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[54] FLOW REGULATING VALVE

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[51] **Int. Cl.**.....**G05d 7/00**

[58] **Field of Search** 137/501, 504, 503; 138/44, 138/42

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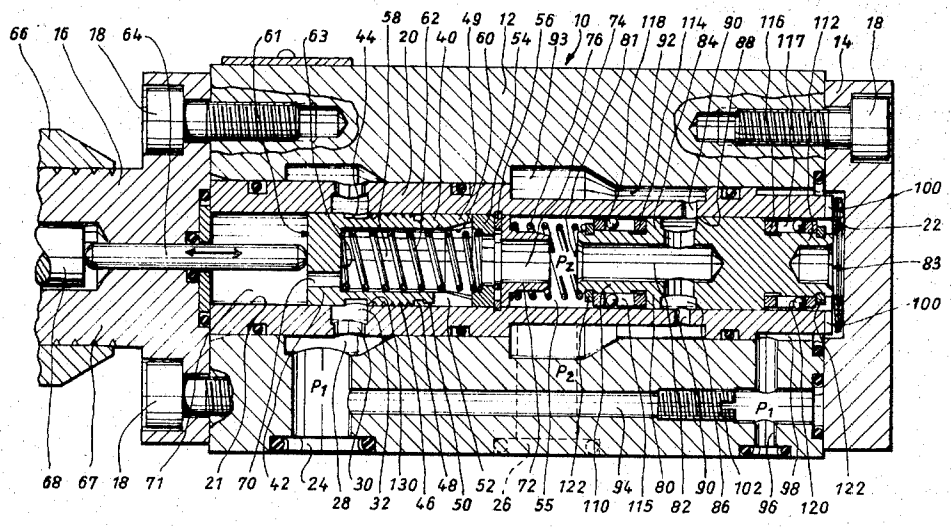
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[57] **ABSTRACT**

In a chamber of a housing provided with at least one inlet and at least one outlet is accommodated a sleeve having a longitudinally extending passage as well as first and second flow ports communicating with the passage and with the inlet and outlet respectively. A throttling piston is located in the passage displaceable longitudinally thereof with reference to the first ports. A substantially fluid-flow throttling passage is provided in either the outer circumferential surface of the throttling piston or in the inner circumferential surface bounding the passage, and in any case extends longitudinally of the passage and communicates with the respective ports. A control piston is located in the passage downstream of and axially adjacent to the throttling piston for maintaining constant fluid flow through the passage.

22 Claims, 4 Drawing Figures



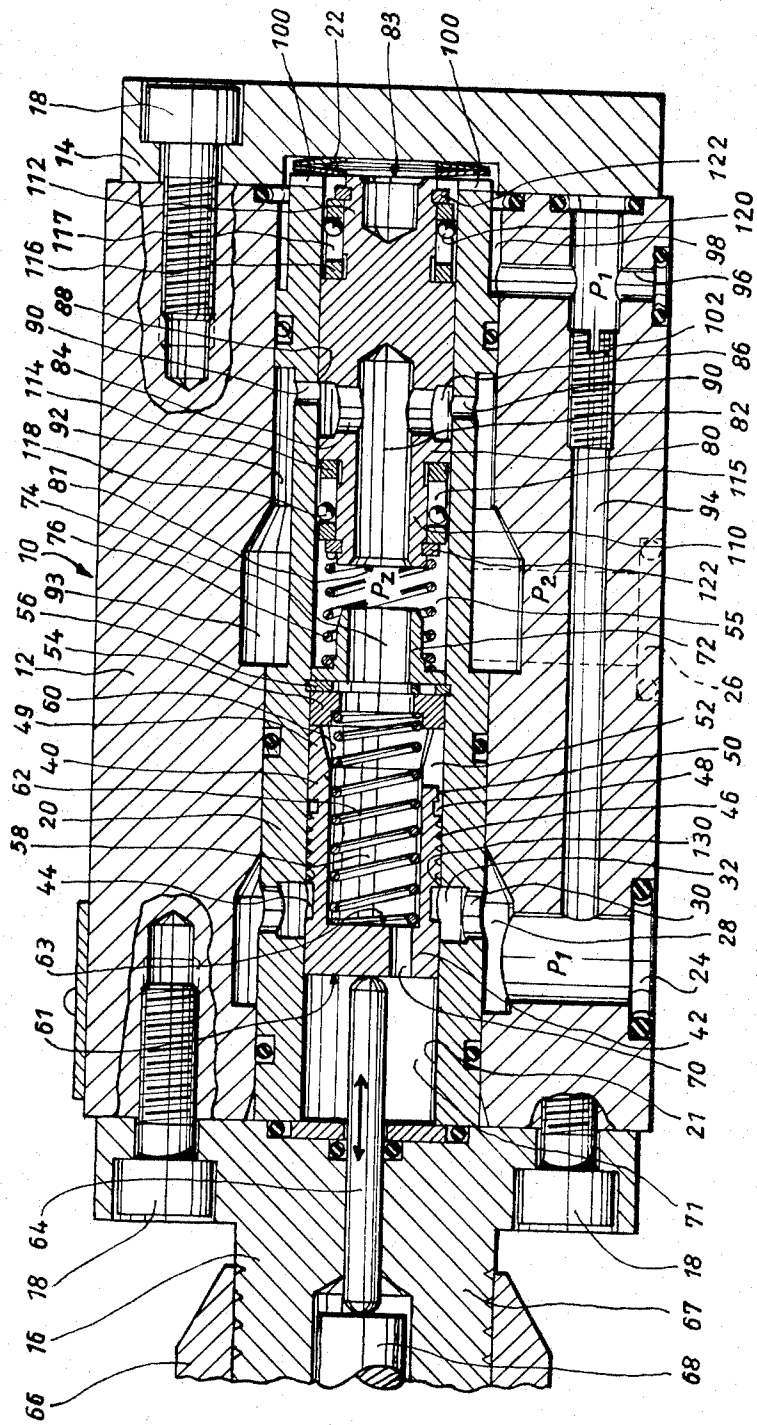


Fig. 1

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Fig. 2

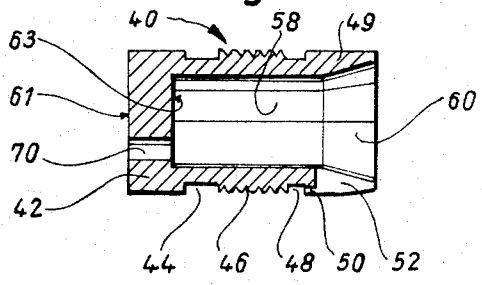


Fig. 3

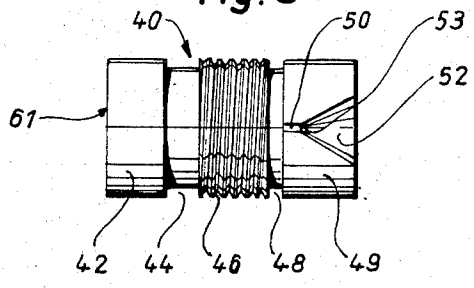
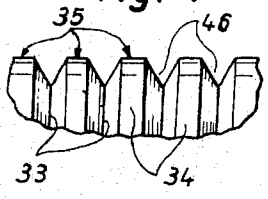


Fig. 4



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FLOW REGULATING VALVE

BACKGROUND OF THE INVENTION

The present invention relates generally to a valve, and more particularly to a regulating valve. Still more particularly the invention relates to a flow regulating valve for liquids and gases, and especially for liquids and gases which flow in small quantities.

Such regulating valves are basically already well known. Generally speaking they are provided with a throttling aperture the cross-sectional area of which—through which the fluid flows—can be varied to thereby vary the quantity of fluid which flows through per unit of time. The known constructions are all more or less serviceable, but it has been found that if the quantity of fluid passing through the valve is lower than approximately 30 cm^3/min, the known valves of this type no longer assure reliable and certain regulating of the flow quantity. The reason for this is that throttling apertures having a diameter of approximately 0.4 mm and less have a tendency to become clogged after a relatively brief period of time, even if the fluid being passes through the valve is well filtered. It is not yet entirely clear, in fact, why this should be the case despite proper precautions taken to assure good filtration, but the fact remains that this is what happens. For this reason the known constructions are not satisfactory where small quantities of flowing fluid are to be regulated.

SUMMARY OF THE INVENTION

It is, accordingly, an object of the present invention to overcome the aforementioned disadvantages.

MOre particularly it is an object of the present invention to provide an improved fluid flow regulating valve which permits reliable and accurate regulation of even small and very small quantities of fluid flowing through the valve.

An additional object of the invention is to provide such a valve which is comparatively simple in its construction.

A concomitant object of the invention is to provide such a valve which is not subject to clogging and similar malfunctions during operation.

In pursuance of the above objects, and others which will become apparent hereafter, one feature of the invention resides, briefly stated, in a flow regulating valve for fluids which comprises a housing having a chamber provided with at least one inlet and at least one outlet. A sleeve is accommodated in the chamber and has a longitudinally extending passage bounded by an inner peripheral surface and first and second flow ports communicating with the passage and with the inlet and outlet, respectively, and which flow ports are at least in part bounded by flow-controlling edge portions. A throttling piston is accommodated in the passage displaceable longitudinally thereof with reference to the first ports and to the edges bounding the same and has an outer peripheral surface. A substantially helical fluid-flow throttling passage is provided in and extends longitudinally of one of the surfaces and a control piston is accommodated in the passage downstream of and axially adjacent to the throttling piston for maintaining the flow of fluid through the throttling passage constant.

The provision of the throttling passage in helically convoluted form, or in a form approaching helical convolutions such as an essentially spiral configuration, makes it possible to regulate and control reliably and constantly quantities of flowing fluid as low as 0.1 cm^3/min and even lower. Such regulation is independent of the pressure prevailing at the inlet and the outlet of the valve. The tolerance of the through-put quantity, that is the quantity of fluid which flows per unit of time through the valve, is approximately ±1 percent of the predetermined through-put value, that is the value which has been previously set with the valve.

The valve according to the present invention has a wide range of applicability. It is particularly suitable for controlling the flow of hydraulic oils, especially those having a viscosity range of approximately 1 to approximately 3 cSt (centistoke) and for pressures up to approximately 300–350 kp (kilopound) per cm^2. It is emphasized, however, that the valve is well suited for regulating the flow of other media, also, that is for liquids as well as for gases. The valve according to the present invention may be a two-way valve or a three-way valve and may be actuated—for displacing of the throttling piston in a sense varying the through-put quantity—in mechanical, hydraulic, pneumatic or electric manner.

While it has been indicated before that the throttling passage may be provided either in the outer peripheral surface of the throttling piston or in the inner peripheral surface bounding the passage of the sleeve in which the piston is accommodated, it is particularly advantageous to provide the throttling passage in the outer peripheral surface of the throttling piston itself. In this case it is preferred to provide an annular groove at the upstream end and the downstream end of the passage, also in the outer peripheral surface of the throttling piston. The throttling passage may have—but need not have—a cross-sectional area of approximately 0.2 mm^2 and a length of 15 cm. Naturally this is only by way of example and can be varied in dependence upon particular requirements. If the cross-sectional area is approximately 0.2 mm^2, this corresponds to a conventional throttling aperture of the prior art having a diameter of approximately 0.5 mm. It has been found that if in the prior art throttling apertures of this dimension could be used for controlling the small fluid throughputs which are involved, they would not become clogged if the fluid is previously filtered, and accordingly the helical throttling passage according to the present invention also does not become clogged, being the equivalent of a conventional throttling passage of the aforementioned diameter, but permitting the control of small fluid-flow quantities which a throttling passage of the conventional type and having the aforementioned diameter would not permit.

It is advantageous that the pitch between the convolutions of the throttling passage be greater than is the width of the throttling passage in the region of the outer circumferential or peripheral surface of the throttling piston. If so constructed, the ribs located between the individual convolutions have at the outer peripheral surface of the throttling piston cylindrical surfaces which are of course annular surface portions of the outer peripheral surfaces of the piston. This, in turn, makes possible a precise guidance of the throttling

piston in the sleeve and prevents the liquid streaming through the throttling passage from skipping axially of the throttling passage by leaking outside the passage between the outer circumferential surface of the piston and the inner circumferential surface of the passage in the sleeve.

It is advantageous to locate a separate throttling aperture downstream of the spiral throttle which is constituted by the provision of the aforementioned throttling passage. This separate or additional throttling passage is also provided on the throttling piston and by having the helical throttling passage and the additional throttling passage or throttling aperture in communication with one another, a combination is obtained in which fluid throughput quantities of approximately 0.1 to approximately 40 cm^3/min through the spiral throttle can be precisely and constantly regulated. These small quantities of fluid pass through the throttling aperture without being influenced thereby but quantities in excess of approximately 40 cm^3 per minute are regulated and controlled through the throttling aperture. If such quantities occur, that is if 40 cm^3/min are exceeded, they remain uninfluenced by the spiral throttle.

If a separate throttling aperture of the type discussed above is provided, then it is advantageous to configurate it in form of a triangular recess in the outer peripheral surface of the throttling piston, extending axially of the latter and having one corner which merges into and communicates with an axially extending groove connecting it with the outlet of the helically convoluted throttling passage. More specifically, this groove preferably communicates with the annular groove provided in the throttling piston at the downstream end of the helical throttling passage and the cross-sectional area of the axially extending groove should preferably be the same or larger than the cross-sectional area of the helical throttling passage itself.

It will be appreciated that precise regulating of the through-put in the region of the spiral throttle constituted by the helical throttling passage, can be obtained and reproduced because of the considerable length of the helical throttling passage which, as pointed out before, may be on the order of 15 cm. This is a feature which cannot be obtained with throttling apertures in the prior-art constructions because the small adjustments required in varying the cross-sectional area of such conventional throttling apertures make it impossible to provide precise adjustments for very small throughput quantities. Where throttling apertures are used, throughput quantities of 30 cm^3/min and greater are controllable reliably and with only small dependency on the viscosity of the fluid itself. In spiral throttles, such as the one constituted by the helical throttling passage provided in accordance with the present invention, throughputs in excess of 40 cm^3/min are more strongly dependent on the viscosity of the fluid. The low flow speed of the fluid, for instance oil, in conduits at throughput quantities up to 40 cm^3/min a complete temperature equalization between the oil and the ambient temperature takes place. Because the ambient temperature is almost constant in air-conditioned spaces, temperature variations such as they occur in the oil storage containers of hydraulic systems, have no influence on the constancy of regulation of throughput in spiral throttles. However, where throughput exceeds 40 cm^3/min, the greater flow speed of the fluid, such as oil, makes it impossible to obtain this temperature equalization without considerable difficulties and for this reason the throttling aperture provided according to the present invention and located downstream of the spiral throttle acts at flow speeds in excess of 40 cm^3/min, and provides control, this being clearly advantageous because its throughput quantity is independent of the viscosity of the flowing fluid. It is clear, therefore, that by constructing the valve as just pointed out, as a combination throttle, small and very small throughput quantities can be regulated through the spiral throttle component, and larger and large throughput quantities up to any specified desired size can be reliably and constantly regulated through the throttling aperture component, and in the latter case of course there is no danger that the throttling aperture could become clogged, just as there is no danger that the spiral or helical throttling passage could become clogged at the lower through-put quantities.

According to a further concept of the invention which is highly advantageous, it is advisable to mount and guide the regulating or control piston of the valve according to the present invention in antifriction bearings, e.g., ball bearings, because especially if precision-manufactured bearing balls are utilized, the control piston is centrally guided in the passage of the sleeve. This assures uniform radial play between the circumference of the control piston and the inner surface bounding the passage of the sleeve, and substantially reduces leakage losses as compared to the previously utilized control pistons. For instance, by resorting to this concept of the present invention the leakage of oil due to play between the control piston and the inner surface bounding the passage of the sleeve is reduced due to the concentric position of the control piston in the passage by a factor of approximately 2.5 as compared to what is experienced when the control piston is located in the passage in eccentric relationship. Thus, the control piston according to the present invention has a fluid leakage which is approximately 250 percent smaller than that of corresponding control pistons known from the art.

Furthermore, the very low friction obtained in this manner between the control piston and the sleeve in which it is accommodated, reduces or entirely eliminates disadvantageous influences upon the operational accuracy of the valve. In the prior art, where the control pistons were slidably mounted rather than by means of antifriction bearings, the functional accuracy of such valves was substantially limited and disadvantageously influenced, particularly at higher pressures, due to static and dynamic friction. Of course it will be appreciated that as a result of the reduced very low friction experienced by mounting the control piston according to the present invention on antifriction bearings, lower regulating forces are obtained with the result that the cross-sectional area of the throttling channel, the spiral throttle and the cross-section of the throttling aperture can be made larger than would otherwise be possible, thereby reducing the danger of clogging still further. In addition, the easy movement of the regulating or control piston guarantees a constancy of the once-set throughput quantity within a tolerance of approximately ±1 percent, an advantage which was

impossible to achieve in the known flow regulating valves. Finally, the provision of antifriction bearings for mounting the control piston in the passage of the sleeve has the additional advantage that the uniform radial play between regulating piston and sleeve over the entire periphery of the former, prevents the regulating piston from becoming located in a position which is skew with reference to the axis of the passage in the sleeve, an occurrence which is found frequently in the prior-art constructions where the control piston may be pressed against the inner surface bounding the sleeve passage by the fluid under pressure.

The novel features which are considered as characteristic for the invention are set forth in particular in the appended claims. The invention itself, however, both as to its construction and its method of operation, together with additional objects and advantages thereof, will be best understood from the following description of specific embodiments when read in connection with the accompanying drawing.

BRIEF DESCRIPTION OF THE DRAWING

FIG. 1 is an axial section through a currently preferred embodiment of my invention;

FIG. 2 is a sectional detail view, on an enlarged scale, showing a detail of the throttling piston of the valve in FIG. 1;

FIG. 3 is a side view of the throttling piston shown in FIG. 2; and

FIG. 4 is a fragmentary enlarged detail view of FIG. 3.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Discussing now the embodiment illustrated in FIGS. 1–4 it will be seen that reference numeral 1 identifies in toto a flow regulating valve according to my present invention. The valve comprises a housing 12 which in the illustrated embodiment has its interior chamber closed at opposite axial ends by end caps or end walls 14 and 16 which may be secured to the housing 12 by means of screws or in other suitable manner which is not a part of the present invention. It will also be understood that ordinarily suitable seals, which are not separately shown, such as O-rings or the like, are used to seal the juncture between the end walls and the housing 12 to prevent an escape of the fluid flowing through the valve.

Accommodated in the chamber of the housing 12 extending axiall thereof is a control sleeve or sleeve 20 whose length corresponds substantially to that of the housing 12. Again, suitable seals are not illustrated but will be provided between housing and sleeve and this is a conventional expedient and need not therefore be further discussed. One end of the sleeve 20 may abut against the end wall 16 whereas the other end of the sleeve may be subjected to a biasing force, for instance by means of a dished spring 22 which abuts against the end wall 14 so that the sleeve 20 is constantly biased against the end wall 16.

As FIG. 1 also shows, at least one inlet 24 is provided in the housing 12 for admission of the fluid—e.g., oil—whose flow is to be controlled. After entering the inlet 24 the fluid advances into an annular channel 28 provided in the housing 12 and passes through the valve to finally exit through the exit or outlet 26 of which at least one is provided (see the broken-line showing in FIG. 1). The pressure P_1 prevails in the inlet 24 whereas a different pressure p_2 prevails in the outlet 26 as will be discussed further below. Once having entered the annular channel 28, the fluid flows through one or more radially or substantially radial bores 30 in the sleeve 20 to the interior of the latter to enter an annular channel 32 provided for this purpose within the sleeve 20. FIG. 1 further shows that the sleeve 20 is also provided with an axial passage or bore 21 extending over its entire length and accommodating a throttling piston 40 in the left-hand side of FIG. 1.

The throttling piston 40 is shown in FIG. 1, and details are most readily visible from FIGS. 2, 3 and 4 where it will be seen that it has a cylindrical portion 2 which is followed in downstream direction—that is towards the right-hand side in FIG. 1—by an annular channel 44 provided in this instance by the outer peripheral surface of the piston 40. The annular channel 44 is located in a plane normal to the longitudinal axis of the piston 40 and is followed by and in communication with a helically convoluted throttling passage 46 provided also in the outer peripheral surface of the piston 40 and extending axially of the latter. At the downstream end of the passage 46 there is provided in communication with it an additional annular channel 48 which is also located in a plane normal to the longitudinal axis of the piston 40. Downstream of the channel or groove 48 there is provided a further cylindrical portion 49 of the throttling piston 40, as for instance shown in FIG. 3. It is to be mentioned parenthetically that the helical throttling passage 46 could also be provided in the inner circumferential surface bounding the passage in the sleeve 20, as already pointed out before.

The throttling piston 40 is axially displaceable in the passage of the sleeve 20 by means which will be discussed subsequently. It is provided in the peripheral surface of the cylindrical portion 49 with a throttling aperture 52 which is clearly visible in FIGS. 2 and 3 in particular, and which in the illustrated embodiment is in form of a triangularly configurated recess, a corner 53 of which (compare FIG. 3) communicates with the annular channel 48 via a groove 50 provided in the outer surface of the piston 40 and extending axially thereof.

Reference to FIG. 1 will show that the right-hand end (in FIG. 1) of the piston 40 abuts against an annular insert 54 which in turn is held against displacement towards the right in FIG. 1 by a retaining ring 56, for instance a circlip, which has a spring characteristic and in part projects into a recess provided for this purpose whereas in part it abuts against the insert 54. The piston 40 is hollow, being provided with an axial bore 58 which begins at an inner bottom wall 63 (see the left-hand side in FIG. 1) and which at the right-hand end portion (that is the downstream end portion) of the piston 40 merges into a conically diverging bore portion 60 (compare FIG. 2).

As already pointed out earlier, the piston 40 can be displaced axially in the passage of the sleeve 20. FIG. 1 shows that an expansion spring 62, for instance a helical spring, is accommodated between the insert 54 and the end face 63, permanently tending to displace the

piston 40 towards the left-hand side in FIG. 1 so that its outer end face 61 abuts against a pin 64. In a manner which is not illustrated in detail but which will be obvious to those skilled in the art the pin 64 can be axially displaced as indicated by the double-headed arrow associated with it, via a setting member 66 which is threaded onto a projection 67 of the end wall 16 and acts via an intermediate element 68 upon the pin 64. The "means" which has been diagrammatically illustrated in FIG. 1 indicates that displacement of the pin 64 can be effected in mechanical manner, in hydraulic manner, in pneumatic manner or in electric manner, all means of this type being well known to those skilled in the art and therefore requiring no specific discussion. The spring 62 urges the piston 40 against the inner end portion of the pin 64 so that turning of the member 66 and a resulting axial displacement of the pin 64 will necessarily result in a concomitant axial shifting of the throttling piston 40 within and relative to the sleeve 20. An axial bore provided in the portion 42 of the throttling piston 40 connects the bore 58 with the space 71 in the sleeve 20 to provide for a pressure equalization when movement is to be effected.

The cross-sectional configuration of the helical throttling passage 46 may be selected within a considerable range of possibilities. It may be generally polygonal, it may be quadratic, it may be rectangular, it may be at least substantially semi-circular or, as illustrated in the embodiment here described and as shown in FIG. 4 in particular, it may be triangular. In the illustrated embodiment the width of the passage 46 in the region of the outer periphery of the throttling piston 40 is smaller than the pitch between the convolutions of the helical throttling passage 46, so that ribs 34 remain between the individual turns 33 of the passage 46, which each have a cylindrical outer circumferential surface 35. It is preferable but not necessary that the cross-sectional configuration of the groove 50 be the same as that of the helical throttling passage 46, and preferably the cross-sectional area of the groove 50 should also be the same as that of the throttling passage 46. It is emphasized that the cross-sectional passage of the groove 50 may also be larger than that of the throttling passage 46, but that it should be no smaller for obvious reasons.

FIG. 1 shows that there is accommodated in the sleeve 20 in downstream direction from the circlip 56 a tubular portion 72 having a bore 74 and abutting with its upstream end against the circlip 56. Also located downstream of the piston 40 and in alignment therewith there is accommodated in the sleeve 20 a control or regulating piston 80 which is similarly axially displaceable within and with reference to the sleeve 20. FIG. 1 shows that it is provided with a blind bore 82 with which there communicates at least one radial bore 84 which in turn communicates with an annular channel 86 provided in the outer periphery of the control piston 80.

A control edge 88 on the annular channel 86 is located in the illustrated embodiment in a plane normal to the longitudinal axis of the control piston 80 but could also be located in a plane which is inclined to the longitudinal axis. It cooperates with radially extending control bores 90 provided in the sleeve 20 which in turn communicate with an annular channel 92 in the housing 12 an enlarged portion 93 of which channel communicates with the outlet 26. It is clearly shown in FIG. 1 that the control bores 90 in the sleeve 20 are offset with reference to one another in axial direction of the sleeve 20.

A spring 76, such as a helical spring or the like, is accommodated between the end 81 of the control piston 80 and the tubular portion 72, continuously urging the control piston 80 towards the right in FIG. 1, that is in downstream direction as seen with reference to the flow of fluid through the valve. The rear or downstream end 83 of the control piston 80 communicates with the inlet 24 and the incoming fluid therein via a conduit 94 and/or a direct connection 96, an annular space 98 and at least one radially oriented passage 100. This means, in other words, that the unthrottled inlet pressure P_1—which prevails in the inlet 24—also acts upon the rear or downstream end 83 of the control piston 80. If desired, a damping or regulating screw 102 may be positioned in the conduit 94 as shown.

It has already been pointed out earlier that according to a preferred embodiment of the invention the control piston 80 is mounted in the sleeve by means of antifriction bearings. FIG. 1 shows that it is provided for this purpose at its two axial ends with respective cylindrical projections 110 and 112. A bearing cage 114 surrounds the projection 110 and a similar cage 116 surrounds the projection 112 and both are provided with axially extending slots 115 and 117, respectively, in which bearing balls 118 and 120 are accommodated, respectively. Preferably but not necessarily the bearing balls 118 and 120 will be precision-manufactured steel balls which roll directly upon the cylindrical outer surfaces of the projections 110 and 112 on the one hand, and on the cylindrical inner surface bounding the passage of the sleeve 20, thereby assuring an extraordinarily precise journalling and guidance of the control piston 80 in the sleeve 20. This eliminates all danger that the regulating or control piston 80 could become inclined or skew with reference to the axis of the passage of the sleeve 20, causing difficulties in its axial displacement and in the execution of its control functions. Retaining rings 122, such as circlips or the like, retain the cages 114 and 116 against axial displacement.

In operation of my novel regulating valve as illustrated in the aforedescribed exemplary embodiment, liquid—e.g., oil—enters the inlet 24, being for instance supplied by a non-illustrated pump. From the inlet 24 it passes into the annular channel 28, from the there through the bores 30 and the annular channel 32 into the annular channel 44 of the piston 40. At this time the oil still has the inlet pressure P_1 and FIG. 1 shows the throttling piston 40 in the position in which the throughput per unit of time is the smallest. In other words, from this position the piston 40 can be displaced to increase the throughput per unit of time.

The oil passes from the annular channel 44 into the first turn 33 of the helically convoluted throttling passage 46, and then proceeds to traverse the entire length of the throttling passage 46 to exit into the annular channel 48 from where it passes via the groove 50 and the throttling aperture 52 into the space 55. In this space there exists an intermediate pressure P_Z.

In dependence upon turning of the element 66 and a consequent axial displacement of the pin 64, the throt-

tling piston 4 can be displaced towards the left in FIG. 1 from the illustrated position to thereby increase the throughput per unit of time to a desired value. A control edge 130 provided on the sleeve 20 and located in a plane normal to the longitudinal axis of the sleeve, controls the flow of liquid or gas into the throttling passage 46. It will be appreciated that depending upon the axial position of the throttling piston 40—as selected by axial displacement of the pin 64—a portion of the throttling passage 46 of greater or lesser length will be covered by the inner wall of the sleeve 20 or will be uncovered thereby, depending upon how far the piston 40 is moved towards the left in FIG. 1 so that a requisite portion of the throttling passage 46 has moved past the control edge 130. Evidently, incoming fluid need not pass through that length of the helical throttling passage 46 which in FIG. 1 is located towards the left of the control edge 130. It follows from this that an exceedingly precise setting and control of the throughput quantity can be obtained. Moreover, the pitch and the length of the throttling passage 46 can be freely selected according to particular requirements encountered in actual use.

It will be appreciated that depending upon the position of the throttlIng piston 40 within the sleeve 20, the fluid must flow either through the entire length of the throttling passage 46 (as is the case in the position illustrated in FIG. 1) or only through a portion of the length of the throttling passage 46. In every case, however, the fluid will also flow through the throttling aperture 52. In fact, it is possible by displacing the throttling piston 40 all the way to the left in FIG. 1, to have the fluid flow only through the throttling aperture 52 and not at all through any portion of the throttling passage 46.

When the fluid arrives in the space 55 it is at an intermediate pressure PZ which acts upon the end 81 of the control piston 80, together with the force of the spring 76. It will be appreciated that when it is said that the force acts upon the end 81 the entire cross-sectional area exposed to this force is meant. The regulating piston or control piston 80 will be in a position of equilibrium only when the force T_1 acting upon its opposite or downstream end 83 (again the entire cross-sectional area at this end 83 is meant) is equal to the sum of the forces which act upon its upstream end 81. In other words, the force P1 must equal the force of the springs 76 and the intermediate force PZ. This equilibrium is obtained and maintained by the control edge 88 of the regulating piston or control piston 80. This control edge 88 covers, depending on the axial position of the piston 80, a larger or smaller portion of the area of the control bores 90 and thus provides for the desired constant pressure drop at the throttling piston 40. The fluid passes from the control bores 90 into the annular channel 92 and from there via the outlet 26 to the user device.

Due to the continuous control operation exerted by the control piston 80, the pressure drop P_1:PZ remains constant, independently of the value of the pressure at the inlet 24 and the outlet 26. This means that the once-selected throughput quantity of the valve also remains constant. However, it should be kept in mind that the inlet pressure P_1 should be larger by at least approximately 2 kp/cm² than the outlet or working pressure P_2.

The throughput quantity per unit of time depends on the length of the throttling passage 46 through which the fluid must flow, or upon the selected throughput cross-section of the throttling aperture. This quantity can be determined in accordance with the following formula:

$$Q = (\Delta p \cdot d^4)/(\nu \cdot l \cdot c)$$

wherein

Q = throughput quantity per unit of time
d = hydraulic diameter of the cross-section of the helical throttling passage
ν = viscosity of the fluid flowing through the valve
l = length of that portion of the throttling passage through which the fluid flows
p = pressure drop at the spiral throttle
c = a constant.

The throughput quantity for the throttling aperture 52 is determined by the following formula:

$$Q = \alpha \cdot F \cdot \sqrt{\Delta p \cdot 2g/\gamma}$$

wherein

Q = throughput quantity per unit of time
α = a throughput coefficient
E = flow-through-aperture cross-section
g = gravity acceleration
Δp = pressure drop at the throttling aperture
γ = specific weight of the flowing fluid

If the inlet pressure P_1 is subject to fluctuations, then the control piston 80 whose surface does not contact the inner surface of the sleeve 20 because of its mounting in the ball bearings is capable of following with great precision such pressure fluctuations. Furthermore, because due to the uniform radial play and the small friction between sleeve 20 and control piston 80 a clamping and retention of the latter in the sleeve 20 is impossible, the control bores 90 were capable of being displaced or offset with reference to one another axially of the sleeve 20, so that for instance at the smallest throughput quantities one of the two bores 90 is completely covered by the control edge 88 of the control piston 80 and is thereby closed, so that the fluid can stream outwardly to the outlet 26 only through the remaining control bore 90. This arrangement of the control bores 90 substantially reduces the uncontrollable leakage of fluid over what is known from the conventional valves of this type where the control piston is not mounted in antifriction bearings and where the control bores are located in a plane normal to the longitudinal axis of the control piston in order to prevent hydraulic clamping or seizing of the control piston, particularly at higher pressures.

As already pointed out before, due to the low-friction automatic adjustment of the control piston 80 in dependence upon pressure fluctuations, the once-determined or set throughput quantity can be maintained within a tolerance of approximately ±1 percent with reference to the preset quantity, and this can be maintained constant at all times. Furthermore, a rapid and precise setting of the desired throughput quantity per time unit is made possible due to the fact that the throttling piston 40 is in constant abutment with and thus follows without play precisely any axial displacements of the pin 64, due to the spring acting upon it and urging it against the pin 64.

Because the throughput quantity through the spiral throttle in the embodiment illustrated in the drawing depends on the length of the helical throttling passage **46**, it is possible to correspondingly increase the cross-sectional area of the throttling passage **46** as the throttling passage is made longer, thereby further reducing the possibility that the throttling passage might become clogged.

A non-illustrated flange or similar means is provided via which the valve is connected to the conduit system with which it is to be associated, with suitable seals of course again being provided.

It will be understood that each of the elements described above, or two or more together, may also find a useful application in other types of constructions differing from the types described above.

While the invention has been illustrated and described as embodied in a flow regulating valve, it is not intended to be limited to the details shown, since various modifications and structural changes may be made without departing in any way from the spirit of the present invention.

Without further analysis, the foregoing will so fully reveal the gist of the present invention that others can be applying current knowledge readily adapt it for various applications without omitting features that, from the standpoint of prior art, fairly constitute essential characteristics of the generic or specific aspects of this invention and, therefore, such adaptations should and are intended to be comprehended within the meaning and range of equivalence of the following claims.

What is claimed as new and desired to be protected by Letters Patent is set forth in the appended claims:

1. A flow regulating valve for fluids, comprising a housing having a chamber provided with at least one inlet and at least one outlet; a sleeve accommodated in said chamber and having a longitudinally extending passage bounded by an inner peripheral surface and first and second flow ports communicating with said passage and with said inlet and outlet, respectively, and which are at least in part bounded by flow-controlling edge portions; a throttling piston in said passage displaceable longitudinally thereof with reference to said first ports and to the edges bounding the same, said throttling piston having an outer peripheral surface; a substantially helical fluid-flow throttling passage provided on an extending longitudinally of one of said surfaces; and a control piston in said passage downstream of and axially adjacent to said throttling piston for maintaining constant the flow of fluid through said passage.

2. A flow regulating valve as defined in claim **1**, wherein said one surface is said outer peripheral surface of said throttling portion.

3. A valve as defined in claim **2**, wherein said throttling passage is provided in said outer surface.

4. A valve as defined in claim **2**; and further comprising a pair of annular grooves in said outer peripheral surface at the upstream and downstream ends of said throttling passage, respectively, and communicating with the latter.

5. A valve as defined in claim **2**, said throttling passage being of polygonal cross-sectional configuration.

6. A valve as defined in claim **5**, wherein said throttling passage is of triangular cross-sectional configuration.

7. A valve as defined in claim **5**, wherein said throttling passage is of substantially quadratic cross-sectional configuration.

8. A valve as defined in claim **2**, said throttling passage being of at least substantially semicircular cross-sectional configuration.

9. A valve as defined in claim **2**, wherein the cross-sectional area of said throttling passage is on the order of 0.2 mm^2.

10. A valve as defined in claim **2**, said throttling piston having an outer diameter of substantially 12 mm, and said throttling passage having a length of substantially 15 cm.

11. A valve as defined in claim **3**, said helical throttling passage having a pitch which is greater than its width in the region of said outer surface so as to form between the turns of said helical throttling passage ribs which are bounded by cylindrical sections of said outer surface.

12. A valve as defined in claim **2**, said throttling passage constituting a spiral throttle; and further comprising a throttling aperture provided in said throttling piston downstream of and communicating with said throttling passage.

13. A valve as defined in claim **12**; further comprising a circumferential groove in the outer periphery of the throttling piston and connecting said throttling passage and throttling aperture for communication of the same with one another.

14. A valve as defined in claim **13**, said throttling passage and a first portion of said throttling aperture having at least substantially identical cross-sectional areas.

15. A valve as defined in claim **13**, said throttling aperture including a second portion configurated as an axially extending triangular recess provided in said peripheral surface of said throttling piston and having a corner facing upstream and communicating with said groove.

16. A valve as defined in claim **1**, wherein said one surface is said inner peripheral surface.

17. A valve as defined in claim **1**; and further comprising antifriction bearing means journalling and guiding said control piston for movement thereof axially of said passage.

18. A valve as defined in claim **17**, said bearing means comprising ball bearings.

19. A valve as defined in claim **1**; and further comprising mechanical displacing means for axially displacing said throttling piston in said passage.

20. A valve as defined in claim **1**; and further comprising electrical displacing means for axially displacing said throttling piston in said passage.

21. A valve as defined in claim **1**; and further comprising hydraulic displacing means for axially displacing said throttling piston in said passage.

22. A valve as defined in claim **1**; and further comprising pneumatic displacing means for axially displacing said throttling piston in said passage.

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