ABSTRACT: A fluid pressure energy-translating device such as a fluid pump or motor, having a pressure-loaded element, such as a bushing, port plate or cheek plate, held in position in the device by fluid under pressure, which pressure is relieved by a rate of pressure gain control mechanism whenever pressure surges or peaks occur in the pressure loading.
FLUID PUMP HAVING INTERNAL RATE OF PRESSURE GAIN LIMITING DEVICE

The primary objective of this invention has been to provide an improved structure in fluid pressure energy-translating devices, particularly in fluid pumps operable when the device is subjected to operating conditions that would normally cause sudden increases in pressure or shock loads to instantaneously prevent such shock or peak pressure leading of a port or check plate so that the plate may momentarily move away from sealing engagement with the pumping chambers, and momentarily reduce the pump delivery into the system, and thereby prevent damage to the pump and system components.

The invention described and claimed herein is applicable to all fluid pumps or motors which include a bushing, port plate or check plate, the position of which is determined by fluid pressures. For purposes of describing one preferred embodiment of the invention, it is illustrated and described herein as applied to a vane-type hydraulic pump, but it is to be understood that its application is not limited to such devices.

Fluid pumps and energy-translating devices have long been known of the type that include a stator which encompasses or surrounds a rotor and in which there is a movable element, such as an end bushing, port plate or check plate which forms a wall at one side of the stator and rotor and is urged toward them by a spring and fluid pressure means. When these known devices are pumping fluid, the movable element, such as a check plate, is urged toward the rotor and stator by fluid pressure so that the check plate is clamped very tightly to the stator and forms a seal therewith to positively prevent any flow of fluid between the check plate and stator. In this type of pump, the occurrence of a sudden pressure peak or surge tends to cause the check plate to move away from the rotor and the vanes until a partial short circuit develops between the check plate and the rotor, thereby tending to relieve the pressure overload condition. Upon the development of this partial short circuit, some fluid leaks from the pressure port or exhaust of the pump to the low-pressure intake, the high shock pressure being somewhat relieved, since reduced pump volume is delivered into the circuit.

However, the ability for this action to occur in former pump designs is limited and does not, in all cases, prevent undesirable pressure peaks from occurring that can not only damage the pump, but other system components as well. Former designs are limited because of the limited compressibility of the small volume of fluid in the chamber that provides clamping pressure to the check plate.

It has therefore been an objective of this invention to augment the ability of the pressure loaded movable element or check plate to move to increase the temporary short circuit during shock conditions to eliminate undesirable peak pressures in the system.

These objectives are accomplished and this invention is predicated upon the concept of placing a rate of pressure gain valve that employs a hydraulic spring and poppet principle in a pump in a position to relieve pressure peaks from the clamping side of a pressure-loaded check plate, bushing, port plate, and/or pressure plate. This valve is operative in the event of a pressure peak or tendency for a sudden increase in pressure that exceeds a predetermined rate to allow the check plate to move away from the rotor and the vane and thereby create a substantial, temporary short circuit on the pressure side of the plate. This short circuit allows the fluid to pass from the pressure zone of the pumping plate to the suction side that thereby limiting the pressure causing the peak. By simultaneously and temporarily limiting the pressure on the clamping side of the check plate by venting it through a rate of pressure gain control valve, the ability of the pump to develop high pressure is also temporarily limited. Thus, the ability of the pump and pressure generator is limited until the reason for the occurrence of pressure peaks is eliminated. In other words, this rate of pressure gain valve allows an acceptable rate of pressure buildup on the clamping side of the check plate during normal operation, but prevents excessively high rates of pressure buildup which often cause peak system pressures substantially higher than relief valve settings, when slow response relief valves are used.

Another object of this invention is to provide means within a pump to momentarily bypass some of the pump displacement volume from its outlet pressure zones to its inlet port when, for any reason, the outlet port senses a sudden increase in pressure of a magnitude that would result in pressure peaks over and above a desired safe maximum system pressure, wherein the rate of pressure gain to which the mechanism responds can be changed.

A further object is to provide means for controlling the duration of the time period during which the bypassing of pump displacement volume is sustained.

The construction of the valve which relieves these conditions is such that it requires no minimum pressure to actuate it and it is therefore equally responsive to a dangerous pressure rising condition at 10 pounds per square inch as it is to a pressure gain which is initiated at 1,000 pounds per square inch. Consequently, a single valve is applicable to all types and sizes of fluid pumps and motors.

These and other objects and advantages of this invention will be more readily apparent from the following description of the drawings in which:

FIG. 1 is an axial section of a vane-type pump incorporating a preferred embodiment of the invention, and
FIG. 2 is a partial transverse or radial section taken along line 2-2 of FIG. 1.

The invention of this application is illustrated in the drawings as applied to a vane pump. It should be appreciated, however, that the invention is equally applicable to any type of pump or motor, such as a gear or piston type of pump or motor to control or limit the rate of pressure rise of fluid in the system which also acts upon various pressure-loaded elements such as port plates, check plates, bushings and/or pressure-responsive holdown mechanisms.

The pump to which the invention is applied by way of illustration includes a housing or casing formed by a body casting having a generally cylindrical interior chamber, and an end cap having a cylindrical boss which telescopes into the end of the body and is sealed thereto by an O-ring 4. The end wall 5 of the body opposite the end cap 4 includes a bore through which the pump-operating shaft 6 extends. Shaft 6 is supported for rotation in this bore by a bearing (not shown) which is secured against axial movement in the bore. Shaft 6 extends from the body 1 into end cap 2 and is carried for rotation therein by a needle-type roller bearing 7 mounted within a central bore in the end cap.

Cylindrical boss 3 of the end cap is finished to form a flat inner surface which is clamped against a side or radial face 8 of a cam ring 9. It may be mentioned here that the cam ring itself as well as the housing and cam ring together are sometimes referred to in the art as a stator.

A fluid intake passageway 10 extends radially into body 1 and communicates with a pair of internal annular channels 11, 12 which encircle the internal cavity within the body. These annular channels 11, 12 distribute fluid from the intake passageway 10 to suction ports to be described.

The cam ring 9 is supported radially by an annular rib 13 formed in the body 1 between the annular channels 11, 12. The cam ring 9 encircles a rotor 14 which is connected to shaft 6 through a motion-permitting spline joint 15 that permits proper running alignment between the rotor, the flat surface of the cylindrical boss 3, and a movable check plate 16. As can best be seen in FIG. 2, the rotor 14 is provided with a plurality of radial vane slots 17 in each of which a vane 18 is mounted.

The cam ring 9 has a cam surface 19 that is contoured to provide a balanced pressure pump construction in which there are diametrically opposite low pressure or suction zones 20, fluid transfer zones 21, high pressure or exhaust zones 22, and sealing zones 23 formed between the cam surface 19 and the rotor 14 (see FIG. 2). In order to provide the
opposed zones, the cam surface 19 is formed in part from a first pair of arcs of equal radii which extends across the fluid transfer zones 21 and, in part by a second pair of arcs of shorter radii than the first pair of arcs which extends across the sealing zones 23. These pairs of arcs are interconnected by cam surfaces which extend across the low- and high-pressure zones 20 and 22 respectively.

Cheek plate 16 is fitted to provide a smooth flat surface on the inner side thereof which abuts the cam ring 9, and has a central bore 24 surrounded by a cylindrical boss 25 which extends into the bore in the wall 5 of the body 1 and is sealed thereto by an O-ring 26. The outer cylindrical surface of cheek plate 16 is sealed to body 1 by an O-ring 27 and is urged into sealing engagement by a spring (not shown herein) described and shown in U.S. Pat. No. 3,076,414, issued Feb. 5, 1963 and assigned to the assignee of this application.

The cheek plate 16 is movable axially in the body 1 and is urged toward rotor 14 by fluid pressure supplied from the high-pressure zone 22 through passageway 28 and orifice 29 to a pressure chamber 30 formed between the body and the outer face 31 of the cheek plate. The cheek plate functions in the nature of an axially movable, nonrotatable piston under the pressure supplied by the fluid in chamber 30 to maintain in engagement with the adjacent side face of the cam ring 9. A light spring (not shown) biases the cheek plate axially toward engagement with the adjacent side face of the cam ring 9.

Intake passageway 10 communicates through annular channels 11, 12 around the cam ring 9 to suction ports spaced 180° apart. Two suction ports, one of which is shown at 50 in FIG. 2, are formed in cheek plate 16 and are fed by channel 12, and two additional suction ports (not shown) are formed in end cap 42 and are fed by channel 11. These suction ports in the end cap and cheek plate are identical in shape and are axially aligned with the suction zones 20 between rotor 14 and cam surface 19. Each suction port has a branch passage, the opening of one of which is designated at 51, whereby the suction port communicates with the inner ends 32 of vane slots 17 in the rotor 14 as well as with the inlet zones 20.

As shown in FIG. 1, the end cap 2 includes two diametri-
cally opposed crescent-shaped exhaust or pressure ports 52, 53 which are spaced substantially 90° from the suction ports. Similarly, pressure ports 56, 57 are formed in cheek plate 16 which are axially aligned with the pressure zones 22 and with ports 52, 53 in the end cap. Each pressure port 52 and 56 communicates with the inner ends 32 of the vane slots 17 in the rotor 14. These ports pass the pressure fluid in the inner end 32 of the vane slots 17 through a branch passageway 54. Pressure ports 52, 53 are connected with a fluid outlet or delivery chamber 34 by a passageway 35 in the end cap 2.

In the direction of rotor movement (clockwise as shown by the arrow in FIG. 2), the cam surface 19 progressively recedes from the periphery of the rotor 14 across the suction zones 20. In the transfer zones 21 cam surface 19 has a nearly constant radial spacing from the rotor, and across the exhaust zones 22 the cam surface progressively approaches the rotor 14 so as it comes into close proximity with the periphery of the rotor 14 in the sealing zones 23. Fluid from the suction ports 50 is drawn into the fluid transport pockets defined between the successive vanes as each pocket becomes larger when the vanes 18 move through the suction zones 20. The fluid is posi-
tively displaced from the pockets as the volume thereof diminishes when the vanes move through the pressure zones 22, to thereby effect a pumping action.

Each vane 18 is provided with grooves 36 which are formed in its outer edge and opposite side edges. One or more chan-
nels or bores 37 are also provided in each vane which commu-
nicate between the outer groove 36 and the inner end 32 of the vane slot. The grooves 36 and channels 37 insu-
are that the fluid pressure acting on the first area or outer end sur-
face or tip of any given vane will be substantially balanced at all times by the pressure acting on the second area or inner end surface of that vane.

For the pump to operate at high efficiency it is necessary to maintain a continuous sealing engagement between the vane tips or outer end surface with the cam surface 19, regardless of changes in the arcuteness of the cam surface. To provide the hydraulic pressure for this purpose, one or more radial bores or piston cylinders 38 is formed in the rotor 14, extending inwardly from the inner end 32 of each vane slot 17. The bores 38 are interconnected at their inner ends with an annular pres-

10 sure chamber 57 having fluid under pressure therein. Thus, fluid can flow into and out of pressure chamber 57 only through radial bores 38.

As best seen in FIG. 2, the pressure chamber 57 commu-
nicates with the bores 38 through orifices or flow restrictors 16 having cross-sectional dimensions which are all relative to the diameter of the bores 38. Within the pressure chamber 57, and intermediate the flow restrictors 46, are fluid velocity reducing chambers 58. The pressure chamber 57 is con-
structed, in part, by defining in rotor 14 an annular groove 46. The chambers 58 are formed in a sleeve 40. Thereafter the sleeve is fitted and sealed in an axial recess in rotor 14 with the chambers 58 intermediate the annular groove 46, thus form-
ing the pressure chamber 57. Hence, the pressure chamber 57 includes the annular groove 46 and the chambers 58.

A generally cylindrical pin or piston valve element 41 is received in each radial bore 38. Each piston 41 includes an axial bore 42 and is slidable in its cylinder 38 with which it is closely fitted so that leakage of fluid along the external walls of the piston is less than on the cam ring face. The rear end of each piston 41 is conically tapered as at 43 and forms a valve with the flat inner edge surface 44 of each vane 18. The lower end surface of each piston is preferably chamfered, as at 47, and presents a surface with which the fluid pressure force within the chamber 39 may cooperate to force the piston outwardly against the inner edge surface 44 of each vane. The admission of fluid to the radial bores 38 is regulated by the balance of forces between the fluid pressure force acting inwardly upon the conical taper 43, tending to open the valve, and the opposing force arising from the fluid pressure in chamber 39 and the centripetal force, tending to close the valve. The length of the piston 41 is such as to permit it to move into and out of en-
gagement with the flat inner edge surface 44 of the vane 18, regardless of the position of the vane in its slot.

In operation, valve 41, 44 at the outer end of the piston functions in the manner of a check valve. When fluid pressure at the inner end 32 of a vane slot acting upon the conical taper 43 of the piston 41 sufficiently exceeds the pressure in chamber 39, the piston is moved inwardly in its bore 38 and forces the fluid in the inner end 32 of the vane slot 17 flows inwardly through bore 42 toward chamber 57 and restores, maintains, or increases the fluid pressure in the pressure chamber as necessary to balance the fluid pressure acting to open the valve 41, 44. This action occurs when the vane slot 17 traverses a pressure zone 22, for fluid pressure in the zone 22 is usually the highest in the pump and the pressure in chamber 57 is somewhat lower. When a vane 18 is traversing a suction zone 20, the fluid pressure in chamber 57 exceeds the opposing fluid pressure on end 43 of piston 41, and the piston is held against the inner end 44 of the vane, to close valve 41, 44 and to urge the vane 44 radially outwardly in its slot 17.

The pump per se heretofore described is well known and is completely described in U.S. Pat. No. 3,401,641, issued Sept. 17, 1968, and U.S. Pat. No. 3,076,414, issued Feb. 5, 1963, and assigned to the assignee of this application. It forms no part of the invention of this application except in combination with the novel valve 60 hereinafter completely described.

The invention of this application consists of the provision of a control valve in combination with the pump. This valve is connected to the pressure chamber 30 and is operative to re-
lieve sudden pressure surges in the chamber 30 before those pressure changes can result in damage to the pump. Pressure surges of the type to be relieved by valve 60 often far exceed the pressure setting of any relief valve in the hydraulic system which is being supplied with fluid by the pump. Because of their suddenness these short duration pressure peaks cannot
be relieved by conventional pressure relief valves, because these valves cannot react quickly enough to relieve the condition. The valve 60 functions in cooperation with check plate 16 to momentarily seal one of the pump delivery back to the suction or inlet port, to relieve the pressure peaks in the pump outlet 34, and thereby minimize or eliminate damage to the pump and other system components which would otherwise result from these pressure peaks.

Valve 60 is located within the casing 1 and comprises a bore 61 connected at its inner end by a passage 62 to the pressure chamber 30. A shoulder 63 is defined by a stepped section of the bore 61 between a small diameter inner section 64 and a larger diameter intermediate section 65. The outer end of the section 65 of the bore is threaded as at 66 and accommodates a threaded closure or sealing plug 67.

A piston 68 is slidably mounted within the intermediate section 65 of the bore. It has a conically shaped valve or valve closure 69 formed on its inner end. This conically shaped end section or valve 69 is engageable with the shoulder 63 to form a seal between the end section 64 and the intermediate section 65 of the bore. In other words, the shoulder 64 acts as a valve seat and the conical end section 69 of the piston 68 acts as a valve closure between the end section 64 and the intermediate section 65 of the bore section 61.

The end section or chamber 64 is connected by the conduit 62 to the high-pressure chamber 30 and the intermediate section or chamber 65 is connected by a conduit 70 to the inlet port or suction port 10 of the pump.

A closed fluid compression chamber 71 is defined by or between the inner end 72 of the plug 67 and the outer end of the piston 68. This chamber 71 is enlarged by a central bore 75 which extends from the outer end of the piston to a point adjacent to the inner end. An orifice 79 in check plate 16 connects chamber 30 to the source of pressure at outlet port 56, and a small restricted orifice 76 connects the chamber 71 to the pressure chamber 64, so that under static conditions or when pressures in the system change slowly, the pressure behind the piston 68 urging it inwardly to close the valve seat 63 rises and falls in unison with the control pressure in the chamber 30 which urges the check plate into engagement with the cam ring. This is the same pressure as that on the exhaust port or pressure port of the pump and is the highest pressure in the pump, except when pressure in the exhaust port changes rapidly.

The area of the piston on the outside or valve closure side 73 of the piston 68 is slightly greater than the area on the inside of the piston and the hydraulic fluid is supplied to the side of pressure chamber 30. Consequently, the net force differential lightly urges the valve 69 to a closed condition. To further assist this small differential force in maintaining the valve closed and to hold the valve closed when there is no fluid in the valve 69, a low force spring 78 is located between the inner end 72 of the plug and outer end of the piston. The spring 78 also functions in cooperation with orifice 76 to control the rate of closing of the valve 69 after it has been opened. It should be noted that neither the small fluid force differential nor the spring offer much resistance to opening the valve 69.

In operation, the rate of pressure gain control valve 60 remains closed at all times except upon the occurrence of a very high rate of pressure increase or gain. In the event of a sudden pressure gain in the pressure port 52 and then in the chamber 30, pressure in the chamber 64 increases at a faster rate than it can be equalized by flow of fluid through the restricted orifice 76. In other words, the event that a sudden stoppage of flow in the system tends to cause a very sudden pressure increase, flow through the restricted orifice 76 is insufficient to provide immediate pressure equalization on opposite sides of the piston 68. Since the hydraulic fluid is slightly compressible, the higher pressure in the chamber 64 causes the fluid in the chamber 71 to be compressed by stroking of piston 68, and thereby opens the valve 69. This allows fluid from the chamber 30 to spill through passage 62, chamber 64, and the valve seat 63 to the intake or suction port 10. Since chamber 30 is supplied by fluid flowing from pressure port 56 through restricted orifice 29, a loss of fluid from chamber 30 creates a pressure differential on the opposite side of check plate 16. Therefore, the check plate 16 will move away from the cam ring 9 which causes a momentary short circuit of fluid from the outlet zones of the pump. The volume of fluid being pumped to the system is momentarily reduced and therefore results in a reduction of the rate of pressure increase in the system.

After the high rate of pressure rise has subsided, the spring 78 is able to reclose the valve 69 as sufficient flow is able to pass through the orifice 76 into the chamber 71. The rate of closing can be controlled by the strength of the spring 78, the size of the orifice 76 and the volume of chamber 71. The check plate is now free to move back relatively slowly by flow-entering chamber 30 through orifice 29, and the check plate will again be clamped in sealing engagement with the cam ring 9, which stops the temporary short circuit.

If desired, the rate of pressure gain to which the valve is responsive may be made adjustable or variable by substituting a threaded screw for the tap and plug 67. By varying the size of the compression chamber 71 with this screw, the volume of fluid contained in the chamber 71 may be varied and the rate of pressure gain required to compress the fluid to the degree required to open the valve 69 may be altered.

It should be understood that the check plate 66 described and illustrated in U.S. Pat. No. 3,076,414 could readily be substituted for the check plate 16 illustrated herein and thereby obtain the advantages of this invention, as well as the advantages of the invention claimed in U.S. Pat. No. 3,076,414.

While I have described only a single preferred embodiment of my invention, those persons skilled in the arts to which this invention pertains will readily appreciate numerous changes and modifications which may be made without departing from the spirit of my invention. Therefore, I intend to be limited only by the scope of the appended claims.

I claim:

1. A fluid energy translating device having a zone of high pressure and a zone of low pressure, an inlet and an outlet for said zone of high pressure and zone of low pressure, said device including a housing, rotary translating means, a stator encompassing said translating means in said housing, a che"
the valve seat than is presented to the action of the same pressure on the side of the valve closure element located adjacent the valve seat so that the net differential force acting against said valve closure element cooperates with said spring to bias said valve closure element to a normally closed position, said restricted passageway means being of such a size that it restricts the passage of fluid therethrough so that fast rates of pressure increase in said inlet cause said valve to open and spill fluid to said low-pressure zone through said valve seat.

3. The energy-translating device of claim 2 wherein said restricted passageway is located within said piston.

A fluid energy translating device having a zone of high pressure and a zone of low pressure, an inlet and an outlet for said zone of high pressure and zone of low pressure, said device including a housing, a rotary translating means movable within said housing, and a pressure-loaded loaddown element movably mounted within said housing, a fluid chamber defined on one side of said element, said chamber being connected to a source of high pressure so that fluid under pressure in said chamber urges said element to a pressure-loaded position, fluid passage means connecting said pressure chamber to a zone of low pressure, and valve means including a movable valve closure element mounted within said passage means, said valve means being responsive to excessive rates of pressure gain in said pressure chamber to open and dump fluid from said chamber to said low-pressure zone.

5. The device of claim 4 wherein said fluid energy translating device comprises a rotary vane pump and wherein said pressure-loaded loaddown element comprises a check plate of said pump.

6. The energy-translating device of claim 4 wherein said rate of pressure gain responsive valve comprises a valve chamber bore communicating at an inlet with fluid passage means and at an outlet point with said low-pressure zone, said inlet point and said outlet point of said bore being spaced apart, a valve seat in said bore between the points of communication thereof with said inlet and said outlet, said movable valve closure element being resiliently urged by a spring toward said valve seat, said valve closure element including a piston movably mounted within said valve chamber bore, restricted passageway means interconnecting the portions of said bore on opposite sides of said piston to the inlet point of said bore, the end of said valve chamber bore located on the side of said piston which is spaced away from said valve chamber, the end of said valve chamber located on the side of said piston which is spaced away from said valve seat being closed except for said restricted passageway means, said valve closure presenting a larger area to action of the pressure of said pressure chamber on the side away from the valve seat than is presented to the action of the same pressure on the side of the valve closure located adjacent the valve seat so that the net differential force acting against said valve closure cooperates with said spring to bias said valve closure to a normally closed position, said restricted passageway means being of such a size that it restricts the passage of fluid therethrough so that fast rates of pressure increase in said inlet cause said valve to open and spill fluid to said low-pressure zone through said valve seat.

11. The valve of claim 10 wherein the rate of pressure change responsive valve closure is generally cone shaped and said valve seat is circular in cross section.

12. The valve of claim 10 wherein said restricted passageway extends from said inlet point with said low-pressure zone.

13. A hydraulic pump including a stator, a rotor within said stator, a zone of high pressure and a zone of low pressure defined between said stator and said rotor, an inlet and an outlet for said zone of low pressure and zone of high pressure, a check plate which is movable with respect to said stator and said rotor in a direction parallel to the axis of said rotor, one side of said check plate normally abutting said stator, pressure in said zone of high pressure tending to urge said check plate away from said stator, said check plate separating said zone of high pressure from said zone of low pressure, wall means defining a pressure chamber on the side of said check plate which is opposite to said rotor, said check plate forming one wall of said chamber, fluid pressure in said chamber tending to urge said check plate away from said chamber, fluid passage means connecting said chamber at all times to said zone of high pressure, second passage means connecting said chamber to said zone of low pressure, and a normally closed valve means located in said second passage means, said valve means being responsive to excessive rate of pressure gain in said chamber to open and spill fluid from said chamber to said low-pressure zone.

14. The pump of claim 13 wherein said valve means is operable to respond to pressure rates of change irrespective of the initial pressure at which such change is initiated if such change exceeds a predetermined value.

15. The pump of claim 13 wherein said rate of pressure gain responsive valve means comprises a valve chamber communicating at an inlet point with said second fluid passage means and at an outlet point with said low-pressure zone, said inlet point and said outlet point of said valve chamber being spaced apart, a valve seat in said valve chamber between the points of communication thereof with said inlet and said outlet, movable valve closure means resiliently urged by a spring toward said valve seat, said valve closure means including a piston movably mounted within said valve chamber, restricted passageway means interconnecting the portions of said valve chamber on opposite sides of said piston to the inlet point of said valve chamber, the end of said valve chamber located on the side of said piston which is spaced away from said valve.
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16. The energy-translating device of claim 15 wherein said restricted passageway is located within said piston.
17. A fluid energy translating device having a zone of high pressure and a zone of low pressure, an inlet and an outlet for said zone of high pressure and zone of low pressure, said device including a housing, a rotary translating means movable within said housing, translating means including fluid displacement cavities and a pressure-loaded fluid sealing element movably mounted within said housing, a fluid chamber defined on one side of said element, the opposite side of said element being exposed to said displacement cavities, each of said displacement cavities alternately being exposed to said zones of high pressure and zones of low pressure, said chamber being connected through an orifice to a source of high pressure so that fluid under pressure in said chamber urges said element to a position that results in a seal between said zone of high pressure and said zone of low pressure, fluid passage means connecting said pressure chamber to a zone of low pressure, and a valve including a movable valve closure element mounted within said passage means, said valve closure element being responsive to excessive rates of pressure gain in said pressure chamber to momentarily open and bypass fluid from said chamber to said low-pressure zone, thereby permitting said fluid sealing element to move to a position whereby fluid from those displacement cavities which are located in the high-pressure zone is bypassed from said zone of high pressure to said zone of low pressure.
18. The energy-translating device of claim 17 wherein said device includes a housing, rotary translating means having fluid displacement cavities therein, said cavities being alternately exposed to said zones of high pressure and low pressure, a stator encompassing said translating means in said housing, a check plate which is movable in said housing with respect to said stator and said translating means in a direction parallel to the axis of said translating means, wall means defining a pressure chamber on the side of said check plate which is opposite to said translating means, said check plate forming one side wall of said chamber, the opposite of said check plate being exposed to said displacement cavities, fluid under pressure in said chamber urging said check plate toward said stator and translating means to effect a seal between said zones of high pressure and low pressure, a fluid passage means interconnecting said chamber with a low-pressure zone, and a normally closed valve including a movable valve closure element located in said passage means, said valve being operable to open and dump fluid from said chamber to said low-pressure zone when pressure gain in said chamber exceeds a safe operating condition of said fluid energy translating device, the dumping of fluid from said chamber being operable to cause said check plate to move away from and out of sealing engagement with said translating means whereby fluid from those cavities which are located in the high-pressure zone is bypassed from said high-pressure zone to said low-pressure zone.
19. The energy-translating device of claim 21 wherein said device comprises a valve chamber bore communicating at an inlet point with said fluid passage means and at an outlet point with said low-pressure zone, said inlet and said outlet of said bore being spaced apart, a valve seat in said bore between the points of communication thereof with said inlet and said outlet, valve closure element including a piston movably mounted within said valve chamber bore, restricted passageway means interconnecting the portions of said bore on opposite sides of said piston to the inlet point of said bore, the end of said valve chamber bore located on the side of said piston which is spaced away from said valve seat being closed except for said restricted passageway means, said valve closure element presenting a larger area to action of the pressure of said pressure chamber on the side away from the valve seat then is presented to the action of the same pressure on the side of the valve closure element located adjacent the valve seat so that the net differential force acting against said valve closure element biases said valve closure element to a normally closed position, said restricted passageway means being of such a size that it restricts the passage of fluid therethrough so that only fast rates of pressure increases in said inlet cause said valve to open and spill fluid to said low-pressure zone through said valve seat.
20. The energy-translating device of claim 18 which further includes a resilient spring for biasing said valve closure element into a closed position.
21. A fluid energy translating device having a zone of high pressure and a zone of low pressure, an inlet and an outlet for said zone of high pressure and zone of low pressure, said device including a housing, rotary translating means having fluid displacement cavities therein, said cavities being alternately exposed to said zones of high pressure and low pressure, a stator encompassing said translating means in said housing, a check plate which is movable in said housing with respect to said stator and said translating means in a direction parallel to the axis of said translating means, wall means defining a pressure chamber on the side of said check plate which is opposite to said translating means, said check plate forming one side wall of said chamber, the opposite of said check plate being exposed to said displacement cavities, fluid under pressure in said chamber urging said check plate toward said stator and translating means to effect a seal between said zones of high pressure and low pressure, a fluid passage means interconnecting said chamber with a low-pressure zone, and a normally closed valve including a movable valve closure element located in said passage means, said valve being operable to open and dump fluid from said chamber to said low-pressure zone when pressure gain in said chamber exceeds a safe operating condition of said fluid energy translating device, the dumping of fluid from said chamber being operable to cause said check plate to move away from and out of sealing engagement with said translating means whereby fluid from those cavities which are located in the high-pressure zone is bypassed from said high-pressure zone to said low-pressure zone.
22. The energy-translating device of claim 21 wherein said device comprises a valve chamber bore communicating at an inlet point with said fluid passage means and at an outlet point with said low-pressure zone, said inlet point and said outlet point of said bore being spaced apart, a valve seat in said bore between the points of communication thereof with said inlet and said outlet, valve closure element including a piston movably mounted within said valve chamber bore, restricted passageway means interconnecting the portions of said bore on opposite sides of said piston to the inlet point of said bore, the end of said valve chamber bore located on the side of said piston which is spaced away from said valve seat being closed except for said restricted passageway means, said valve closure element presenting a larger area to action of the pressure of said pressure chamber on the side away from the valve seat than is presented to the action of the same pressure on the side of the valve closure element located adjacent the valve seat so that the net differential force acting against said valve closure element biases said valve closure element to a normally closed position, said restricted passageway means being of such a size that it restricts the passage of fluid therethrough so that only fast rates of pressure increases in said inlet cause said valve to open and spill fluid to said low-pressure zone through said valve seat.
23. The energy-translating device of claim 22 which further includes a resilient spring for biasing said valve closure element into a closed position.
24. The energy-translating device of claim 23 wherein said restricted passageway means is located within said piston.
25. A hydraulic pump including a stator, a rotor within said stator, a zone of high pressure and a zone of low pressure defined between said stator and said rotor, an inlet and an outlet for said zone of low pressure and zone of high pressure, a fluid-sealing element which is movable with respect to said stator and said rotor to effect a seal between said zone of high pressure and zone of low pressure, one side of said element partially defining a pressure chamber for urging said sealing element into engagement with at least one of said rotor and stator, fluid passage means communicating between said pressure chamber and a zone of low pressure, a valve including a movable valve closure located within said passage means, said valve being responsive to excessive rates of pressure increase in said chamber to open and dump fluid from said chamber to said low-pressure zone, the dumping of fluid from said chamber being operable to cause said valve to move away from and out of sealing engagement with said one
of said rotor and stator whereby fluid is bypassed between said element and said one of said rotor and stator from said zone of high pressure to said zone of low pressure.

26. The hydraulic pump of claim 25 wherein said element comprises a cheek plate which is movable with respect to said stator and rotor in a direction parallel to the axis of said rotor.

27. The pump of claim 25 wherein said valve comprises a valve chamber bore communicating at an inlet point with said fluid passage means and at an outlet point with said low-pressure zone, said inlet point and said outlet point of said bore being spaced apart, a valve seat in said bore between the points of communication thereof with said inlet and said outlet, said valve closure element including a piston movably mounted within said valve chamber bore, restricted passageway means interconnecting the portions of said bore on opposite sides of said piston to the inlet point of said bore, the end of said valve chamber bore located on the side of said piston which is spaced away from said valve seat being closed except for said restricted passageway means, said valve closure element presenting a larger area to action of the pressure of said pressure chamber on the side away from the valve seat than is presented to the action of the same pressure on the side of the valve closure element located adjacent the valve seat so that the net differential force acting against said valve closure element biases said valve closure element to a normally closed position, said restricted passageway means being of such a size that it restricts the passage of fluid therethrough so that only fast rates of pressure increases in said inlet cause said valve to open and spill fluid to said low-pressure zone through said valve seat.

28. The energy-translating device of claim 27 which further includes a resilient spring for biasing said valve closure element into a closed position.

29. The energy-translating device of claim 28 wherein said restricted passageway means is located within said piston.

30. A hydraulic pump including a stator, a rotor within said stator, a zone of high pressure and a zone of low pressure defined between said stator and said rotor, an inlet and an outlet for said zone of low pressure and zone of high pressure, a cheek plate which is movable with respect to said stator and said rotor in a direction parallel to the axis of said rotor, one side cheek plate normally abutting said stator, said cheek plate sealingly separating said zone of high pressure from said zone of low pressure, wall means defining a pressure chamber on the side of said cheek plate which is opposite to said rotor, said cheek plate forming one wall of said chamber, fluid pressure in said chamber tending to urge said cheek plate toward said rotor to seal said zone of high pressure from said zone of low pressure, first fluid passage means connecting said chamber at all times to said zone of high pressure, second passage means connecting said chamber to said zone of low pressure, and a normally closed valve located in said second passage means, said valve being responsive to excessive rate of pressure gain in said chamber to open and spill fluid from said chamber to said low-pressure zone, the spilling of fluid from said chamber being operable to permit said cheek plate to move away from said rotor and thereby bypass fluid directly from said zone of high pressure to said zone of low pressure.

31. The pump of claim 30 wherein said valve is operable to respond to pressure rates of change irrespective of the initial pressure at which such change is initiated if such change exceeds a predetermined value.

32. The energy-translating device of claim 30 wherein said valve comprises a valve chamber bore communicating at an inlet point with said second fluid passage means and at an outlet point with said low-pressure zone, said inlet point and said outlet point of said bore being spaced apart, a valve seat in said bore between the points of communication thereof with said inlet and said outlet, said valve closure element including a piston movably mounted within said valve chamber bore, restricted passageway means interconnecting the portions of said bore on opposite sides of said piston to the inlet point of said bore, the end of said valve chamber bore located on the side of said piston which is spaced away from said valve seat being closed except for said restricted passageway means, said valve closure element presenting a larger area to action of the pressure of said pressure chamber on the side away from the valve seat than is presented to the action of the same pressure on the side of the valve closure element located adjacent the valve seat so that the net differential force acting against said valve closure element biases said valve closure element to a normally closed position, said restricted passageway means being of such a size that it restricts the passage of fluid therethrough so that only fast rates of pressure increases in said inlet cause said valve to open and spill fluid to said low-pressure zone through said valve seat.

33. The pump of claim 32 which further includes a resilient spring for biasing said valve closure element into a closed position.

34. The energy-translating device of claim 33 wherein said restricted passageway means is located within said piston.
UNIVERSAL STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,578,888 Dated May 18, 1971

Inventor(s) Cecil E. Adams

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Column 9, line 52, after "piston" and before "is", insert -- which --.

Column 10, line 10, after "opposite" and before "of", insert -- side --.

Column 10, line 55, after "includes", change "sa" to -- a --.

" 10, " 70, after "within", delete one of the "said"s.

" 11, " 43, after "one side", insert -- of said --.

Signed and sealed this 19th day of October 1971.

(SEAL)
Attest:

EDWARD M. FLETCHER, JR. ROBERT GOTTSCHALK
Attesting Officer Acting Commissioner of Patents