

April 23, 1963

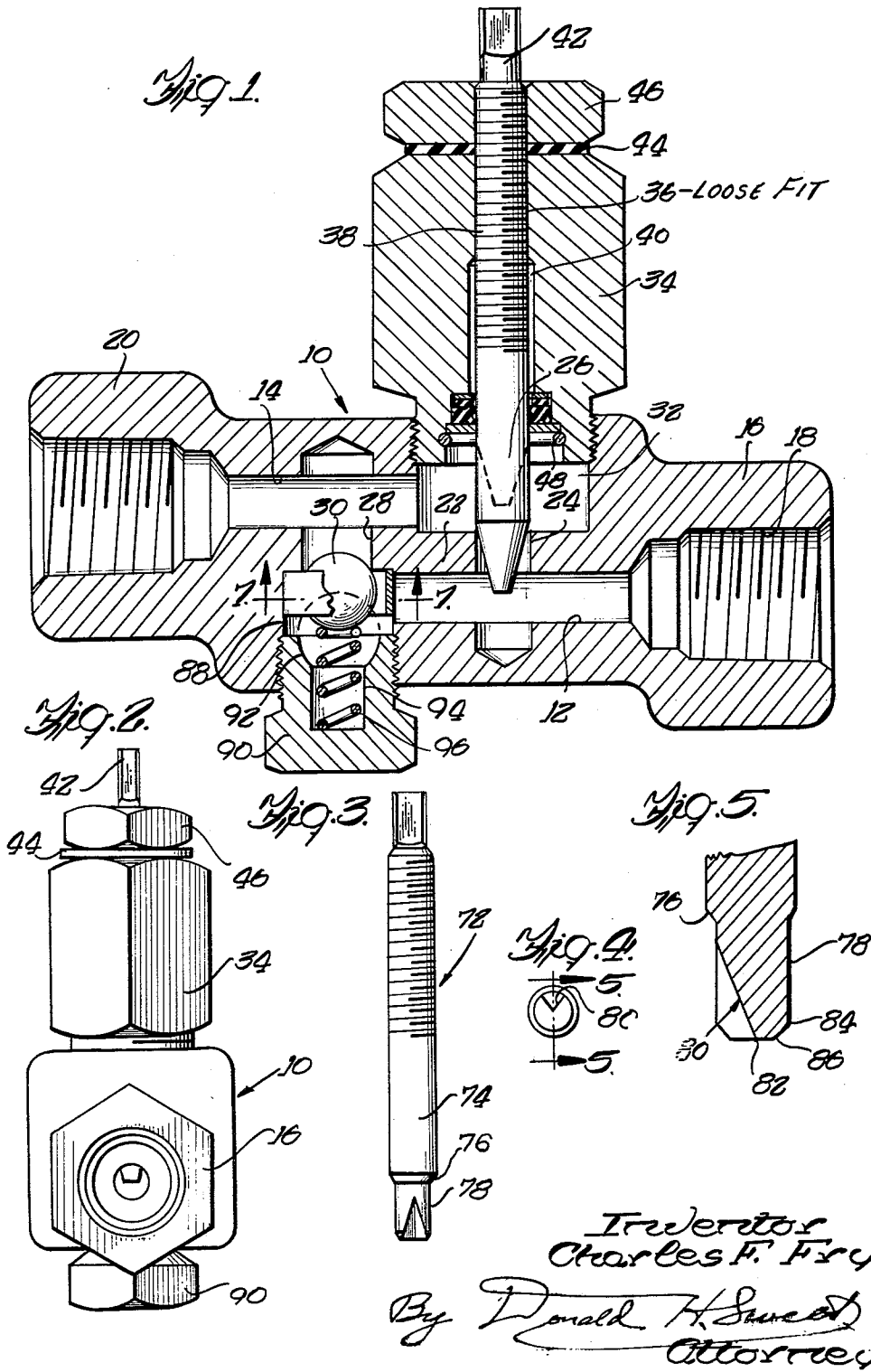
C. F. FRYE

3,086,749

NEEDLE VALVE FOR EXTREME PRESSURE

Filed Dec. 7, 1959

2 Sheets-Sheet 1



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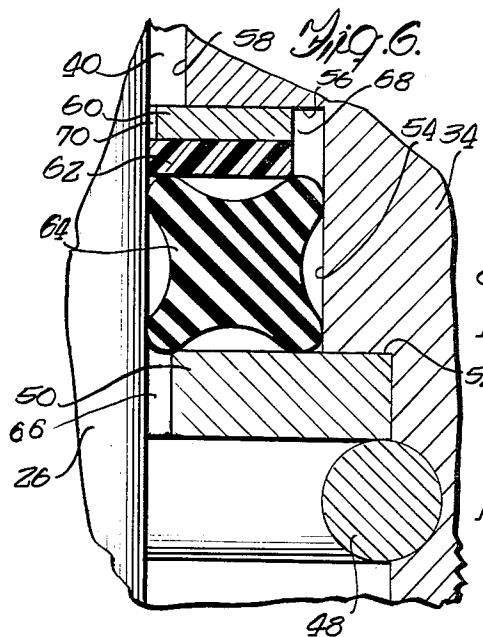
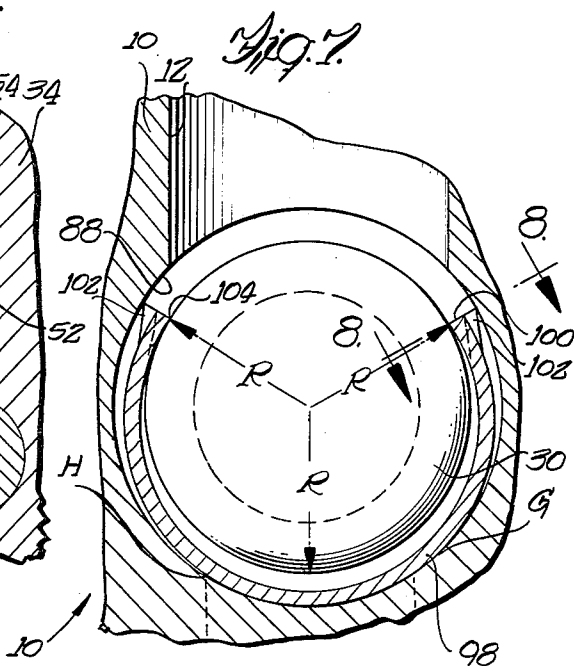
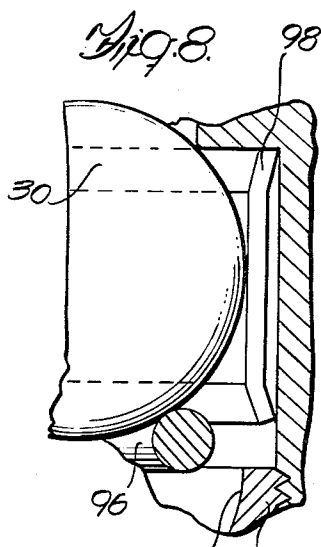
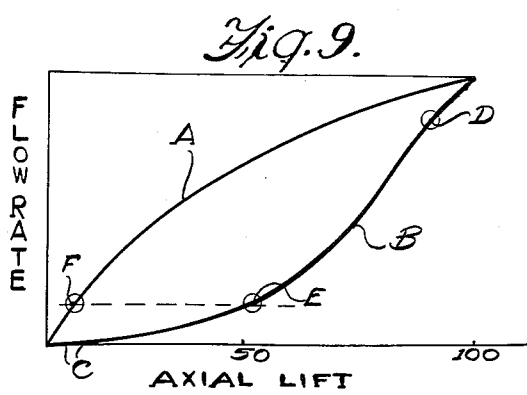
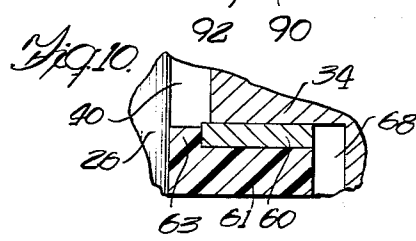
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NEEDLE VALVE FOR EXTREME PRESSURE

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2 Sheets-Sheet 2



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3,086,749

NEEDLE VALVE FOR EXTREME PRESSURE

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9 Claims. (Cl. 251-205)

My invention relates to control valves for regulating fluid flow and includes among its objects and advantages, extremely rapid flow with minimum pressure drop when desired; slower flow calibrated with high precision; and adjustment of calibrated flow at extremely high pressures entirely without leakage and substantially without binding action in the adjustment means due to the high pressure. Further objects and advantages will become apparent as the description proceeds.

In the accompanying drawings:

FIG. 1 is a sectional view of a complete valve unit according to the invention;

FIG. 2 is an end elevation looking from the right end of FIG. 1;

FIG. 3 is an elevation of a modified calibrating needle;

FIG. 4 is an end elevation of the same needle;

FIG. 5 is a fragmentary section as on line 5-5 of FIG. 4;

FIG. 6 is a greatly enlarged section of the packing for the floating needle valve;

FIG. 7 is an enlarged section as on line 7-7 of FIG. 1;

FIG. 8 is a detail section on line 8-8 of FIG. 7;

FIG. 9 is a flow rate diagram; and

FIG. 10 is a detail of a modified washer.

In the embodiment selected to illustrate the invention, the main body of the valve assembly is a forging defining a central body 10 having the shape of a rectangular parallelepipedon with rounded edges and corners. In the position of FIGS. 1 and 2, the body has a lower bore 12, and an upper bore 14. For purposes of identification, the inlet is the passage 12 receiving fluid in the direction that closes the ball check valve, and passage 14 is the outlet. These designations are reversible, depending on the nature of the complete installation. The inlet is continued beyond the main body 10 in a hexagonal boss 16 coaxial with the bore 12 and the bore has stepped enlargements ending in an internally screw threaded outer end portion at 18 adapted to be connected to a supply or delivery pipe. The outlet 14 is continued in a boss 20, which duplicates the boss 16, except that it is offset upwardly instead of downwardly with respect to the body 10.

The bores 14 and 12 are offset far enough to leave a partition, or septum 22, between them. Through this septum, two cross bores are provided. The small cross bore 24 cooperates with a needle valve 26 for securing accurately calibrated throttled flow upwardly through the bore 24. The larger cross bore 28 cooperates with the spring pressed ball 30 in permitting free and unobstructed downward flow when the pressure difference is in that direction and permitting no flow at all when the pressure is in the opposite direction, whereby the rate of flow is determined by the needle valve 26.

This basic relationship has been well-known in the art for a long time and is illustrated, for instance, in my earlier Patent 2,841,174 of July 1, 1959.

High Pressure Seal and Flow Needle Valve

When the fluid employed is a liquid, many of the desirable applications for this valve involve working pressures as high as 5,000 p.s.i., and working pressures twice as great are occasionally desired. Of course, air and other common gaseous mediums cannot be employed at such pressures because they would liquefy. However,

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the advantages of the seal of FIG. 6 are only slightly less significant when the medium is air at working pressures of 500 or 600 p.s.i.

The bore 24 opens into a chamber 32 which receives the threaded lower end of the block 34. This block closes the top of the chamber and houses the needle valve 26 and its packing.

The block 34 has threaded engagement at 36 for a short distance along the threaded portion 38 of the needle valve 26 and is then enlarged to define a chamber 40 encircling the needle valve above the packing. This general arrangement is well-known in the art, but at high pressures extreme difficulty in making adjustments of such a needle valve has been the universal rule, so far as I am aware. According to the invention, the threads for mounting the block 34 on the body 10 are machined with tolerances that are minimum for quantity production operations, but no available precision in this respect can prevent deviations such that the geometrical axis of the bore 24 will often be two- or five-thousandths of an inch offset from the geometrical axis of the threads 36. When this is the case, every time the needle valve is closed, the final closing movement would, theoretically, flex the needle 26 between its threaded engagement with the block 34, and its seat in the bore 24. This introduces serious mechanical stresses and strains in the metal and serious wear on the valve seat and the threads.

According to the invention, no attempt is made to eliminate the lack of alignment between the bore 24 and the threads 36. But screw threads are commonly manufactured with a substantial variety of clearances, including clearances so small that considerable force is required to turn the parts at all, due to the friction in the threads, and proceeding step by step to increased clearances such that the two members in threaded engagement remain loose to various degrees. In the particular embodiment illustrated, a relatively loose, No. 2 female thread in the block 34 and a corresponding thread on the pin 26, does permit the tip of the needle valve to wobble materially without flexure of the valve. Because the threads 36 are short, and the valve seat is separated from the threads by a long space, this wobble is magnified correspondingly. Accordingly, the expense of high precision tight threads at 36 and abnormal stresses and strains in the parts are both eliminated. The commercial designation for the type of thread that happens to fulfil this condition in the embodiment disclosed is the No. 2 thread.

However, the elimination of binding due to imperfect alignment of the valve seat does not eliminate the worst difficulty encountered in the prior art valves. The prior art packings with which I am familiar for effecting the necessary seal in the block 34, not only grip the surface of the needle 26 with rather heavy friction, but they come close to preventing the wobble above described, that is needed to allow for a free seat in the bore 24. They also rarely fail to permit extremely minute leakage up into the chamber 40 when exposed to pressures of the order of magnitude of 5000 p.s.i. for long periods of time. When a condition is reached with the chamber 40 full of liquid, movement of the needle valve 26 in the direction of opening requires the male threads on the needle valve to move up into the female threads in the block 34 and to be replaced at the bottom end with an equal length of the pin 26 of full cross section. This reduces the volume of the chamber 40 by the volume of metal removed from the pin in forming the screw threads. Small as this reduction is, when it has to be made with the chamber 40 already filled with incompressible liquid at 5000 p.s.i., and especially when the threads 36 are of the tight precision fit customarily employed in such heavy duty installations, turning of the

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pin 26 in the direction of opening can be accomplished only by extruding part of the contents of the chamber 40 back through the packing in the opposite direction. Assuming that the normal working pressure was 5000 p.s.i., if the opening movement is abrupt, the momentary pressure in the chamber 40 may easily rise to some such value at 25,000 p.s.i. The pressure reversal on the packing and the momentary abnormally high pressure injures the packing, and the mechanical force that must be applied to the polygonal end 42 to accomplish the displacement becomes surprisingly large. An operator with previous experience may succeed in making the adjustments with a powerful wrench by exerting a steady pull that rotates the stem slowly at the rate of about 30 seconds for 180° of rotation, without twisting the end 42 off the needle.

The conventional friction washer 44 and lock nut 46 perform their conventional functions independent of the problems now under discussion.

Referring now to FIG. 6, the pin 26 passes through a packing held in position by a retaining ring 48 snapped into a semi-circular groove in the block 34. The next packing element is a metallic washer 50 clamped against axial movement between the snap ring 48 and a shoulder 52. This ring is not locked against rotation but there are no forces tending to rotate it. Above the shoulder 52 the block 34 has a bore of intermediate diameter at 54 ending at a second shoulder 56 which extends into the final bore 58 defining the outer wall of the chamber 40.

In the central bore 54, I position at the top a metallic washer 60 and below that a plastic washer 62 and below that a neoprene quad ring 64. These three elements substantially fill the chamber, with the quad ring definitely in contact with the bottom washer 50 but not materially distorted.

It will be noted that the bottom washer 50 has a substantial clearance at 66 between its inner surface and the pin 26. This permits relative radial movement so that the pin 26 can have the wobble previously described. At the top, the metal washer 60 is a free fit on the pin 26 to prevent marring and has a clearance at 68 between its outer surface and the bore 54, of substantially the same radial extent as the clearance 66. The plastic washer 62 is a snug fit on the pin 26 and its outer diameter is the same as that for the metal washer 60. It will be apparent that the full liquid pressure will obtain in the clearance at 66 and that when this pressure is very high the quad ring 64 may be compressed upwardly. The resulting distortion may reduce the clearance spaces on the low pressure side of the quad ring especially with respect to the clearance at 68.

I have found in practice that such a packing as that just described will function very well up to about 1500 p.s.i., without the nylon washer 62. At higher pressures a tendency develops to extrude the neoprene into the joint between the washer 60 and the pin 26. After a substantial portion of the neoprene has been transferred in that way into the chamber 40 the packing is ruined. With the neoprene washer 62 in place, various and repeated adjustments of the pin at 5,000 p.s.i., or higher, still result in no marring of pin 26 or leakage into chamber 40. I believe this to be due to the value for the coefficient of friction between the neoprene of the quad ring 64 and another surface also of plastic, nylon washer 60, compared with the friction between neoprene and metal. Whatever the reason may be, the presence of the plastic washer 62 next the plastic ring 64 accomplishes a surprising increase in the working pressures that can be successfully handled.

Except in very small sizes, I prefer to prevent the washer 60 from even engaging the pin 26 at all. In FIG. 10 the modified nylon washer 61 has a short nipple 63 and there is no sliding movement between the washers 60 and 61. Washer 60 and shoulder 56 carry the mechani-

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cal load, but the high pressure never gets into the space at 68.

In miscellaneous installations, especially for automation, one of the commonest assemblies for such a valve is with a working cylinder connected to the outlet 14 and other valve means for connecting the inlet 12 alternately to a source of high pressure fluid and to atmosphere, or a return line to a storage reservoir.

It will be obvious that the working stroke with fluid entering the inlet 12 can be throttled down by the needle 26 so that the working stroke may cover a period of time from 5 to 500 or even 5,000 times the time it takes for the return stroke to take place with the ball 30 wide open and both passages 14 and 12 directly connected to atmosphere. Under such circumstances the short time for the return becomes relatively immaterial, so far as precise adjustment is concerned, and all that is needed is to get the piston back ready to start the next working stroke. But rather precise adjustment of the working stroke is often required to move a work piece at the proper speed or merely to prevent getting the parts that are being moved to their final position without accelerating them to such high velocities that they injure something at the end of their movement.

Referring now to FIGS. 1, 3 and 9, it will be apparent that axial withdrawal of the needle 26 will open a net area through the passage 24 equal to the difference between the area of the passage and the area of the valve 26. This rate of increase will be greatest when the valve is first cracked open and will decrease materially so that the actual flow increases at a slower rate during the later stages of the opening movement than during the early stages. In FIG. 9 I have indicated the amount of valve opening on the horizontal or X axis and the flow rate on the vertical or the Y axis and the shape of the curve A is typical of the effective response of such a valve when used for throttling. It will be apparent that above the 20 or 30% opening, there is a relatively high sensitivity and relative adjustments can be made by the operator, but close to the closed position, where it is often necessary for the equipment to operate, the sensitivity is low and the rate of response undesirably high.

In FIG. 3, I have indicated an alternative construction in which the threaded portion and top of the needle 72 may be identical with the needle 26 of FIG. 1. The cylindrical portion 74 of full diameter also duplicates the corresponding portion of the pin 26. Below the portion 74 I provide first, a short conical portion 76 adapted to engage the lip of the valve seat when the valve is completely closed. Below the conical portion 76, cylindrical portion 78 extends down as indicated in FIGS. 3 and 5. This portion is only from 0.0005" to 0.00025" smaller than the bore 24. A short upper portion of the obturator 78, next the conical portion 76 is of circular contour. Below that, I fashion a diagonal V-shaped groove 80 that has its side faces in the same plane throughout its length, but its apex, or crotch, is inclined inward diagonally. The extreme bottom end of the obturator is flat at 82, and when the groove 80 reaches the axis of the pin the slot leaves open approximately one-sixth of the area of the end portion 82. A desirable ratio between the radius of the main body 78 of the obturator and the radius of the flat bottom 82 is about three to two, to secure a smooth transition in the curve of FIGURE 9 as the end of the obturator moves out of the bore 24.

As the needle of the FIGS. 3, 4 and 5 moves up, the initial separation of the cone 76 from the valve seat will permit only a very trifling leakage between the obturator 78 and the bore, until the upper end of the groove 80 is reached. Thereafter, the groove increases relatively slowly at first, but more rapidly as the groove itself becomes larger. Finally, when the corner at 84 above the level of the bore lets the conical tip portion 86 start opening a much wider diagonal clearance between the needle and the lip of the bore will be established.

In FIG. 9 the curve B is intended to indicate the type of movement of fluid that can be obtained with a needle according to FIG. 3. Up to some such point as C there is only leakage around the obturator 78. Then the groove 80 produces the result indicated by the curve B up to some such point as D, at which time the area of flow begins to be defined by lip of bore 24 and the face of the cone 86. It will be noted that the flow rate corresponding to the point E on curve B is also obtainable at the point F on curve A, but that the valve displacement to get exactly that rate of flow is about 10 times as great with needle 74.

Long Life, Wide Opening Check Valve

Referring to FIGURES 1, 7 and 8, the ball 30 reciprocates in a vertical bore 88 coaxial with the bore 28. The bore 88 is closed by a threaded plug 90 having a spherical open portion 92 adapted to receive part of the ball 30, and a deeper central bore 94 housing a compression spring 96.

The movement of such a ball is often very rapid and forceful and many such valves must function smoothly for millions of openings and closings without wearing out. Accordingly, the guidance for the movement of the valve becomes a critical matter and it is also necessary to let the ball move down into the pocket 92 so that the path of the fluid from passage 12 to passage 14 is of substantially no greater flow resistance than these passages themselves. Otherwise, the valve structure itself would throttle down the flow and deprive the user of the maximum flow obtainable through passages 12 and 14 of any given size.

The bore 88 is materially larger than required to accommodate the movement of the ball 30 and the primary guidance of the ball 30 does not depend on the walls of the bore 88 but on the interior surface of a C-shaped sleeve 98 seated in the bore 88. Referring especially to FIGURE 7 the sleeve 98, in assembled condition, fits snugly in the bore 88 over substantially 90° of the circumference, up to points indicated at H and G in FIGURE 7. It is originally manufactured with a radius of curvature materially larger than that of the bore 88 and has to be strongly compressed to force it into assembled position. It would remain in perfectly snug contact with the bore 88 throughout its entire extent if the extreme edges 104 were not forced in at each corner by turning small triangular areas at the top and lower ends of those edges diagonally outward, as indicated at 102. These little triangular points 102 tend to gouge into the metal of the body 10 as the sleeve is pushed up into the position of FIGURE 8 and a quite heavy friction between the sleeve and the bore obtains over the entire periphery from point H to point G and also where the horns 102 ride on the bore. Thus, at the upper and lower corners of each end edge, the material of the ring, or sleeve, is forced inward, so that the inner edge of the sleeve lies at 104 in FIGURES 7 and 8 at substantially same radial distance R from the axis of the bore, as the radius R of the ball 30. In FIGURE 7 the location of the seat is indicated in dotted lines, and it will be apparent that the ball has continuous contact over the entire periphery of the seat and is also in substantially actual contact with parallel guides at 104.

During the rapid movement of fluids in through outlet passage 14, and inlet 12, the impact of the fluids coming down through passage 28 moves the ball 30 all the way down into the pocket 92.

Referring to FIGURE 7, it will be noticed that the passage between the edges of the liner 98 has a width equal to the diameter of the passage 12, so that there is no constriction of the flow out into the passage 12. The fluid coming in through passage 14 encounters a body of stationary fluid at the right side of the larger passage 28 and is pushed down diagonally into the end of the passage 12 with minimum turbulence and substantially no

throttling action. The ball occupies the dotted line position indicated in FIGURE 1. This is the flow condition when a power cylinder is being vented to atmosphere, and at such times the action of the valve cannot be too rapid.

The mere cessation of flow to the right ends the dynamic action that holds the ball down and the ball will get back up to the full line position at substantially the instant that the flow ceases. The subsequent very rapid rise of the pressure in passage 12 up to full line pressure anchors the ball firmly on its seat and the needle valve 26 will govern the rate at which the power stroke is completed.

Whether the ball 30 gets fluid pressure below it before it rests on its seat or not, the return to the seat is guided with a precision approximating 0.001", and tests have shown that such a return movement can take place from 5,000,000 to 15,000,000 times without hammering the seat out of round. Prolonged service actually improves the working seal along the line of contact with the ball. With ears 102 at both ends of the sleeve, the guide rails are fully parallel with the axis of the bore, and the ball is guided to closed position on its seat and to wide open position in the socket 92, without any opportunity to deviate or hammer on the various contacts, which are at all times either in physical engagement with it or within one- or two-thousandths of an inch of engagement. Relative movement up to even as much as five-thousandths of an inch will not let the ball develop enough velocity to deliver an impact that willpeen, or otherwise permanently alter, the shape of the parts.

Others may readily adapt the invention for use under various conditions of service by employing one or more of the novel features disclosed, or equivalents thereof. In FIG. 7, I have shown turned out corners 102 at both ends of the guides 104. With valves of large size for operating on gaseous media with frequent, rapid opening and closing, the omission of the remote turned out corners lets the ball edge over a few thousandths of an inch toward the mouth of the passage 12 and ride down the very slightly inclined guides against the stream in the bore 28 to a seating cushioned by the dying stream. With valves of large size operating on liquid media with very high pressures and relatively slow opening and closing velocities, omission of the turned out corners adjacent the valve seat can permit a slightly reduced radial clearance between ball and sleeve when the ball is wide open, and if the full seating flexes the proximate guide ends open a half-thousandth of an inch, there is ample pressure available to maintain the seated condition without power loss, while the increased clearance on the side next the debouching passage 12 avoids any initial delay in letting liquid refill the socket 92 when the ball starts to leave.

As at present advised with respect to the apparent scope of my invention, I desire to claim the following subject matter:

1. A high pressure needle valve adapted to retain high pressure fluid on both sides of the closed valve, comprising, in combination: walls defining an enclosed chamber, a first bore leading out of said chamber, and a second bore coaxial with said first bore within a predetermined manufacturing tolerance; a needle in said first bore; said needle having screw threaded engagement with said first member and a head adapted to engage a seat in said second bore; said head and seat including contact faces shaped to center said head on the axis of said second bore; said screw threaded engagement occupying a fraction of the length of said needle, remote from said seat; said needle having a terminal portion beyond said threaded engagement adapted to receive torque for turning said needle; said screw threaded engagement having predetermined looseness sufficient to permit the head of said needle to wobble; the amount of wobble being not less than said predetermined manufacturing tolerance; and a pressure-tight packing interposed between said needle and said first bore adjacent the inner end of said bore;

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said packing including annular members having radial clearance in said bore, to permit said packing to shift and allow the head of said needle to wobble.

2. A combination according to claim 1, in which said packing includes a pressure-seal washer; a self-sealing annulus of rubber-like consistency engaging said pressure-seal washer on the side facing toward said seat; and a retaining washer engaging said annulus on the side facing toward said seat.

3. A combination according to claim 2, in which said first bore has a shoulder facing toward said seat; said shoulder abutting said pressure-seal washer.

4. A combination according to claim 3, in which said first bore has a first cylindrical enlarged portion contiguous to said first shoulder and receiving said pressure-seal washer and said annulus.

5. A combination according to claim 4, in which said pressure-seal washer fits snugly on said needle and has wobble clearance in said first enlarged bore portion.

6. A combination according to claim 5, in which said bore has a second shoulder at the inner end of said first enlarged portion, and a second enlarged portion contiguous to and extending inwardly from said second shoulder; said retaining washer resting on said second shoulder; said retaining washer fitting snugly in said second enlarged portion and having wobble clearance around said needle.

7. A combination according to claim 5, in which said pressure-seal washer is in two parts; said two parts being a metal washer bearing on said first shoulder and a plastic washer bearing on said metal washer; both washers having wobble clearance in said first enlarged bore portion.

8. A combination according to claim 2, in which said annular self-sealing annulus is a quad ring, having a generally square cross section with four rounded corner lobes and four shallow depressions, one in the middle of each face between the adjacent corner lobes.

9. A high sensitivity needle valve for calibrated flow control comprising, in combination: a septum apertured by a bore to define an annular valve seat having a sharp metal edge; a needle valve having a head adapted to enter said aperture and control flow therethrough; said head having an upper tapered sealing portion; said sealing portion having a large end of greater diameter than said seat and a small end of less diameter than said seat;

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said head having a short cylindrical transition portion immediately below said sealing portion; said bore, when said valve is closed, extending down at least as far as said transition portion, and having an inside diameter with free clearance around said transition portion; said free clearance being the minimum consistent with mass production, and of the order of magnitude of 0.001" of diameter; said head having a calibrating portion below said transition portion; said calibrating portion being longer than said transition portion, and having the geometrical shape of an axial continuation of said transition portion, segmentally cut away by a groove; the cross section of said groove increasing downwardly; the area of the cross section of said groove at any transverse plane being substantially proportional to the square of the distance from the upper end of said groove down to said plane; said groove having plane sides subtending a dihedral angle of approximately 64°; the apex of said dihedral angle extending diagonally downward and radially inward and ending substantially at the axis of said head; the inward inclination of said apex being approximately 23°; said calibrating portion having a flat bottom end face normal to the head axis; said flat end face extending radially out about two-thirds as far as said transition portion; said calibrating portion having a short conical end running up at an angle of substantially 45° from said flat end face to the outer periphery of said calibrating portion; and manual adjustment means for adjusting the axial position of said valve and head.

References Cited in the file of this patent

UNITED STATES PATENTS

214,286	Enright	Apr. 15, 1879
351,828	Bushnell	Nov. 2, 1886
1,151,230	Witson	Aug. 24, 1915
1,289,104	Britton	Dec. 31, 1918
1,671,609	Roberts	May 29, 1928
2,013,822	Bradbury	Sept. 10, 1935
2,567,527	Parks	Sept. 11, 1951
2,602,629	McChesney	July 8, 1952
2,757,993	Flick	Aug. 7, 1956
2,765,185	Mott	Oct. 2, 1956
2,841,174	Frye	July 1, 1958
2,980,392	Greenwood	Apr. 18, 1961
2,992,817	Templeton	July 18, 1961