle no less than the central angle between adjacent channels in said one member.

12. A fluid power converter comprising a hollow outer member and an inner member mounted within said outer member for relative rotation about a fixed axis, said members having opposing substantially cylindrical peripheries concentric to said axis and spaced apart to provide an annular chamber therebetween, each of said members having a plurality of movable valve elements extending axially at substantially equispaced intervals along the periphery thereof, one of said members having more valve elements in the periphery thereof than the other of said members, said other member having a plurality of fluid inlets and outlets therein, there being one fluid inlet opening to said annular chamber adjacent one side of each valve element therein and one fluid outlet opening to said annular chamber adjacent the other side of each valve element therein, the central angle defined by the inlet and outlet between each pair of adjacent valve elements being no greater than the central angle between adjacent valve elements in said one member, said valve elements moving outwardly into said annular chamber to engage the opposing periphery and partition said annular chamber in response to a fluid pressure differential between said inlets and outlets, each valve element also being movable oppositely away from the opposing periphery upon contact with a valve element in the opposing periphery so as to enable the valve elements of one member to pass over the valve elements of the other member, the construction and arrangement being such that the introduction of fluid pressure into said annular chamber causes said valve elements to partition said annular chamber and react with the fluid therein.

13. A fluid power converter according to claim 12 wherein said valve elements each comprise an elongate vane, and means supporting a portion of said vane for pivotal motion in the periphery of the member with which said valve element is associated.

14. The power converter according to claim 13 wherein said vanes each comprise substantially right angularly disposed flanges and wherein the means pivotally supporting each said vane comprises an axially extending shaft rotatably journalled within the body of the member with which the vane is associated, one flange of said vane fixedly engaging said shaft so as to be pivotal therewith.

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