Aug. 25, 1964 K. EICKMANN 3,145,662

DEVICE FOR DAMPING VIBRATIONS, FLUCTUATIONS OF PRESSURE AND FLUCTUATION OF DELIVERY VOLUME OR SUCTION VOLUME OF ROTARY PISTON MACHINES

Filed Sept. 6, 1960 2 Sheets-Sheet 1

Fig. 2

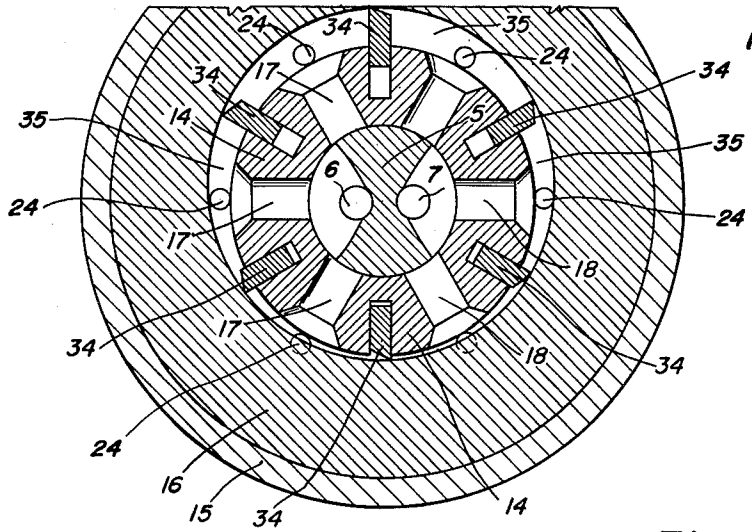
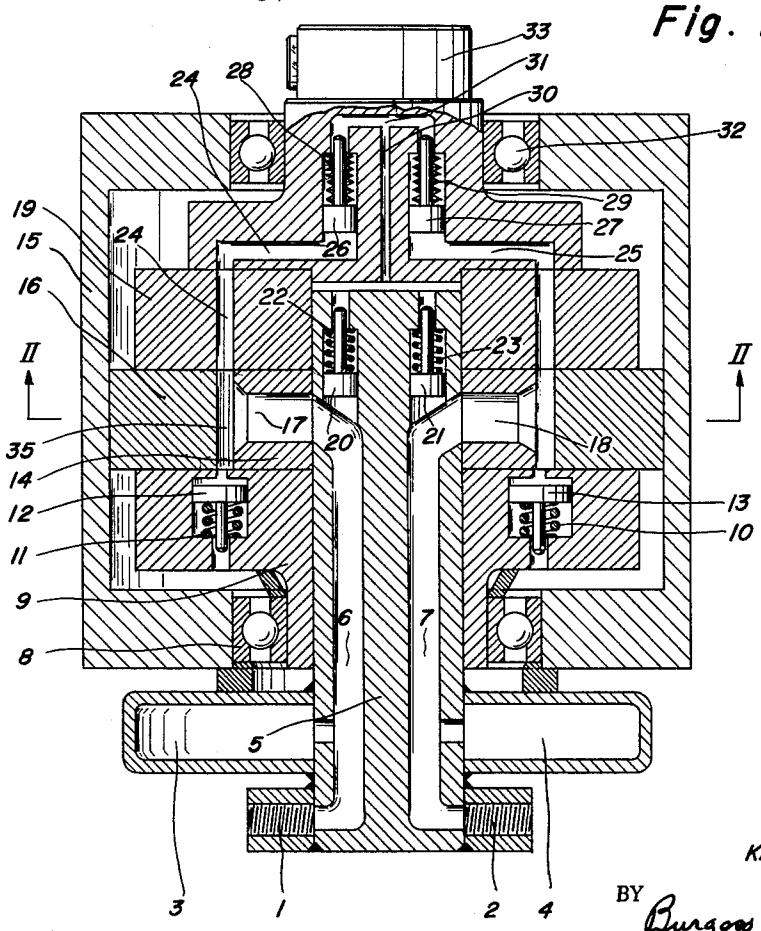


Fig. 1



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Fig. 3

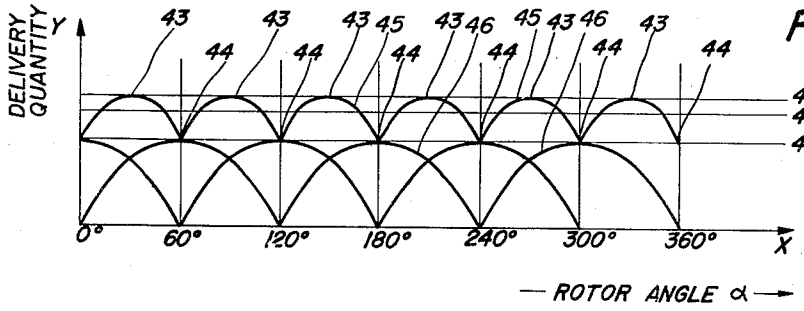


Fig. 4

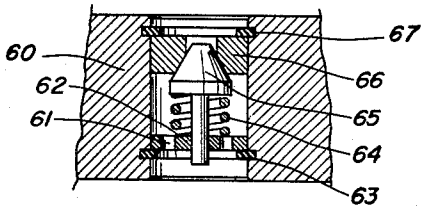


Fig. 5

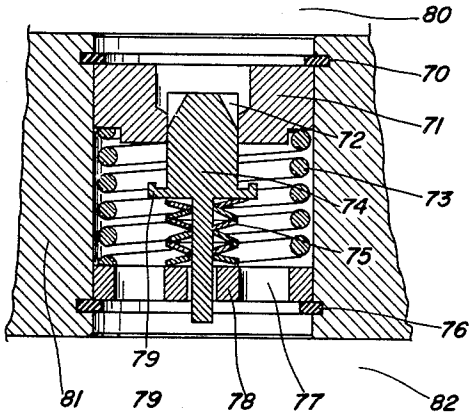
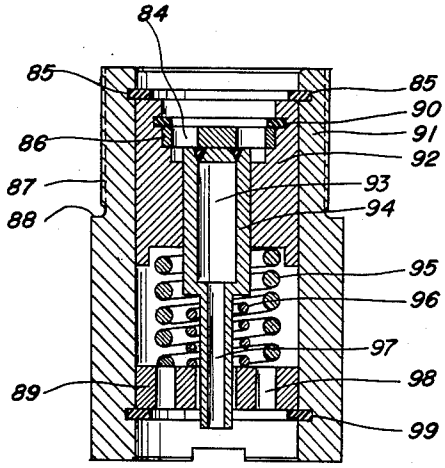


Fig. 6



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DEVICE FOR DAMPING VIBRATIONS, FLUCTUATIONS OF PRESSURE AND FLUCTUATION OF DELIVERY VOLUME OR SUCTION VOLUME OF ROTARY PISTON MACHINES

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The present invention relates to a device for damping vibrations, and fluctuations of pressure and delivery volume or suction volume in rotary piston machines, particularly of radial and axial type piston and vane cell or intervane space machines. More particularly, the invention relates to the provision for damping means in the fluid volume space of such rotary piston machines for damping the fluid pressure therein by adjusting the volume during peak fluctuations.

Rotary piston machines of the radial and axial type having piston cells or vane cells, i.e. intervane spaces, therein normally operate with a swinging slant plate or an eccentrically disposed ring so as to drive the vane or pistons in the rotor of the machine. In this connection, as is commonly known, the individual vanes or pistons work so as to increase or decrease the volume of the working cells or intervane spaces of the machine during each rotation of the rotor. The movement of the vanes or pistons during the rotation of the rotor assume the form of a sine curve corresponding to the inclined angle and eccentricity of their particular guide means.

Moreover, where a plurality of pistons or vanes is provided, the delivery volume of each piston and vane is superimposed in dependence upon the movement of the pistons and vanes and the increase or decrease of the volume of the working cells or intervane spaces is produced in that manner. Nevertheless, where the volumes of a plurality of cells or intervane spaces are superimposed, a delivery volume curve which never fluctuates cannot be obtained. The delivery volume will fluctuate between a maximum and minimum value in dependence upon the number of vanes or pistons in the rotor. Accordingly, the higher the number of vanes or pistons, the less will be the fluctuation of the delivery volume. Consequently, the operation of the machine will be kept comparatively quiet. This awareness has led to the manufacture of rotary piston or vane machines having increasing numbers of pistons or vanes in the rotor.

It is an object of the present invention to provide a damping cushion means for a rotary piston or vane machine for damping the fluid pressure in the machine by adjusting the volume during peak fluctuations.

Other and further objects of the invention will become apparent from a study of the within specification and accompanying drawings in which,

FIG. 1 is a sectional view of a rotary vane machine provided with damping means in accordance with the invention.

FIG. 2 is a cross-sectional partial view taken along the line II—II of the apparatus of FIG. 1,

FIG. 3 is a graph representing the curve of the theoretical delivery volume of a rotary vane pump, having six intervane spaces, and

FIGS. 4–6 are longitudinal sectional views of damping valves in accordance with different embodiments of the invention applicable for cushioning exceedingly high fluctuating frequencies.

In accordance with the invention, it has been found that efficient damping of vibrations and fluctuations in pressure and volume of a rotary vane or piston machine may be provided by employing in the fluid volume space of the machine damping means responsive to the fluid pressure fluctuation in the fluid volume space which is caused by the fluid volume fluctuation, so as to dampen the fluid pressure by adjusting the volume during peak fluctuations. The damping means of the invention may be used in connection with all types of rotary piston or vane machines having radially or axially disposed piston or vane cells, i.e. intervane spaces, such as internal combustion, steam, liquid, compressed air and gas rotary piston or vane machines as well as liquid and air pressure pump machines of this type, etc.

Thus, in accordance with the invention, the damping means are positioned in the fluid volume space of a rotary vane machine. The machine generally has a rotor operably provided with vanes. The rotor is capable of rotation within the casing of the machine. The rotor and vanes generally define with the casing a plurality of cells or intervane spaces which increase and decrease in volume during the rotation of the rotor and vanes due to the provision in the casing of eccentric means, such as an eccentric sliding ring. Generally, a shaft having an inlet conduit for passing fluid to each of the intervane spaces and an outlet conduit for passing fluid from each of the intervane spaces is provided for distributing and withdrawing the pressure fluid during the rotation of the rotor. The conduits and intervane spaces of the machine, therefore, define a fluid volume fluctuation space during the rotation such that while certain intervane spaces increase in volume, other intervane spaces correspondingly decrease in volume to attain the desired operation.

Preferably the damping means are provided in immediate and direct flow communication with the intervane spaces and inlet and outlet conduits of the shaft. A damping cushion or expansion chamber is also preferably provided remote from the rotor in flow communication with the shaft inlet conduit and separately in flow communication with the shaft outlet conduit for further cushioning peak fluctuations.

The damping means may take the form of fluid pressure responsive piston means slidable within cylinder means so as to increase and decrease the fluid volume fluctuation space. The damping means may also take the form of closure valve means whereby upon peak fluctuations, the valve will open allowing the passage of fluid therethrough so as to decrease the pressure.

While it is true that the higher the number of vanes or pistons in the rotary machine, the less will be the fluctuation of the delivery volume and therefore the quieter the operation, it is to be noted that there is a limit to the number of pistons or vanes which may be provided in the rotor beyond which no further advantages will be forthcoming. Among the considerations involved in this regard is the fact that there is often insufficient spaced for disposing the pistons and vanes in the desired number. Moreover, the larger the number of pistons and vanes, the greater will be the manufacturing expenses of the machine as well as the operating cost of the machine. Inasmuch as a larger number of pistons and vanes may bring about a leakage and occasionally high operational friction, the benefits are correspondingly offset by a loss in efficiency.

In accordance with the invention, the damping means may overcome and offset the undesired fluctuation of volume and pressure, as well as the attendant vibrations, even where only a small number of vanes and pistons are used. The damping means or damping cushion, in accordance with the invention is disposed in the fluid volume fluctuation space of the machine so as to increase the total volume in order to decrease the peak pressure and vice versa.

FIG. 3 shows a diagram of the theoretical curve of the delivery volume in an oil pump having six intervane

spaces. Curve 45 indicates the irregular fluctuation of the delivery volume which results in the irregular fluctuation of the pressure. It is typical for this irregular fluctuation of pressure that sharp, high pressure peaks are produced as a result of the fluctuation of the delivery volume. Such high pressure peaks exert an impulsive load on the bearings and parts of the machine, causing undesirable noisiness and vibration of the machine. These peaks may be leveled off and somewhat eliminated by disposing damping means in accordance with the invention such that the flow of the medium fluctuated by the high pressure is first partially forced into a flexible cushion area and then resiliently forced out therefrom in such a manner that the leveling of the delivery volume and pressure is attained.

This cushioning action may be carried out by means of a piston slidable within a cylinder and urged outwardly therefrom by spring action so that the increase in volume will be compensated for by movement of the piston so as to increase the fluid volume space, and therefore decrease the pressure. Upon the corresponding decrease in volume, the resiliently urged piston will change its direction of movement so as to close the cylinder and decrease the volume of fluid space. This action may also be performed by means of gas or air cushions in a pressure accumulating cell or expansion chamber or by providing liquid cushions in a pressure accumulating cell or expansion chamber. The gas, air, or liquid used is accordingly compressed and expanded with the changes in pressure of the machine.

Specifically, the fluctuation of the pressure and delivery volume may be substantially leveled off by means of a sufficiently large cushion cell or expansion chamber filled with liquid since, as it is known, liquid is not completely non-compressive. For example, oil, depending upon its type, may be compressed with a compression rate of $50\text{–}80\times10^{-6}$ cm.^{2}/kg. sq. cm. Another property of oil is the fact that its fluid propagates with comparatively high speed. In this regard, it is known that pressure waves are propagated through air at the speed of sound which is 330 meters per second, this quantity varying in dependence upon the particular conditions. On the other hand, the propagating speed of pressure, for example, in oil is markedly higher, i.e. around 500–1400 meters per second, depending upon the temperature, viscosity and density of the particular oil.

Thus, in accordance with the invention, an advantage has been taken of these phenomena such that sufficiently large expansion chambers may be filled with liquid where the same are disposed quite near to the source which generates the pressure fluctuation, so as to suitably level the fluctuation and cushion the rapid fluctuation of pressure and delivery volume of the liquid in consequence of the compressibility of the liquid and the comparatively high propagating speed of the pressure waves therein.

Generally, in accordance with the invention rotary machines may be operated at high revolutions, such as in the order of 1500 revolutions per minute. In the aircraft industry, as well as other industries, pressure generators or motors are required which must work at revolutions as high as 12,000 r.p.m. and in certain cases even as high as 100,000 r.p.m.

In rotary machines having uneven numbers of pistons and vanes, the frequency of maximum and minimum volume of delivery per rotation of the rotor is twice that where an even number of pistons and vanes is involved. This is true since the delivery volumes of the individual piston cells or vane cells or intervane spaces are superimposed upon one another. Consequently, for example, where the machine is operated at 12,000 revolutions per minute and the machine has 7 working cells or intervane spaces, i.e. an uneven number, the maximum and minimum values of the delivery volumes fluctuate 168,000 times per minute. Within such an interval of time as $\frac{1}{168,000}$ of a minute, pressure waves cannot be propagated over very far distances even if the propagating speed of the pressure in the fluid is high.

It is essential for the pressure waves to reach the damping cushion or damping means sufficiently rapidly to exert the timely cushioning or damping; otherwise the cushioning will be too late to prevent the occurrence of the undesired pressure peaks produced during the rotation of the rotor of the machine. Therefore, the pressure waves must be transmitted in proper phase for the desired cushioning action to take place.

In accordance with the invention, this is accomplished by providing the damping means as close as practicable to the place where the fluctuation is generated, i.e. in the rotor or in the shaft or control shaft of the rotary machine. In this manner the distance to the particular damping means will permit the fluctuating waves to be effectively transmitted from the working cells or intervane spaces and fluid pressure space in the desired proper phase of rotation.

Where the frequency of fluctuation is exceedingly high, a liquid cushion may be used so that the damping may take place sufficiently rapidly. A liquid damping cushion will, of course, work more rapidly than a gas-or-air-pressure cushion where high frequency fluctuations are involved.

Generally, the pressure fluctuation in the rotary piston or vane machine is directly related to the number of revolutions per unit time where the average delivery volume is the same. A higher number of revolutions will result in a higher frequency of pressure fluctuation. Where such revolution rate is exceedingly high, it is possible that the mass of the damping device, such as a spring loaded piston, disposed in a cylinder, may be too high to accommodate the frequency corresponding to the rapidly changing delivery pressure between the maximum and minimum values. In such instance, if it is not possible nor practical, or if there is insufficient space, to provide a liquid volume cushion or expansion chamber, as the damping means for controlling the volume of fluid in the control shaft or vane cell in the manner of a valve, the fluctuation of the delivery volume and pressure may be offset instead by providing a small escape valve having very little mass as the damping device.

Such escape valve may be constructed of a much smaller size than a spring-loaded piston damping cushion or damping device, and may also have a smaller amplitude, whereby a high rate of vibration of the escape valve may be carried out effectively. Although the escape valve has the drawback that the pressure liquid in the machine will escape through the valve during the peak period, thereby causing a lower efficiency, this loss of pressure liquid is comparatively negligible. In this connection, such escape valve is of particular importance and effect for damping vibrations and noise and is operable at high speeds whereby because of its small size, the same can be provided in the control shaft of the rotor. In this manner, the valve can act on both the delivery side and suction side of the control shaft. Since the delivery volumes of the individual intervane spaces begin to superimpose substantially at the shaft inlet conduit on the delivery side, by placing the escape valve adjacent this point high pressure peaks will not be generated and the motor will operate efficiently.

Accordingly, the damping device or escape valve will remarkably reduce the noise and fluctuation of the delivery volume and pressure. Hence, in accordance with the invention, rotary piston and vane machines may now be efficiently operated free from excessively large fluctuations of delivery volume and pressure heretofore encountered due to the kinetic and/or hydrodynamic forces generated during the rotation of the rotor.

Referring to FIGURES 1 and 2, a rotary vane machine is shown having a control shaft 5 which is provided with an inlet conduit 6, an outlet conduit 7, and tube connections 1 and 2 for the delivery and withdrawal of the

pressure liquid used in the machine. Casing 15 is provided with bearings 8 and 32 on which the rotor 14 is mounted via side portion 9 and side portion 19. Shaft 33 is provided on the rotor at the side portion 19. Rotor 14 revolves within eccentric slide ring 16 fixed to casing 15 so as to effect the increase and decrease in cell or intervane space volume. Side portions 9 and 19 as well as rotor 14 form a plurality of working cells or intervane spaces 35 with slide ring 16. Vanes 34 slidably disposed in rotor 14 divide the space between rotor 14 and slide ring 16 into a plurality of intervane spaces 35. Liquid is passed through connection tube 1, inlet conduit 6 and rotor conduit 17 to the intervane spaces 35. In the same way, liquid is withdrawn from the intervane spaces 35 through rotor conduits 18, outlet conduit 7, and connection 2.

Damping cells for expansion chambers 4 and 3 flow communicate with inlet conduit 6 and outlet conduit 7 respectively at a point in shaft 5 remote from rotor 14 while pistons 20 and 21 spring urged in forward direction by springs 22 and 23, respectively, flow communicate with conduits 6 and 7 at a point immediately adjacent rotor 14. Pistons 12 and 13, spring urged in forward direction by springs 10 and 11, flow communicate with corresponding intervane spaces 35 of the machine on one lateral side thereof, while pistons 26 and 27 spring urged in forward direction by plate springs 28 and 29 flow communicate with intervane spaces 35 by means of flow passages 25 and 24 respectively on the other lateral side of intervane spaces 35. The rear side of pistons 26 and 27 may optionally be in flow communication, via conduits 31 and 30, with the rear sides of pistons 20 and 21.

Thus, the damping means of the invention include damping cells 3 and 4 in the inlet and outlet conduits 6 and 7 remote from rotor 14 as well as damping pistons or valves 20 and 21 in flow communication with inlet conduits 6 and 7 immediately adjacent rotor 14 and damping pistons or valves 12, 13, 26, and 27 in immediate and direct flow communication with the intervane spaces 35.

In operation, suction fluid flows into conduit 6 via the outlet connection 1 such that any pressure peak in inlet conduit 6 is propagated into damping cushion cell 3 flow communicating therewith such that the pressure waves are exerted against the liquid or gas disposed in damping cell 3. In this manner an absorption or cushioning of pressure peaks within inlet conduit 6 remote from rotor 14 will take place effectively. On the other hand, when the pressure peak subsides in conduit 6, the compressed fluid in damping cell 3 is once more expanded into conduit 6 whereby fluctuations in pressure and volume in this portion of the machine are leveled off.

The suction fluid passes from conduit 6 through rotor conduit 17 and enters the intervane space 35 located between rotor 14, slide ring 16, and the rotor side portions 9 and 19. Where a pressure peak is produced in intervane space 35, it is immediately propagated to pistons 12 and 13 which are avially movable in rotor 14 at side portion 9. By the rearward movement of pistons 12 and 13 in consequence of the increased pressure in intervane space 35, the volume of the particular intervane space 35 is increased. Springs 10 and 11 are accordingly compressed by the movement of pistons 12 and 13 whereupon after the pressure peak has subsided in intervane space 35, springs 10 and 11 once more urge pistons 12 and 13 in a direction toward the particular intervane spaces 35 so as to reduce the volume of the intervane spaces. Due to the resilient force of springs 10 and 11, the fluctuation of the pressure and volume in the particular intervane spaces 35 is leveled off. Compressive springs 10 and 11 are constructed in this embodiment as compressive coil springs, although any suitable resiliently urging means may be provided, such as, for example, plate springs and the like.

At the end of shaft 5, adjacent rotor 14, piston 20, which is loaded by spring 22, is in flow communication with the portion of inlet conduit 6 adjacent rotor conduit 17. In the same way, piston 21, loaded by spring 23, is in flow communication with outlet conduit 7, adjacent rotor 14, at rotor conduit 18. These pistons 20 and 21 are preferably located as close as possible to the particular intervane spaces 35. Thus, when a peak pressure and volume is produced in conduit 6 or 7, the particular piston 20 or 21 will move in rearward direction away from the conduit so as to increase the volume thereof and accordingly overcome the peak fluctuation of pressure and volume. When the peak has subsided, the appropriate spring 22 or 23 will resiliently urge the piston toward the conduit so as to decerase the volume. In this manner, the leveling off of pressure and volume fluctuations in conduits 6 and 7 will be attained.

The fluid will reach pistons 26 and 27 from the corresponding intervane spaces 35 by means of conduits 24 and 25 respectively. Immediately upon reaching the peak volume and pressure in the appropriate intervane space 35, the piston 26 or 27 will move rearwardly from the intervane space whereby the cell volume will be correspondingly increased. Springs 28 and 29, provided for pistons 26 and 27 respectively, are constructed as plate springs in this embodiment although other suitable spring means may be employed. The rear end of the cylinders housing pistons 26 and 27 are in flow communication with the casing cell of the machine by means of conduits 30 and 31. In this way, a counter-acting compressive force will not arise behind pistons 26 and 27. Where, of course, such compressive action behind pistons 26 and 27 is desired, such condits 30 and 31 may be omitted. Immediately, upon the volume and pressure peak subsiding, springs 28 and 29 resiliently urge the corresponding pistons 26 and 27 once again in a direction toward the appropriate intervane space 35. In this manner, the volume of the intervane space is accordingly decreased whereby an effective leveling off of the fluctuations of pressure and volume in the intervane spaces 35 is achieved.

In the same way as mentioned above, it is preferred to arrange the damping pistons 26 and 27 as near to the corresponding intervane space 35 as is possible so as to minimize the distance between the intervane spaces and the particular piston. Thus, in FIG. 1, pistons 26 and 27, are actually provided in the main shaft of the machine which rotates, together with rotor 14.

In the embodiment shown in FIGS. 1 and 2, a pressure oil pump, having six intervane spaces is provided which acts to produce the fluctuation of delivery volume and pressure so that oil may be drawn in through conduit 6 during the increasing volume phase of each intervane space 35 during the rotation and thereafter the oil forced out of the intervane space through conduit 18 and outlet conduit 7. It will be appreciated that eccentric slide ring 16 remains stationary while rotor 14, including side portions 9 and 19, rotates by means of shaft 33. These elements which rotate are seated in bearings 8 and 32 for reducing friction to a minimum. Of course, in this embodiment, six intervane spaces 35 are provided, each having a damping piston or valve flow communicating therewith in side portion 9, as well as a damping piston or valve flow communicating therewith by means of a particular conduit 24 or 25. The various damping means, in accordance with the invention, operate during the increase and decrease in volume of the appropriate intervane spaces 35 so as to offset peak fluctuations in volume and pressure consequently produced. By providing damping pistons or valves in the control shaft adjacent the intervane spaces as well as chambers in the control shaft remote from the rotor in addition to the foregoing damping pistons or valves, the complete avoidance of noise, vibration, peak fluctuations, and the like will be attained both during the suction of oil into the intervane spaces as they increase in volume, and during

the pumping of the oil from the intervane spaces as they decrease in volume.

With reference to FIG. 3, the curves 46 indicate the change in delivery volume of the individual intervane spaces during the rotation of the rotor. The curve 45, on the other hand, is the sum of the individual delivery volumes wherein the curve 43, shows the higher portion of the waves and the curve 44 shows the lower portion of the waves of the fluctuating original delivery volume. The line 40 represents the minimal value of the delivery volume while the line 42 represents the maximum value thereof. In this connection, line 41 represents the typical or average delivery volume which is free from fluctuations, this curve of average delivery volume being substantially attained in accordance with the invention.

In connection with the operation of the oil pump shown in FIGURES 1 and 2, the peak 43 of the delivery volume in FIG. 3 produces a compression of the damping cushion or damping means of the invention so as to increase the volume and thereby decrease the pressure. Moreover, the damping cushion or damping means of the invention acts to reduce the depression 44 of the delivery volume caused by the subsiding of the peak fluctuation by reducing the fluid volume fluctuation space of the machine in the aforedescribed manner.

In accordance with the present invention, the damping cushion or damping means is preferably arranged as close as possible to the particular intervane space 35 and the inlet and outlet conduits 6 and 7 of the rotary vane machine. Thus, these damping elements are arranged, for example, in the rotor, the rotor side portions, the casing portion, and the slide ring portion, the drive shaft or transmitting shaft, the control shaft portion, and the like. It will be appreciated that depending upon the construction, the damping cushion or pressure expansion chamber communicating with inlet conduit 6 and outlet conduit 7 as the case may be, may be filled completely or partially with a gaseous and/or liquid medium.

FIGURE 4 illustrates one constructional embodiment of a damping valve which may be used in place of resiliently urged pistons as damping means in accordance with the invention. The damping valve of FIGURE 4 is provided so as to accommodate exceedingly high frequencies of fluctuation. In this connection, with respect to the embodiment of FIG. 1, a comparatively high variation of the volume of the intervane spaces is encountered and the pistons 12, 13, 20, 21, 26, and 27 must be able to move at comparatively high speeds along comparatively large distances, in order to effectively level off the peak fluctuations. These damping pistons must have a stroke volume which corresponds sufficiently to the range of fluctuation of the delivery volume. Necessarily with higher fluctuating frequencies and higher fluctuating amplitudes, a correspondingly higher acceleration of the piston in each instance is required. Where the damping piston and damping spring therefor have a larger mass and the frequency is constant, only a certain maximum amplitude is possible.

On the other hand, in accordance with the construction shown in FIGURE 4, much higher frequencies can be accommodated than with the damping piston arrangement shown in FIG. 1 due to the shortening of the amplitude over that in accordance with the construction of the valve of FIG. 4.

Thus, in FIG. 4, a short amplitude valve is provided in the control shaft or rotor 60 or other portion of the machine as desired, so as to effectively carry out the damping of the peak fluctuations. Valve seat 66 is retained by fixed ring 67 in the cylinder wall defined by part 60 while bore member 62 provided with bore openings 61 is maintained by retaining ring 63, fixedly mounted on part 60. The piston or valve member 65 is provided with a conical head capable of sealing engagement with valve seat 66 correspondingly shaped with a conical surface for this purpose. Valve 65 is urged into abutting engagement with valve seat 66 by means of coil spring 64, seated against bore member 62. Valve 65 will be displaced in rearward direction upon the application of a pre-determined pressure upon valve 65 so as to overcome the spring force of coil spring 64.

In operation, the valve or piston 65 is displaced very slightly from the valve seat 66 defining a conical slot therebetween of narrow dimensions. This conical slot clearance is produced when the peak fluctuation of the pressure is attained such that a portion of the pressure liquid escapes through this clearance. The amplitude of displacement of valve or piston 65 with respect to valve seat 66 may suitably be less than 1 mm. Upon the discharge of pressure liquid at the peak fluctuation, the clearance is closed once more due to the counter force of spring 64. The fluid which escapes may pass through the bore openings 61 of bore member 62; however, the valve or piston construction of FIG. 4 has the drawback that a certain portion of the pressure liquid is lost during peak fluctuations by escaping through the valve. The output of the rotary pump is, therefore, incomplete and a loss of efficiency to a certain extent will result.

To overcome this loss of efficiency, a further preferred embodiment may be used in accordance with the invention as illustrated in FIGURE 5. In this case a cylindrical piston 74 is employed as the valve. Piston 74 is provided with a recess or slit in its forward surface of predetermined dimensions. This valve embodiment may be effectively provided in the control shaft, for instance, of the machine. Fixed retaining ring 70 provided adjacent the delivery passage 80 of the control shaft 81 retains valve seat 71 in position. In the same way fixed retaining ring 76 maintains bore member 78 having bore openings 77 defined therein in fixed position adjacent the suction passage 82 of the machine. Valve seat 71 is axially displaceable against the action of coil spring 73 while piston 74 concentric therewith is axially displaceable against the action of springs 75. Valve piston 74 is provided with a stop flange 79. Thus, while valve seat 71 and piston 74 may move in axial direction against the action of their respective springs 73 and 75, valve seat 71 cannot move axially past valve piston 74 due to the stop flange 79 carried by piston 74. By providing the valve in the web portion 81 of the control shaft which separates the suction passage 82 from the delivery passage 80, a particularly suitable arrangement is effected.

According to the construction shown in FIG. 5, smaller fluctuations are counter-balanced by the vibration of piston 74. Only the higher fluctuating peak pressures will force the piston 74 rearwardly in axial direction beyond the displacement of valve seat 71, such that the slit 72 will flow communicate the delivery passage 80 with the suction passage 82. In this regard, a portion of the pressure liquid will only escape through the slit 72 at the very end of the peak fluctuation due to the relative counter-force exerted by springs 73 and 75 against their respective displacement members 71 and 74.

The construction of FIGURE 5 is essentially a combination of the damping due to the motion of the piston being displaced in its cylinder and the damping due to the opening of the valve. Consequently, the valve of FIGURE 5 is more advantageous in efficiency than the valve of FIGURE 4. Moreover, the valve of FIGURE 5 includes, in addition to the valve piston 74, a cylinder or valve seat 71 which is axially displaceable therewith. The cylinder 71 actually works in the manner of a pre-compressed piston depending upon the tension of spring 73.

In operation, where the average pressure in the delivery passage varies, for example, where the pressure becomes high, such pressure exerts against the pre-compressed cylinder 71 and in turn against the spring 73. Moreover, this pressure urges valve piston 74 and in turn its spring 75 in the same way. Cylinder or valve seat 71 during

each fluctuation will, therefore, occupy the piston depending upon the increase in average pressure and the counter force of spring 73.

Consequently, valve piston 74 will not be opened by the force of the pressure liquid even where the average pressure varies but will only be opened upon the pressure liquid reaching a fluctuating peak which is higher by a certain pressure difference than that of the average amplitude of cylinder 71.

Specifically, the amplitude and pressure differences are determined by the mass and the force of both springs 73 and 75 and particularly by the size of the pre-stressed piston 71 as well as the mass of cylinder 71 and valve piston 74. Upon escape of the pressure liquid, the same will be discharged through bore openings 77.

The particular embodiment shown in FIGURE 5 is of especial advantage when provided between the delivery passage and suction passage of the control shaft adjacent the rotor whereby the damping means may act as a valve for said shaft. In this case, the valve preferably occupies a position immediately next to the fluctuation producing source whereby the same may respond to the fluctuations of volume and pressure without time lag. In this manner, the damping device of the invention will achieve its maximum effect.

In FIGURE 6, an exchangeable or replaceable valve unit is provided having a casing 91 for the damping spring piston arrangement. The unit may be completely assembled and inserted in the proper location by means of threads 87 provided along its periphery, the shoulder 88 affording a sealing surface, together with the bore opening to which the unit is attached. Cylinder valve seat 92 is actually displaceable within casing 91 and is placed under the tension of spring 95 whereby retaining ring 85 limits the forward axial movement of valve seat 92. Retaining ring 99 maintains bore member 89 having bore openings 98 defined therein, in fixed relation within casing 91. The rear end of coil spring 95 abuts bore member 89 while the forward end of spring 95 urges cylinder valve seat 92 in axially forward direction against retaining ring 85. Damping piston 94 is axially displaceable within valve seat 92 and is urged in forward axial direction by means of coil spring 96, seated against bore member 89. Piston 94 is developed with hollow chambers 93 and 97 to lighten the mass of the piston. Retaining ring 90 maintains bore member 86 having openings 84 defined therein in fixed position with respect to valve seat 92. Bore member 86 prevents the axial displacement in rearward direction of valve seat 92 beyond the forward end of piston 94. It will be appreciated that valve seat 92, springs 95 and 96, and bore openings 98 in bore member or perforated disc plate 89 all function in the same manner as described with respect to valve seat 71, springs 73 and 75, and bore openings 77 in bore member or perforated disc plate 78 of FIGURE 5. While the damping piston 94 is shown without slit means, slit means such as those of the type shown by slit 72 in FIGURE 5 may also be used.

An important advantage of the piston 94 is the fact that its mass is markedly reduced by the provision for hollow bores 93 and 97 therethrough. Of course, piston 94 may be advantageously formed in such a manner that after bores 93 and 97 have been made, welded plugs may be added to the ends of the piston to prevent the pressure liquid from passing through bores 93 and 97. Due to the reduced mass of piston 94 in this manner, the frequency and amplitude of the damping piston may be raised to the maximum limit. Accordingly, the device shown in FIGURE 6 has the further advantage that it may be stored within a casing or cylindrical tube and readily inserted into the rotary machine.

It will be appreciated by the artisan that the embodiments of the present invention which have been specifically illustrated are merely exemplary constructions and that the invention is not to be limited thereto. In creating a practical design for damping means in accordance with the present invention, the artisan must be well acquainted with the kinematic and hydrodynamic values of the rotor of the particular machine to be damped. The damping action, it will be appreciated, is only effective where the gas, air, or liquid medium used as a damping cushion in a damping chamber, or the spring, piston or mass of valve employed are adapted specifically to the fluctuation of the delivery volume and pressure of the particular machine and where these damping means are located close enough to the fluctuation generating source to be effective. Naturally, the shorter the distance between the damping cushion means and the appropriate fluid pressure fluctuation space, the quicker will be the response of the damping cushion means to peak fluctuations.

What is claimed is:

1. Rotary vane device comprising in combination a casing having an inner cylindrical wall, a cam ring disposed in said wall and having a radially inner sliding surface, a rotor disposed in said ring, said rotor being eccentrically rotatably mounted with relation to the inner surface of said ring, rotor side walls connected to the axial ends of said rotor for rotation therewith and having medial wall surfaces adjacent the rotor ends, which extend radially outwardly past the ring inner surface, slots formed in said rotor, vanes disposed in said slots for reciprocation therewith, to define between the vanes, the ring inner surface, the outer radial surface of said rotor and the medial surfaces of said rotor side walls, intervane spaces, control means having an inlet conduit for passing fluid to said intervane spaces and an outlet conduit for passing fluid from said intervane spaces, and damping means provided in at least one rotor side wall in immediate and direct flow communication with said intervane spaces, said damping means being pressure responsive to the fluid pressure fluctuation in said intervane spaces caused by the fluid volume fluctuation during operation of the device for damping the fluid pressure by adjusting the volume during peak fluctuations.

2. Device according to claim 1 wherein a corresponding damping means is provided in said rotor side wall in flow communication with each of said intervane spaces.

3. Rotary vane device comprising in combination a casing having an inner cylindrical wall, a cam ring disposed in said wall and having a radially inner sliding surface, a rotor disposed in said ring, said rotor being eccentrically rotatably mounted with relation to the inner surface of said ring, rotor side walls connected to the axial ends of said rotor for rotation therewith and having medial wall surfaces adjacent the rotor ends, which extend radially outwardly past the ring inner surface, slots formed in said rotor, vanes disposed in said slots for reciprocation therewith, to define between the vanes, the ring inner surface, the outer radial surface of said rotor and the medial surfaces of said rotor side walls, intervane spaces, control means having an inlet conduit for passing fluid to said intervane spaces and an outlet conduit for passing fluid from said intervane spaces, a rotatable shaft having an enlarged shaft end portion, said shaft being connected at said shaft end portion to one of said rotor side walls for rotation therewith and with said rotor, axial passages defined in said one rotor side wall, and damping means provided in said shaft end portion in immediate and direct flow communication with said intervane spaces through said shaft end portion and said rotor side wall axial passages, said damping means being pressure responsive to the fluid pressure fluctuation in said intervane spaces caused by the fluid volume fluctuation during operation of the device for damping the fluid pressure by adjusting the volume during peak fluctuations.

4. Device according to claim 3 wherein a corresponding damping means is provided in said shaft end portion in flow communication with each of said intervane spaces.

5. Rotary vane device comprising in combination a casing having an inner cylindrical wall, a cam ring dis-

posed in said wall and having a radially inner sliding surface, a rotor disposed in said ring, said rotor being eccentrically rotatably mounted with relation to the inner surface of said ring, slots formed in said rotor, vanes disposed in said slots for reciprocation therewith, to define intervane spaces between the ring inner surface and the outer radial surface of said rotor, control means having an inlet conduit for passing fluid to said intervane spaces and an outlet conduit for passing fluid from said intervane spaces, said control means extending axially into the center portion of said rotor, inlet and outlet radial passages defined in said rotor, said inlet and outlet conduits communicating with said intervane spaces through said inlet and outlet radial passages in said rotor, and damping means provided in said control means radially inwardly of and axially adjacent said rotor radial passages, one in immediate and direct separate flow communication with said inlet conduit and one with said outlet conduit, said damping means being pressure responsive to the fluid pressure fluctuation in said intervane spaces caused by the fluid volume fluctuation during operation of the device for damping the fluid pressure by adjusting the volume during peak fluctuations.

6. Device according to claim 5 wherein further damping means are provided in the control means axially remote from the rotor, one in immediate and direct separate flow communication with the inlet conduit and one with said outlet conduit.

7. Rotary vane device comprising in combination a casing having an inner cylindrical wall, a cam ring disposed in said wall and having a radially inner sliding surface, a rotor disposed in said ring, said rotor being eccentrically rotatably mounted with relation to the inner surface of said ring, rotor side walls connected to the axial ends of said rotor for rotation therewith and having medial wall surfaces adjacent the rotor ends, which extend radially outwardly past the ring inner surface, slots formed in said rotor, vanes disposed in said slots for reciprocation therewith, to define between the vanes, the ring inner surface, the outer radial surface of said rotor and the medial surfaces of said rotor side walls, intervane spaces, control means having an inlet conduit for passing fluid to said intervane spaces and an outlet conduit for passing fluid from said intervane spaces, said control means extending axially into the center portion of said rotor, inlet and outlet radial passages defined in said rotor, said inlet and outlet conduits communicating with said intervane spaces through said inlet and outlet radial passages in said rotor, a rotatable shaft having an enlarged shaft end portion, said shaft being connected at said shaft end portion to one of said rotor side walls for rotation therewith and with said rotor, axial passages defined in said one rotor side wall, first damping means provided in at least one of the rotor side walls in immediate and direct flow communication with said intervane spaces, second damping means provided in said shaft end portion in immediate and direct flow communication with said intervane spaces through said shaft end portion and said rotor side wall axial passages, third damping means provided in said control means radially inwardly of an axially adjacent said rotor radial passages, one said third damping means being in immediate and direct separate flow communication with said inlet conduit and one with said outlet conduit, and fourth damping means provided in the control means axially remote from the rotor, one said fourth damping means in immediate and direct separate flow communication with the inlet conduit and one with the outlet conduit, all of said damping means being pressure responsive to the fluid pressure fluctuation in said intervane spaces caused by the fluid volume fluctuation during operation of the device for damping the fluid pressure by adjusting the volume during peak fluctuations.

8. Device according to claim 7 wherein said first, second and third damping means are defined by coacting fluid pressure responsive piston means and cylinder means disposed in said rotor side wall, shaft end portion, and control means, respectively, said piston means being slidable within said cylinder means and said piston means and cylinder means being in immediate and direct separate flow communication at their forward portions with said intervane spaces, passages and conduits, respectively, said piston means being provided with resilient means normally resiliently urged said piston means to occupy said cylinder means, and said fourth damping means is defined by fluid expansion chamber means, one in immediate and direct separate flow communication with said inlet conduit and one with said outlet conduit.

9. Device according to claim 8 wherein an end portion of said control means within said rotor is adjacent said shaft end portion at said one rotor side wall, common conduit means being provided which extend from said control means end portion through said one rotor side wall to said shaft end portion, the rearward portions of said piston means and cylinder means of at least a portion of said second damping means being in flow communication with the rearward portions of said piston means and cylinder means of said third damping means through said common conduit means.

10. Device according to claim 9 wherein the other of said rotor side walls is provided with axial bore channel means extending laterally therethrough to the casing of the device, the rearward portions of said piston means and cylinder means of said first damping means being in flow communication with the casing remote from the rotor through said axial bore channel means.

11. Device according to claim 7 wherein said first, second and third damping means are defined by closure valve means in immediate and direct separate flow communication with said intervane spaces, passages and conduits respectively, said valve means being provided with resilient means normally resiliently urging said valve means into closed position.

12. Device according to claim 11 wherein said valve means includes a flow conduit having a valve seat and a spring loaded closure member normally urged into closing abutting engagement with said valve seat, said closure member being provided with a conical head and said seat being provided with a corresponding conical surface for sealing abutment with said conical head along their common abutting portions.

13. Device according to claim 11 wherein said valve means includes a flow conduit having a valve seat and a spring loaded closure member normally urged into closing abutting engagement with said valve seat, the valve means of at least one of said first, second and third damping means having a cylinder valve seat and a coacting piston closure member slidably extending thereinto, both said valve seat and closure member being provided with spring means and being limited by axially displaceable in said flow conduit with respect to each other and with respect to said flow conduit and normally resiliently urged by said spring means in forward axial direction into closed position with respect to one another, said closure member being provided at its forward end with rearwardly and radially outwardly extending groove means, said groove means terminating prior to the rearward side of said valve seat and being normally sealed from flow communicating one side of said valve seat with the other when said closure member and said valve seat are urged into forward closed position, and upon axial displacement of said closure member in the rearward axial direction for a greater predetermined distance than said valve seat, said groove means extending rearwardly past the rearward side of said valve seat and flow communicating one side of said valve seat with the other.

14. Device according to claim 13 wherein said closure member is resiliently urged by spring means under less spring force than that of the spring means under which said valve seat is urged, said closure member being provided with stop means for limiting the axial distance said

closure member extends into said valve seat in said forward direction.

15. Device according to claim 11 wherein said valve means includes a flow conduit having a valve seat and a spring loaded closure member normally urged into closing abutting engagement with said valve seat, the valve means of at least one of said first, second and third damping means having a cylinder valve seat and a coacting piston closure member having a forward end slidably extending thereinto, both said valve seat and closure member being provided with spring means and being limitedly axially displaceable in said flow conduit with respect to each other and with respect to said flow conduit and normally resiliently urged by said spring means in forward axial direction into closed position with respect to one another, said closure member being hollow and sealed from passage of fluid therethrough, the forward end of said closure member normally preventing flow communication from one side of said valve seat to the other when said forward end of said closure member extends into said valve seat in forward closed position, and upon axial displacement of said closure member in the rearward axial direction for a greater predetermined distance than said valve seat, the forward end of said closure member slidably extending from said valve seat for flow communicating one side of said valve seat with the other.

16. Improvement according to claim 15 wherein said closure member is resiliently urged by spring means under less spring force than that of the spring means under which said valve seat is urged, said valve seat being provided with stop means for limiting the axial distance said closure member extends thereinto in said forward axial direction.

17. Improvement according to claim 16 wherein said valve seat and said closure member are operably mounted in a tubular member as an exchangeable self-contained valve unit, said tubular member having means for releasably securing said unit in said machine as said valve means.

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UNITED STATES PATENT OFFICE

CERTIFICATE OF CORRECTION

Patent No. 3,145,662 August 25, 1964

Karl Eickmann

It is hereby certified that error appears in the above numbered patent requiring correction and that the said Letters Patent should read as corrected below.

Column 2, line 54, for "spaced" read -- space --; column 5, line 58, for "avially" read -- axially --; column 6, line 43, for "spaces" read -- space --; column 11, line 60, for "an" read -- and --; column 12, line 8, for "urged" read -- urging --; line 54, for "limited by" read -- limitedly --.

Signed and sealed this 9th day of February 1965.

(SEAL)
Attest:

ERNEST W. SWIDER
Attesting Officer

EDWARD J. BRENNER
Commissioner of Patents