

BALL VALVE

This invention relates to ball valves.

The invention is particularly applicable to ball
5 valves of the type having a so-called "floating ball"
and will be described with particular reference thereto.
However, it will become readily apparent to those
skilled in the art that the invention is capable of
broader applications and could be adapted for use in
10 other types and styles of ball valves.

Ball valve constructions in commercial use
typically employ annular seats or seat rings formed
of a resilient and deformable plastics material for
sealing engagement with the ball member. A pair of
15 such seat rings is positioned adjacent inlet and
outlet openings. The ball member itself is mounted
for a slight amount of free movement or shifting
axially of the seat when the ball member is in a
valve-closed position under fluid pressure conditions.
20 Such shifting causes the ball member to act against and
flex the downstream seat ring to enhance its sealing
engagement with the ball member. The amount of such
flexing varies in accordance with the fluid pressure
involved.

25 The floating ball concept has been successfully
used in many arrangements and designs for small ball

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valves, and for lower pressure ball valves. As fluid pressure increases and causes downstream shifting of the ball member, the ball member is moved away from the upstream seat and all the force is applied to the downstream seat. Since the seat comprises an annulus of a smaller area, seat stress is always greater than the fluid pressure. In addition, as the valve size itself increases, the ball force increases as a function of the square of the seal diameter.

However, typically it is impractical to increase the annular seat area at the same rate. Therefore, in a large valve or at high pressure, the seat stress can reach levels that may crush ordinary plastic type seats.

An overall objective of ball valve seat designs is to obtain a valve which will seal at both low pressure and high pressure and which will not require an unreasonable amount of operating torque for rotating the ball member. The floating ball type of concept is particularly advantageous in that it facilitates elimination of the more expensive trunnion-mounted type of ball member as well as more complex pressure actuated valve seats.

Various forms and types of ball valve seat designs have heretofore been suggested and employed in the industry, all with varying degrees of success. It has been found that the defects present in most

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prior ball valve seat designs are such that the devices themselves are of limited economic and practical value.

Prior ball valves employing an elementary seat have included a seat design typically comprised of annular resilient plastic seats merely compacted between the ball member and opposed end fittings. The ball-engaging surface of each seat includes a contour which, in mating fashion, engages the periphery of the ball member upon compacting assembly.

Elementary seat designs suffer from a number of inherent problems. In a valve-closed position, the upstream pressure on the ball member forces the ball member into engagement with the downstream seat to seal the valve in a virtual check valve type action. Since the pressure on the ball member is substantially responsible for all the sealing force, at low pressure or vacuum conditions, a low sealing force is present and results in a leaking valve with leak paths between the seats and ball member and between the outer surfaces of the seats and supporting flanges or shoulders of the opposed end fittings. Elementary seat designs provide no compensation for wear or tolerance errors in the annular seats and thus associated leakage problems are compounded over a period of time.

At conditions where the upstream seat seals against the ball member, the additional surface area

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of the upstream seat receiving load pressure is opposed to the downstream seat thereby increasing the load upon the downstream seat resulting in increased downstream seat stress and distortion. The force on the upstream seat to its outer diameter surface is translated through the ball member to the downstream seat. Such increased load area results in increased seat stress and thus causes an undersirable wear rate for the downstream seat. The problems of increased seat stress and wear are particularly noticeable on the larger ball valve sizes.

Also at conditions including upstream seat sealing against the ball member at a valve-closed position, a problem occurs with a "blowing-in" bulge on the upstream seat at the first opening of the valve. As the valve is being opened, the upstream seat must momentarily span the opening or fluid passage extending through the ball and hold back fluid pressure. During this short period of time until the upstream seat seal is relieved by further opening of the valve, the portion of the seat spanning the fluid passage can deform into the passage under fluid pressure. With a small opening in the ball member, the seat is quite rigid when loaded as a beam in bending and can easily bridge the gap. As the valve size and ball opening increase, the section modulus of the seat does not

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proportionately increase to retain the same stiffness. Thus, the seat may deflect and deform further into the ball opening to form a bulge in that sector of the seat ring.

5 As repetitive cycling occurs throughout the life of the valve, the bulge in the upstream seat becomes more pronounced and actually operates to cam a floating ball member against the downstream seat as the ball member is rotated to the valve-open position. The
10 resulting camming action further stresses, even cutting, the downstream seat, and may ultimately distort and wear the seat into a non-operative condition. The bulge on the inlet seat cams the ball member off-centre and holds it away from both seats while closed
15 causing seat leakage in the closed position. It is particularly noteworthy that the industry focussed on the downstream seat assembly for purposes of later improving ball valve operation to prevent valve failures and problems caused by blowing-in bulges. In
20 actuality, however, the upstream seat design was the cause of the problem. The upstream seat permitted the blowing-in bulge and the resultant camming action which was reflected to the downstream seat.

Where soft plastic seats are employed, such
25 as those made of polytetrafluoroethylene (PTFE), particular problems encountered with cold flow creep

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in the seat at non-contained portions aggravate wear and undesirable seat deformation.

Quite a part from the blowing-in bulge, a sealing upstream seat has a generally unsupported annular front face that can creep in towards the centre of the valve when the valve is closed and subjected for a long duration to static fluid pressure drop across the upstream seat. In the elementary seat design, the outside diameter of the seat ring is normally supported only by a shoulder in the valve body. The area of the seat and, therefore, the force acting on it, increases as the square of the diameter. Since the supporting shoulder is usually quite narrow and its area is more closely related to the seat circumference, the supporting shoulder width only increases in a linear fashion. This results in a narrow supporting shoulder which allows the seat to shear past it. Ultimately, as the seat creeps forward, it can cave-in entirely or before that point it typically curls into the orifice of the ball such that, when the valve is operated, the seat is torn. This problem also becomes more pronounced as the valve size increases.

The downstream seat can also creep in towards the centre of the valve by its generally unsupported annular front face. When the valve is in the closed position, the seat may be displaced upstream by the

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ball member pushing into it under pressure load.

Where large ball members are employed in large valves, the ball member weight additionally contributes to the deformation and cold flow problems.

- 5 As the seats deform after a period of use, the ball member may sag to the bottom of the valve, thereby providing a clearance at the top of the valve for a leak path.

- An evolutionary step beyond the foregoing
- 10 elementary seat design is the contoured seat in which the ball-engaging surface of the seat includes a contour different from that of the outer periphery of the ball member. The objective of this contoured design is to obtain a narrow band or line contact
- 15 between the seat and the ball member which flexes slightly under loads and retains contact with the ball member at low pressure situations to give a sealing stress at low pressure, whereby to minimize leakage between the ball member and seat. However, in addition
- 20 to retaining the other problems associated with the elementary seat design, the contoured seat suffers from the additional problems of extreme contact line distortion in high pressure conditions and rapid wear due to high seat stress since the entire sealing force
- 25 was substantially borne by the contact line band. The

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extreme distortion and rapid wear further aggravated the other problems.

In order to overcome the particular wear problems of contoured seat designs, the flexible seat design, as is illustrated in USA Patent No. 2,945,666 evolved. The spring action of the flexible seat provided low pressure contact, wear compensation, tolerance compensation and ball sag compensation. However, this design suffers from the particular problem of providing improved operation for only a short period of time.

The seat is typically constructed of a plastics material which lacks an elastic memory and which deflects under load but does not return to its original shape when the load is removed. In addition, the flexible seat design employs less seat material at the contact line with the ball member. After short periods of exposure to high pressure, the flexible seat will distort to the shape of an elementary seat which is severely worn.

The problems of creep at non-contained portions of both upstream and downstream seats, high downstream seat load due to an upstream seat seal, seat-to-shoulder leak paths, and seat blowing-in bulges all remain with both the flexible seat design and the contoured seat design.

The industry next developed a ball valve seat design comprised of a seat having a seat ring with a metal frusto-conical disc spring disposed in operative engagement with the seat ring rear face. The disc spring allows for improved compensation for seat ring wear, distortion and creep as well as seat tolerance and ball sag, while providing an elastic support which effects a low pressure force bias. The disc spring provides seat elasticity not dependent on the elastic memory of the plastic seat material. However, while this is an improvement over the before-mentioned flexible seat design, the disc spring must, nevertheless, provide its elastic force while overcoming the elastic resistance of the plastic seat material.

A particular problem with this design is that the action of the spring ensures that the upstream seat seals against the ball member. Thus, the problem of the elementary seat design having an effective load area on the downstream seat substantially equal to the surface of the upstream seat is not mitigated. The remaining unsolved problems include a blowing-in bulge, a weak seal at the seat-to-support shoulder contact area and creep on the non-contained portions of both the upstream and downstream seat rings. In addition, a problem with such design is that the disc spring need overcome the elasticity of the seat ring to effect the low pressure force bias.

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In order to meet the problems caused by an upstream or inlet sealing seat, the industry developed an upstream seat bypass arrangement, such as is typically illustrated in USA Patent No. 2,930,576. Various upstream seat bypass methods and designs are employed including seat notches, grooves, holes, body orifices, and check valves. These variations are all for the purpose of communicating line pressure around the upstream seat at the valve-closed position for relieving the upstream seat ring, for reducing the load area by which the ball member imparts a load on the downstream seat, and thus for reducing seat stress, creep and wear at the downstream seat ring. Also upstream seat blow-in and creep under static pressure is circumvented. In addition, the necessary operating torque for opening and closing the valve is lessened.

Particular problems encountered with the upstream seat bypass designs include the need to provide dynamic seals due to bi-directional sealing of a ball-to-seat seal and a seat-to-body seal. A necessary consequence of such complicated bi-directional sealing is reduced reliability of such elaborate sealing. Problems of low pressure seat leakage, seat-to-shoulder leakage, compensation for tolerance, sag and wear are not addressed.

Plugging of the bypass routes with contaminants in the valve system fluid frequently occurs, especially where the valve is required to operate in a dirty

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environment. Where a high viscosity fluid is employed a transient delay is often necessary before pressures equalize about the inlet seat thereby allowing the bypass routes to operate correctly. In both cases this seat design then assumes all the problems of an elementary seat design including a blowing-in bulge and cold flow creep at the non-contained portions of the seat rings.

A recent step in the evolution of ball valve designs comprises the provision of a support ring opposite the disc spring to contain the plastic seat ring, as described in UK Patent specification No. 2065276A. The support ring operates to provide additional seat ring support for bridging the ball passage at valve opening to minimize the blowing-in bulge and the associated reflected distortion on the downstream seat. Additionally, the support ring confines the seat to minimize cold flow creep distortion.

However, those seat designs which include disc springs and support rings are not without problems. Specifically, as with the before-mentioned disc spring and plastic seat design, the spring reserve of the seat-spring combination is determined by the spring only which actually must overcome the plastic resistance of the seat in order to move against the ball member and compensate for wear, tolerance and ball

sag. The support ring, in preventing the seat from caving in also hinders the seat from moving forward against the ball member. Thus, the seat is actually obstructed from moving against the ball member by the support ring. While the support ring provides support against seat creep and blow-in bulge, it holds back the seat and spring combination from compensating for wear, tolerance, and ball sag. As the valve is cycled and the seats are worn, the disc spring forces must overcome both the elasticity of the seat ring and the rigid support of the support ring to effect seat to ball engagement.

It has, therefore, been desired to develop ball valve, e.g., of the floating ball type, which could be employed with higher fluid system pressures while producing a longer life span than has heretofore been possible. Preferably, such a design would eliminate the necessity for utilizing a trunnion-mounted ball member as is generally conventional for ball valves employed in elevated fluid system pressure conditions. Trunnion mountings are not considered practical unless the valve is quite large because such mountings substantially increase the size, complexity and cost of the valves.

An object of the invention is the provision of a new and improved ball valve which allows higher

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pressure ratings and greater cycle life to be obtained from a floating ball type of ball valve.

Another object of the present invention is the provision of such a ball valve which will effect valve sealing even at very low fluid system pressures.

Still another object of the present invention is the provision of a new and improved ball valve which permits the seat rings to sealingly engage the ball member over a surface of substantially equal stress distribution.

The present invention contemplates a new and improved construction which overcomes all of the above referred to problems and others and provides a new and improved ball valve which facilitates increased pressure capabilities and extends the effective valve life and wherein the seats may be formed from a wide variety of materials to suit a wide range of operating conditions or parameters.

The invention resides in a ball valve including a valve body having a generally cylindrical fluid flow passageway; a ball member disposed in said passageway and mounted for selective rotation between valve-open and closed positions with said ball member being shiftable generally axially in said passageway under fluid pressure conditions at least when said valve is in said closed position; a pair of annular seat rings

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disposed in said passageway on opposite sides of said ball member between said ball member and associated valve body shoulders; a pair of annular disc springs interposed between each seat ring and the associated shoulder for continuously urging said seat rings towards sealing engagement with said ball member; and a pair of rigid annular bearing surfaces positioned in said passageway on opposite sides of said ball member axially inwardly of said seat rings to provide support for the associated seat ring, each of said seat rings having a continuous annular surface facing said ball member, which surface has an arcuate contour in a relaxed condition defined by a radius greater than the radius of said ball member, such that, upon assembly of said ball valve, said seat ring is rotatably flexed about its outer periphery generally away from its associated bearing surface to direct said continuous annular surface of said seat ring into continuous sealing engagement with said ball member, said continuous surface having an arcuate contour in the assembled and flexed but fluid-pressure-free condition defined by a radius generally equivalent to said ball member radius.

The invention also resides in a ball valve including a valve body having a generally cylindrical fluid flow passageway; a ball member disposed in said

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passageway mounted for selective rotation between valve-opened and closed positions with said ball member being shiftable generally axially in said passageway under fluid pressure conditions at least when said valve is in said closed position; a pair of annular seat rings disposed in said passageway on opposite sides of said ball member between said ball member and associated valve body shoulders; a pair of annular disc springs interposed between each seat ring and the associated shoulder for continuously urging said seat rings towards sealing engagement with said ball member; and a pair of annular bearing surfaces positioned in said passageway on opposite sides of said ball member axially inwardly of said seat rings to provide support for the associated seat ring, each seat ring having an outer periphery in abutting engagement with a side wall of said fluid flow passageway, a first surface facing an associated one of said pair of annular disc springs, a second surface facing an associated one of said pair of annular bearing surfaces, and a third surface facing said ball member and having, at a pre-assembly reference position, a radius of curvature greater than the radius of curvature of said ball member and being spaced by a radial gap at a radially innermost end of said second surface from said ball member, and having,

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upon assembly and under fluid-pressure-free conditions, a radius of curvature substantially the same as said radius of curvature of said ball member and engaging said ball member in mating fashion over the extent of said third surface, said seat ring
5 second surface further being spaced, upon assembly, from a portion of said annular bearing surface to provide a flexure reserve gap for said seat ring.

Thus, a seat ring with its own mechanical
10 elasticity is co-operatively disposed with an associated disc spring and reinforcing ring to sealingly engage the ball member with a substantially uniform stress distribution over a continuous ball-engaging surface which faces the ball member.

15 Two such seat rings are disposed on opposite sides of the ball member and are continuously urged toward the ball member for maintaining it properly positioned in the valve body and for providing valve sealing regardless of the fluid system pressure. The even
20 distribution of stress over the seat ring substantially improves wear resistance in either high or low fluid system pressure conditions to prolong the valve life

The present invention includes a ball valve
25 comprising a body having a central passageway, a ball member in the passageway and mounted for selective rotation between valve-open and closed positions to control fluid flow through said valve; a pair of radially inward extending continuous shoulders in said
30 passageway disposed circumferentially thereof on opposite sides of said ball member;

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a pair of radially inwardly extending counterbores in said passageway disposed on opposite sides of said ball member, each of said counterbores having an inner end wall and being circumferentially continuous; and

5 a pair of composite seat member assemblies positioned axially in said passageway on opposite sides of said ball member for fluid-sealing engagement with said ball member, each of said pair comprising: a reinforcing ring having a central opening, a first

10 surface facing an associated one of said shoulders, a second surface abutting an associated one of said counterbore end walls and a third surface generally facing said ball member in a spaced relationship therefrom; a seat ring adapted for elastic flexure

15 generally towards and away from said reinforcing ring and including a central opening, a first surface generally facing said associated one of said shoulders, a second surface abutting said reinforcing ring first surface, and a third surface facing said ball member

20 for sealing engagement with said ball member, said seat ring third surface being contoured and having a radius of curvature in a relaxed unassembled condition greater than the radius of said ball member; and

25 a disc spring having a central opening and a generally frusto-conical configuration in an unstressed condition interposed between said seat ring and said associated

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one of said shoulders; said ball member and pair of composite seat member assemblies being sized so that, when assembled axially between said shoulders, each seat ring is flexed away from an associated reinforcing ring and stressed to have a radius of curvature at said seat ring third surface generally equivalent to the radius of said ball member and is disposed in sealing engagement with said ball member over the extent of said seat ring third surface, even under fluid-pressure-free conditions, each disc spring being stressed towards a flattened condition for continuously urging the third surface of the associated seat ring towards engagement with said ball member.

The spring seat ring is preferably adapted for rotational-type flexure generally towards and away from the reinforcing ring.

In accordance with a preferred aspect of the invention, the ball member and pair of composite seat members are sized so that, when assembled axially between the shoulders from a preassembly reference position, the seat rings and disc springs are flexed and stressed to have an axial displacement at the central openings thereof equal to an axial spacing of radially innermost ends of the seat ring second surfaces from the ball member plus an axial displacement of the disc spring at an equiradial position to the axial spacing.

The invention is further described by way of example, with reference to the accompanying drawings, wherein;

Figure 1 is a longitudinal sectional view
5 through a ball valve in accordance with a preferred embodiment of the invention;

Figure 2 is an enlarged cross-sectional view
of the upstream seat assembly just prior to valve
make-up and with the ball member omitted for ease of
10 illustration;

Figure 3 is an enlarged cross-sectional view
of the downstream seat assembly shown just prior to
valve make-up and with the ball member omitted for
ease of illustration;

Figure 4 is a somewhat schematic view of the
15 preferred seat assembly construction shown just
prior to valve make-up for showing various co-operative
relationships between the valve components;

Figure 5 is a slightly enlarged view of a
20 portion of the valve of Figure 1 showing the ball
member in the valve-open position under a no-load
condition;

Figure 6 is an enlarged cross-sectional view
of a portion of the upstream seat assembly of the
25 valve shown in Figure 5;

Figure 7 is an enlarged cross-sectional view
of a portion of the downstream seat assembly of the

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valve shown in Figure 5;

Figure 8 is a cross-sectional view similar to Figure 5 but with the valve in a closed position under the influence of elevated fluid system pressures and at such time that the seat assemblies have undergone considerable wear;

Figure 9 is a enlarged cross-sectional view showing a portion of the upstream seat assembly of the valve shown in Figure 8; and

Figure 10 is an enlarged cross-sectional view of a portion of the downstream seat assembly of the valve shown in Figure 8.

Referring now to the drawings wherein the showings are for purposes of illustrating the preferred embodiment of the invention only and not for purposes of limiting same, Figure 1 shows a ball valve A having a pair of opposed seat assemblies B disposed on opposite sides of a floating type spherical ball member C.

More particularly, and with reference to Figures 1 and 5, ball valve A includes a body or housing 10 having a main or central body section 12 and opposed end fittings 14, 15. Seat assemblies B and ball member C are mounted within the main body section 12 and the ball member is arranged for selective rotation by a stem and actuating handle assembly 18.

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The details of all portions of the valve illustrated in Figures 1, 5 and 8, except for the ball and seat ring assemblies, may be modified as desired and/or necessary to accommodate different types or styles of ball valve constructions. In general, however, and for purposes of describing a preferred embodiment of the present invention, the valve body includes a generally cylindrical central passageway or axially extending fluid flow opening 20 which is only slightly larger in diameter than ball member C. Each of end fittings 14, 16 is releasably connected to central body section 12 by a plurality of longitudinally extending tie bolts 22 (Figure 1). The end fittings are also provided with internal screw threads 24, 26 or any other convenient means to accommodate connecting the valve to an associated fluid system or piping.

The stem and actuating handle assembly 18 includes a stem member 30 having a lower end 32 configured as shown for sliding receipt in a slot or groove 34 in the upper end of ball member C. This arrangement allows the ball member to be rotated between valve-open and closed positions while at the same time permitting the ball member to have some freedom of movement for shifting axially in valve body passageway 20 when the valve is in a closed position and fluid pressure is acting on the ball member.

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Stem member 30 extends outwardly through an opening 36 in central body section 12. Suitable packing rings 38, 40 and 42 are positioned in opening 36 and sealingly engage the opening and stem member 30. As shown, lower packing ring 42 rests upon an inwardly extending flange 44 formed within opening 36. A split thrust washer 46 is positioned below flange 44 and is clamped thereto by an outwardly extending shoulder or flange 48 formed at the base of stem member 30. The stem member is held in position by a packing gland 50 and a packing nut 52. As shown in Figure 1, tightening of packing nut 52 applies pressure to packing rings 38, 40, 42 to ensure a fluid-tight seal about the stem.

Although it is possible to actuate the valve stem by many different types of actuators, including both manual and automatic, a handle 54 has been shown. This handle is releasably secured to stem member 30 by a nut 56 which clamps the handle to the top of packing nut 52. A co-operating flat 58 is advantageously formed on the exterior of the stem member for association with a flat (not shown) in the handle opening for properly positioning the handle on the stem member. However, the position of the handle and, in turn, the position of ball member C are limited by depending stop members 62, 64 carried by handle 54. These stop

members engage suitable surfaces on central body section 12 to provide fixed stops for the valve in the full-open and full-closed positions.

With continued reference to both Figures 1 and 5, the pair of seat ring assemblies B disposed on opposite sides of ball member C are clampingly retained in position on opposite sides of the ball member adjacent opposite ends of the main body section passageway or opening 20. In the preferred embodiment hereunder discussion, the seat ring assemblies are located substantially equidistantly from and on diametrically opposite sides of the axis of rotation of the ball member and include radially innermost central openings 66, 68. While the seat ring assemblies could be maintained in position by many different or alternative arrangements, they are shown in the preferred embodiment as being located by shoulders 70, 72 defined by end faces 74, 76 of end fittings 14, 16, respectively (Figure 5). The inward limit of movement of the seat ring assemblies is defined by a pair of shoulders or steps 78, 80 which are formed by the inner end walls of counterbores extending inwardly of valve body passageway or opening 20. Still further, a seal is provided between central body section 12 and end fittings 14, 16 by means of O-rings 82, 84 which are received in second counterbores 86, 88,

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respectively. Each O-ring is disposed about the outer circumference or outer peripheral surface of a portion of the associated seat ring assembly B.

5 The structural details of ball valve A described hereinabove are with reference to the preferred valve construction. It will be readily apparent to those skilled in the art, however, that modifications may readily be made thereto to accommodate particular operational needs and/or requirements. Such changes
10 are not deemed to affect the overall intent or scope of the present invention.

With reference to Figures 2, 3 and 4, description will be made of the specific details of seat assemblies B. Figure 2 shows the upstream seat assembly disposed
15 adjacent end fitting 14 prior to valve make-up. Figure 3 shows the downstream seat assembly disposed adjacent end fitting 16 and is identical to the upstream seat assembly. The ball member C has been omitted for ease of understanding and appreciating the seat assembly
20 construction. Figure 4 shows a somewhat schematic view of a seat assembly just prior to make-up where the ball member C has been included.

The seat assemblies are each preferably comprised of three components, i.e., reinforcing or support rings
25 100, 102, seat rings 104, 106 and frusto-conical disc springs 108, 110, respectively. Reinforcing rings 100,

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102 have an annular configuration including a central opening and are constructed from a rigid material, such as steel or other suitable metal. As shown in Figures 2 and 3, reinforcing rings 100, 102 include first

5 circumferentially continuous surfaces or end faces 112, 114 which face associated shoulders 70, 72 of end fittings 14, 16. Second continuous surface 116, 118 face and abut counterbore end walls or steps 78, 80 of main body section passageway 20 to positively

10 establish a forwardmost or home position for the reinforcing rings. A third continuous surface 120, 122 of each faces generally toward the ball member (not shown), but is dimensioned to be spaced therefrom in order to prevent any abutment or inter-

15 ference therewith and to inhibit distortion and displacement of the associated seat ring between the reinforcing ring third surfaces 120, 122 and the ball member. The outer circumferences or peripheral surfaces 124, 126 of rings 100, 102 are closely disposed

20 against the sidewall of passageway 20. Outer surfaces 124, 126 and second surfaces 116, 118 may alternatively include a flange or step configured and dimensioned to engage shoulders 78, 80 to further extent portions of support rings 100, 102 axially inward of passageway

25 20 where overall valve dimensions may so require. However, such an additional flange is avoided in the

preferred embodiment shown to obviate additional machining steps.

Referring still to Figures 2 and 3, seat rings 104, 106 also comprise annular or ring-like members having central openings 66, 68 therethrough which are smaller in size than the central openings of the reinforcing rings 100, 102. The seat rings are adapted for flexure generally towards and away from the reinforcing rings 100, 102. Seat rings 104, 106 are configured to exert a spring-like mechanical elasticity against the ball member. First surfaces 128, 130 generally face the associated shoulders 70, 72 of end fittings 14, 16 and, in relation to shoulders 70, 72, are inclined or tapered away from end faces 74, 76 in an unstressed and unflexed condition. Seat ring second surfaces 132, 134 face reinforcing rings 100, 102 for bearing engagement and support against reinforcing ring first surfaces 112, 114. Seat ring third surfaces or ball engaging surfaces 136, 138 generally face the ball member for fluid sealing engagement therewith. Flanges or lips 140, 142 extend axially outward of seat ring first surfaces 128, 130 at outer peripheral surfaces 144, 146. These lips or flanges are preferably continuous about the seat rings and located so that their radially inner surfaces generally correspond to the outside

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diameter of conical disc springs 108, 110. Flanges or lips 140, 142 are bevelled at the radially outermost areas thereof and are slightly rolled over the radially outer edges of the disc springs in the manner shown. While not necessary, this arrangement advantageously maintains the seat rings and disc springs together as a subassembly.

In the preferred arrangement of the invention, seat rings 104, 106 are constructed from a resilient plastic material such as polytetrafluoroethylene or polyethylene. It should be readily appreciated, however, that other types of material, such as acetyl resins, could also be advantageously utilized. The particular material chosen will, to some extent, be dependent upon the operating conditions to which the valve will be subjected.

The diameter at the outer ends 152, 154 of the disc springs is such that a disc spring may be received within the cylindrical cavity defined by the inner wall of the seat ring axial flanges 140, 142 and first surfaces 128, 130. The inner diameter of the springs at their inner ends 156, 158 is substantially equal to the diameter of the opening 66, 68 through the associated seat ring 104, 106.

The disc springs are selected so that their force is sufficient under partial deflection to continuously urge the seat rings towards the ball member.

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The springs must also allow stressing or compression thereof towards a flattened condition to accommodate shifting of the ball member and engagement with seat ring third surfaces 136, 138. In the preferred embodiment here under discussion, seat ring first surfaces 128, 130 are configured so as to substantially fully stress disc springs 108, 110 at valve make-up, i.e., to move the disc springs from a frusto-conical shape towards a substantially flattened configuration.

With reference to Figure 4, the particular dimensional relationships between each seat assembly B and ball member C will be specifically discussed. Seat assemblies B will be discussed with particular reference to the upstream seat assembly disposed at end fitting 14.

It should be appreciated that the downstream seat assembly disposed at end fitting 16 is identical thereto unless otherwise specifically noted. Figure 4 is a schematic representation of seat assembly B and ball member C just prior to make-up at a preassembly reference position. In this reference position, disc spring 108 and seat ring 104 are relaxed and contact is made between seat ring 104 and ball member C at an annular contact line 160 disposed at the radial innermost end of third or ball-engaging surface 136 of seat ring 104.

Seat ring 104 engages ball member C upon assembly such that a substantially uniform stress distribution occurs along the entire extent of the third surface 136 of seat ring 104. Since seat ring 104 is configured to exert a

spring-like mechanical elasticity against the ball member C upon assembly independently of disc springs 108, 110, the arcuate contour of ball-engaging surface 136 must be configured to distort in a predictable and controlled manner for accomplishing the uniform stress distribution objective. It has been found that, in order to obtain this objective, the radius of curvature of surface 136 in an unstressed, unassembled condition must be greater than the radius of curvature on the outer surface of the ball member C. In addition, the seat ring 104 and disc spring 108 are sized so that, when assembled axially between valve body shoulder 70 and ball member C, the seat ring is flexed outwardly towards associated shoulder 70 and stressed to have a radius of curvature at surface 136 generally equivalent to the radius of the ball member, thus sealingly engaging the ball member over the entire extent of the surface 136. The disc spring 108 is simultaneously stressed towards a flattened condition and, therefore, additionally acts to continuously urge third surface 136 of seat ring 104 towards engagement with the ball member. Such a uniformly stressed, full surface engagement will occur when the ball member C and composite seat members B are sized so that, at valve assembly from the preassembly reference position of Figure 4, the seat ring 104 and disc spring 108 are flexed and stressed to have an axial displacement at the area of central opening

66 of the seat assembly equal to an axial spacing of a radially innermost end of second surface 132 of seat ring 104 from the ball member C plus an axial displacement of the disc spring 108 from shoulder end face 74 at an equiradial position to the said axial spacing.

In other words, and with continued reference to Figure 4, seating surface 136 contorts to match the spherical surface of the ball member where dimension K is equal to the sum of dimension I plus dimension J. Dimension K represents the axial displacement at central opening 66 comprising a displacement of disc spring inner end 156 towards the end face 74 of shoulder 70. Dimension J represents the axial spacing of the radially innermost end of seat ring second surface 132 from the ball member C. Dimension I represents the axial displacement of disc spring 108 from end face 74 of shoulder 70 at an equiradial position to the radially innermost end of seat ring second surface 132.

Recognizing this axial spacing or relationship, the diameter D_s of the spherical surface of the seat ring, when relaxed, and the radial gap, dimension g, between ball member C and the seat ring at the radially innermost end of second surface 132 may be calculated, given certain readily determinable dimensions of the ball valve A. More specifically, the radial gap g and the diameter D_s of the spherical surface of the seat ring,

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when relaxed, can be determined using known geometrical methods, given the inside diameter d of the seat ring 104, the diameter D of the ball member C, the outside diameter N of the ball-engaging surface 136 of the seat ring or radially innermost end of its second surface 132, the outside diameter E of the disc spring 108, and the composite seat taper, dimension K in the relaxed state, which is typically determined from the relaxation of the seat ring 104 and disc spring 108 necessary to compensate for the expected seat wear. These dimensions are typically given parameters determined by the size of the valve which is desired and are determined according to conventional industry design standards. There is a unique spherical diameter D_s and a unique seat-to-ball radial gap g for each seat ring 104, 106 corresponding to each set of given valve parameters, D , d , N , E , and K , such that, when the seat ring 104, 106 is deformed upon assembly the ring-engaging surface 136, 138 conforms to the spherical surface of the ball member C.

In the preferred embodiment of the invention as shown in Figures 2 and 3, seat assemblies B are configured so that, at the preassembly reference position, frusto-conical disc springs 108, 110 and seat rings 104, 106 have their smaller diameter ends more closely spaced towards the ball member.

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With reference to the upstream seat assembly of Figure 2, the extent of the taper at the disc spring inner end 156 is predetermined to be a dimensional amount corresponding to the expected wear on the downstream seat ring 106 (Figure 3) over the cycle life of the valve. The first surface 128 of seat ring 104 tapers for mating association with the disc spring 108. The seat ring second surface 132 is disposed for close abutting engagement with the first or bearing surface 112 of the reinforcing ring 100. As best illustrated in Figure 4, and in the preassembly reference position, the third or ball-engaging surface 136 of seat ring 104 is arcuately contoured to have a spherical surface diameter slightly greater than the ball member diameter and axially offset therefrom. The radially innermost end of the ball-engaging surface 136 contacts the ball member C at an annular contact line 160 with a ball clearance gap thus being included at the radially outermost end of ball-engaging surface 136.

Operation

Referring to Figures 5 to 10, description will hereinafter be made with reference to operation of the ball valve and seat assembly. Figure 5 shows the valve in a fully assembled, valve-open position (no-load fluid pressure condition). In this position, the two seat assemblies B have been shifted from the

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unstressed condition shown in Figures 2 to 4 to a fully stressed condition. Sizing of the ball member C, seat assemblies B and shoulders 70, 72 is such as to provide this relationship at ball valve assembly or make-up.

More particularly, and with particular reference to Figures 5 to 7, the seat assemblies are disposed in an assembled condition such that opposed seat rings 104, 106 are compacted between the associated reinforcing rings 100, 102 and disc springs 108, 110 to compress the seat rings. The seat rings 104, 106 are further rotatably flexed, upon valve make-up, away from each other and towards associated shoulders 70, 72 generally about their outer peripheries and against disc springs 108, 110 in response to engagement between ball-engaging surfaces 136, 138 and a ball member C. This action compresses the associated disc springs 108, 110 towards a substantially flattened condition. In the position of Figures 6 and 7, the disc springs 108, 110 are preferably deflected through a travel distance (dimension K in Figure 4) predetermined by the reasonable expectancy of wear on the downstream seat ring 106 over the cycle life of the valve A.

The combined flexure of the seat rings 104, 106 and the disc springs 108, 110 provides a composite

spring seat assembly which has four principal advantages. The first is to provide a controlled preload of the seating surfaces against the ball member. Since some loading is necessary to maintain a seal, most prior art ball valves accomplish the
5 preload by crushing the seat rings between the ball member and the flange or shoulder, upon assembly, which, in effect, clamps the ball member against movement. With seat rings constructed of harder
10 plastic materials, this resulted in an unacceptably high operating torque. The controlled preload of the ball valve of the present invention, however, provides a seat surface applied to the ball member in response to a flexing action rather than a
15 crushing action and thereby offers a reasonable operating torque. Second, the seat ring and disc spring combinations operate to maintain a fluid seal at low seat pressures. Without spring loading of the
seat rings against the ball member, the seats are more
20 likely to leak at low pressures, i.e., when pressure loading against the ball member is less. Thirdly, a seal which compensates for seat wear and manufacturing tolerance is provided. As the seats wear
25 during valve operation, the composite seat ring assemblies relax against the ball member, thereby extending the leak-tight cycle life. Fourthly, the

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combined flexure resists the problem of ball member sag which may deform the seats by the forces of ball member weight.

5 If the only spring action were provided by the disc springs with no additional spring action being provided in the seat rings, the disc springs would have to overcome the elasticity of the seat rings in order to flex them against the ball member for wear compensation purposes.

10 Upon assembly, the disc spring and seat ring composite assemblies are largely flattened and flexed away from the support rings, i.e., seat ring second surfaces 132, 134 are flexed away from portions of the reinforcing ring first surface 112, 114.

15 Such action provides flexure reserve gaps 170, 172 to accommodate seat ring and disc spring relaxation while simultaneously maintaining a preload against ball member C.

20 Five principal advantages are provided by reinforcing rings 100, 102 in combination with the disc springs and seat rings of the ball valve illustrated.

25 The first is that they operate to reduce the seat stress on the downstream seat ring 106. Without the rings 100, 102, pressure behind the upstream seat ring 104 over an effective pressure area out to

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counterbore step 78 imparts a load on the downstream seat ring 106 and disc spring 110. With the inclusion of an upstream reinforcing ring 100, the effective pressure area of load translated to the downstream seat ring is reduced by the additional surface area of reinforcing ring first surface 112 which supports and reinforces upstream seat ring 104 against load pressures. The reduced load on the downstream seat ring 106 in a valve-closed position provides reduced seating area stress and, consequently, reduced seat wear to extend the leak-tight cycle life of the valve.

Secondary, the support ring 100 operates to support the upstream seat 104 against caving in under pressure. In prior systems lacking the upstream reinforcing ring 100, the upstream seat ring 104 can cave in under a long duration static load. Thirdly, during cycling as the ball-engaging surface 136 partially spans the ball opening while maintaining ball-sealing engagement, the upstream reinforcing ring 100 supports the seat ring 104 to prevent a blowing-in bulge. An upstream cave-in or blow-in bulge of the seat ring 104 immediately upsets the sealability of the downstream seat assembly.

Because of these aforementioned advantages, the upstream reinforcing ring 100 retains the seat ring 104 such that the allowable pressure on the seat ring can be increased over that allowable pressure

which is permissible when a reinforcing ring is not present.

Fourthly, the downstream support ring 102 also provides a positive stop against which the second surface 134 of seat ring 106 can form a flange-seat posterior seal. Most conventional seat ring materials readily crush upon assembly between the disc spring and the counterbore step or flange 80. Without the support ring 102, the seat 106 can too readily slip beyond this thin flange in the body of the valve. Particularly with hard plastics, such as polyethylene, a more positive stop is necessary to effect a posterior seal.

An additional, fifth advantage of reinforcing rings 100, 102 resides in retarding back creep of the seat rings 104, 106 and particularly, the downstream seat ring 106. The ball member C, at a valve-closed position with a load acting against the upstream seat assembly and without a downstream support ring 102, more readily displaces the downstream seat ring 106 towards the centre of the valve body, especially if the seat material is polytetrafluoroethylene or the like. Such displacement of creep acts to relieve the seating surface preload necessary to maintain an effective seal.

It should be noted that, even though the present

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invention encompasses a floating ball design, ball member C does not necessarily axially shift or float towards the downstream seat assembly to effect a downstream seat seal in a valve-closed and loaded condition. Since the downstream disc spring 110 is substantially flattened against shoulder 72 at assembly, the downstream seat ring 106 is substantially contained on all four sides and the force against the ball-engaging surface 138 is evenly distributed over the entirety of that surface. The loading force does not distort seat ring 106 to allow an axial shift of ball member C downstream to effect a principal seal at the downstream seat ring. Rather, a seat ring seal is achieved at both the upstream and downstream seat assemblies throughout the life of the ball valve. However, even after excessive wear to the downstream seat ring may have caused a break in the upstream seal, the ball valve will not fail as the downstream seat assembly will still effect a seal.

With reference to Figures 8, 9 and 10, a ball valve A' which has undergone considerable wear at the downstream seat assembly associated with end fitting 15' is illustrated in a valve-closed position under elevated fluid pressure conditions with the direction of fluid flow being designated by the arrow

adjacent the valve inlet. As is there shown, ball member C' is forced in response to fluid pressure to shift axially towards shoulder 72' and against the downstream seat ring assembly such that disc spring 110' remains substantially flattened. Due to the substantial wear of seat ring 106', gap 172' is expanded accordingly. Ball-engaging or third surface 138' of downstream seat ring 106' remains in sealing engagement with the surface of the ball member C' to provide a fluid-tight seal with equal stress distribution across the extent of surface 138'.

The upstream seat assembly associated with end fitting 14' is flexed towards the direction of movements of the ball member and is partially relaxed from its initially assembled condition to accommodate the wear of downstream seat ring 106' while maintaining a fluid-tight seal with the surface of the ball member C'. Disc spring 108' is relaxed from a substantially flattened condition to the extent required to compensate for the downstream seat ring wear. Upstream seat ring 104' has also relaxed to a similar extent to partially close flexure reserve gap 170' against the first surface 112' of the upstream reinforcing ring 100'. Because of the rotational flexing of seat ring 104', the seat ring second surface 132' is moved towards a contacting relationship with the first surface of reinforcing ring 100'. The

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ball-engaging surface 136' of upstream seat ring 104' remains in substantial sealing engagement with the surface of the ball member C'. The ball deforms a radially innermost portion of surface 136' so that at least that portion has a spherical conformation in mating engagement with the ball member. When the valve A' of Figure 8 is placed in the valve-open position, ball member C' will shift axially back towards end fitting 14' such that equal pressure will be exerted on the ball member by the disc springs 108', 110'.

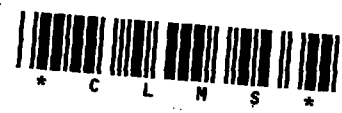
The ball valve A with seat assemblies B described in detail hereinabove is deemed to provide a substantial improvement over those arrangements previously known in the art. The present design permits the seat assemblies to be flexible and deflect in a predictable and controlled manner at initial assembly to facilitate compensation for seat wear, manufacturing tolerances and ball sag, and to provide a controlled preload for improved ease of operation and low pressure sealing. The ball-engaging surfaces of the seat ring assemblies are proportioned and configured for distorting to matable sealing engagement with the ball surface at assembly for providing equal stress distribution across the entire extent of surfaces 136, 138. The inclusion of support rings 100, 102 in combination with the improved seat rings

104, 106 and disc springs 108, 110 prevents the
problems of blowing-in bulge, seat cave-in, high
downstream seat stress and back creep and posterior
seat-to-shoulder leakage, and simultaneously does
5 not inhibit the operational advantage of the seat
ring and disc spring combination.

Other modifications not specifically shown
in the drawings may be readily incorporated into
seat ring assemblies B without in any way departing
10 from the invention. It may, for example, be desirable
to slightly modify the relative dimensional char-
acteristics between the seat rings, reinforcing
rings and disc springs to accommodate particular
operational requirements.

15 The invention has been described with reference
to a preferred embodiment. Obviously, modifications
and alterations will occur to others upon a reading
and understanding of this specification. It is our
intension to include all such modifications and
20 alterations insofar as they come within the scope
of the claims or the equivalents thereof.

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CLAIMS

1. A ball valve including a valve body having a generally cylindrical fluid flow passageway; a ball member disposed in said passageway and mounted for selective rotation between valve-open and closed positions with said ball member being shiftable generally axially in said passageway under fluid pressure conditions at least when said valve is in said closed position; a pair of annular seat rings disposed in said passageway on opposite sides of said ball member between said ball member and associated valve body shoulders; a pair of annular disc springs interposed between each seat ring and the associated shoulder for continuously urging said seat rings towards sealing engagement with said ball member; and a pair of rigid annular bearing surfaces positioned in said passageway on opposite sides of said ball member axially inwardly of said seat rings to provide support for the associated seat ring, each of said seat rings having a continuous annular surface facing said ball member, which surface has an arcuate contour in a relaxed condition defined by a radius greater than the radius of said ball member, such that, upon assembly of said ball valve, said seat ring is rotatably flexed about its outer periphery generally away from its associated bearing surface to direct said continuous



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annular surface of said seat ring into continuous sealing engagement with said ball member, said continuous surface having an arcuate contour in the assembled and flexed but fluid-pressure-free condition defined by a radius generally equivalent to said ball member radius.

2. A ball valve as claimed in claim 1, wherein a flexure reserve gap of said seat ring is formed between a portion of said rigid bearing surface and said seat ring.

3. A ball valve as claimed in claim 1 or 2, wherein, in use, said ball member is axially shifted in said passageway towards one of said valve body shoulders in response to fluid pressure acting thereon in said valve-closed position, said seat ring associated with the other of said valve body shoulders being relaxed towards said one shoulder for maintaining contact with said ball member.

4. A ball valve as claimed in claim 1, 2 or 3, wherein each seat ring is compacted between an associated one of said bearing surfaces and an associated one of said disc springs, said disc springs being compressed to a substantially flattened configuration, at least at initial valve assembly.

5. A ball valve including a valve body having a generally cylindrical fluid flow passageway; a ball



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member disposed in said passageway mounted for selective rotation between valve-opened and closed positions with said ball member being shiftable generally axially in said passageway under fluid pressure conditions at least when said valve is in said closed position; a pair of annular seat rings disposed in said passageway on opposite sides of said ball member between said ball member and associated valve body shoulders; a pair of annular disc springs interposed between each seat ring and the associated shoulder for continuously urging said seat rings towards sealing engagement with said ball member; and a pair of annular bearing surfaces positioned in said passageway on opposite sides of said ball member axially inwardly of said seat rings to provide support for the associated seat ring, each seat ring having an outer periphery in abutting engagement with a side wall of said fluid flow passageway, a first surface facing an associated one of said pair of annular disc springs, a second surface facing an associated one of said pair of annular bearing surfaces, and a third surface facing said ball member and having, at a pre-assembly reference position, a radius of curvature greater than the radius of curvature of said ball member and being spaced by a radial gap at a radially innermost end of said second surface from said ball member, and having, upon assembly and under fluid- pressure-free conditions, a radius of curvature substantially the same as said radius of curvature of said ball member and engaging said ball member in mating fashion over the



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extent of said third surface, said seat ring second surface further being spaced, upon assembly, from a portion of said ~~reinforcing ring~~ ^{annular bearing} first surface to provide a flexure reserve gap for said seat ring.

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6. A ball valve as claimed in claim 5, wherein said seat ring, in moving from said pre-assembly reference position towards said assembly position, is flexed about said outer periphery and urged into sealing engagement with said ball member at said third surface.

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7. A ball valve comprising; a body having a central passageway; a ball member having a fluid flow opening therethrough, said ball member being positioned in said passageway and mounted for selective rotation between valve-open and closed positions to control fluid flow through said valve; a pair of radially inward extending continuous shoulders in said passageway disposed circumferentially thereof on opposite sides of said ball member; a pair of radially inwardly extending counterbores in said passageway disposed on opposite sides of said ball member, each of said counterbores having an inner end wall and being circumferentially continuous; and a pair of composite seat member assemblies positioned axially in said passageway on opposite sides of said ball member for fluid-sealing engagement with said ball member, each of said pair

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comprising: a reinforcing ring having a central opening,
a first surface facing an associated one of said
shoulders, a second surface abutting an associated one
of said counterbore end walls and a third surface
5 generally facing said ball member in a spaced
relationship therefrom; a seat ring adapted for elastic
flexure generally towards and away from said reinforcing
ring and including a central opening, a first surface
generally facing said associated one of said shoulders,
10 a second surface abutting said reinforcing ring first
surface, and a third surface facing said ball member for
sealing engagement with said ball member, said seat ring
third surface being contoured and having a radius of
curvature in a relaxed unassembled condition greater
15 than the radius of said ball member; and a disc spring
having a central opening and a generally frusto-conical
configuration in an unstressed condition interposed
between said seat ring and said associated one of said
-shoulders; said ball member and pair of composite seat
20 member assemblies being sized so that, when assembled
axially between said shoulders, each seat ring is
flexed away from an associated reinforcing ring and
stressed to have a radius of curvature at said seat ring
third surface generally equivalent to the radius of said
25 ball member and is disposed in sealing engagement with
said ball member over the extent of said seat ring third



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even under fluid-pressure-free conditions
surface, each disc spring being stressed towards a flattened condition for continuously urging the third surface of the associated seat ring towards engagement with said ball member.

5 8. A ball valve as claimed in claim 7, wherein said ball member and pair of composite seat member assemblies are sized so that, when assembled from a preassembly reference position, the seat ring and disc spring of each assembly are flexed and stressed to have
10 an axial displacement at said central openings thereof equal to an axial spacing of a radially innermost end of said seat ring second surface from said ball member plus an axial displacement of said disc spring at an equiradial position to said axial spacing.

15 9. A ball valve as claimed in claim 8, wherein said disc spring axial displacement at the central opening thereof comprises a displacement of said disc spring towards said associated one of said shoulders.

20 10. A ball valve as claimed in claim 8 or 9, wherein the centre of curvature of said third surface in the relaxed state is offset from the centre of said ball member.

25 11. A ball valve as claimed in claim 10, wherein said third surface centre of curvature is axially offset from said centre of said ball member.



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12. A ball valve as claimed in any of claims 7 to 11, wherein the central openings in each disc spring and associated seat ring are substantially equal in size and smaller than the central opening in the associated reinforcing ring.

13. A ball valve as claimed in any of claims 7 to 12, wherein said ball member is axially movable in said passageway when said valve is in said closed position to accommodate shifting of the ball member towards one of said shoulders in response to fluid pressure, the seat ring associated with the other of said shoulders being flexed towards the direction of movement of said ball member in cooperation with the influence of its associated disc spring in order that the third surface thereof will be urged towards continuous contact with said ball member.

14. A ball valve as claimed in claim 13, wherein flexure of said seat rings comprises a rotational type flexure.

15. A ball valve as claimed in any of claims 7 to 14, wherein said reinforcing ring first surfaces are configured to provide a rigid bearing surface for said seat rings.

16. A ball valve as claimed in claim 15, wherein each reinforcing ring is spaced from said ball member to an extent which precludes abutment with said ball



member and which inhibits distortion and displacement of the associated seat ring between the reinforcing ring third surface and said ball member.

17. A ball valve as claimed in any of claims 1 to 6, wherein each seat ring is compacted between the associated rigid bearing surface and disc spring to compress said seat ring.

18. A ball valve as claimed in any of claims 1 to 17, wherein said body includes a removable end fitting which defines one of said shoulders.

19. A ball valve as claimed in any of claims 1 to 18, wherein said seat rings each include an axially extending circumferential flange at the first surface thereof for locating the associated disc spring relative thereto.

20. A ball valve as claimed in any of claims 1 to 19, wherein said seat rings are constructed from a resilient material.

21. A ball valve constructed and adapted to operate substantially as herein described with reference to and as illustrated in the accompanying drawings.
