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PUMPING APPARATUS FOR ABRASIVE SLURRY

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Fig. 1.

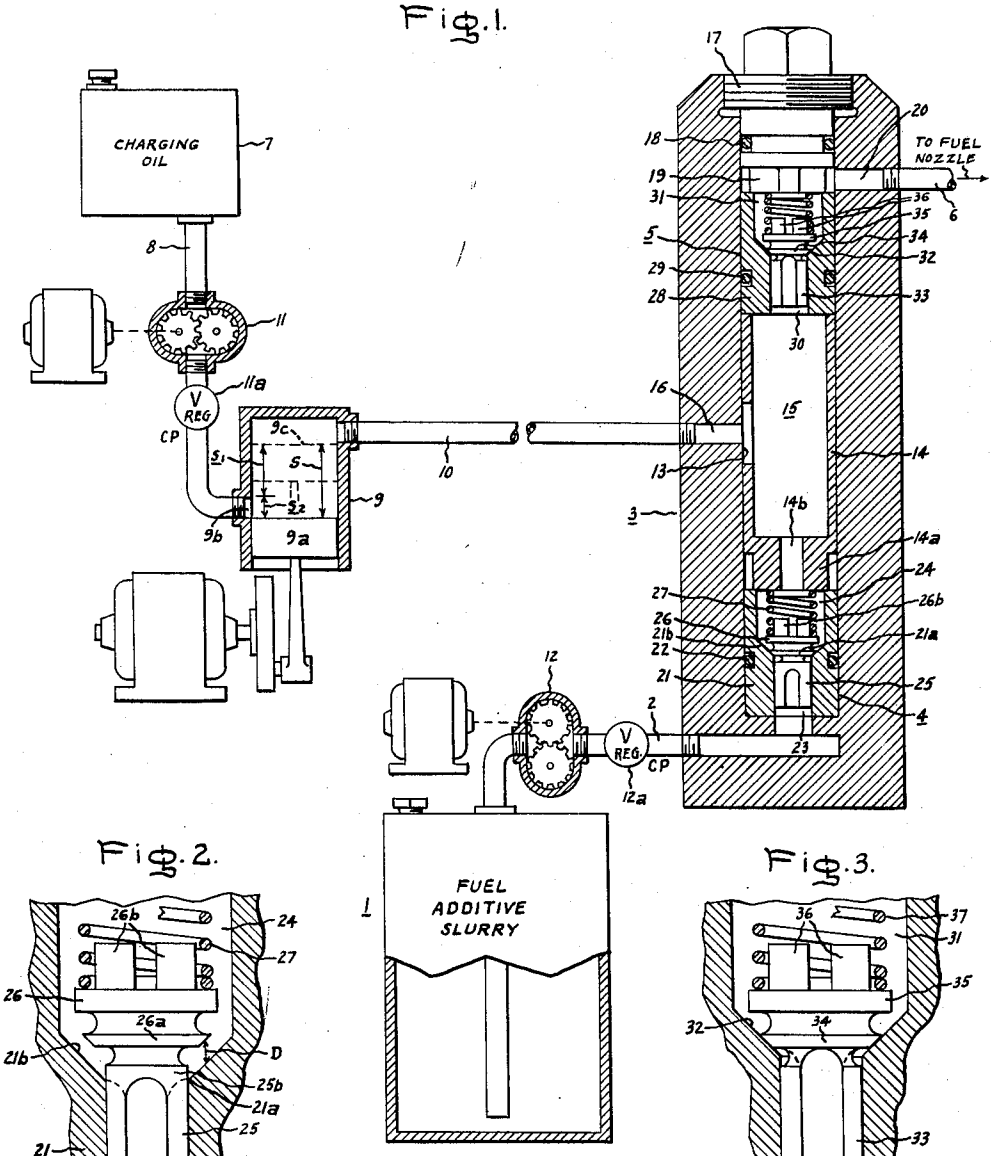


Fig. 2.

Fig. 3.

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PUMPING APPARATUS FOR ABRASIVE SLURRY

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4 Claims. (Cl. 103—6)

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This invention relates to apparatus for pumping a liquid carrying entrained abrasive particles, particularly to a system for supplying a fuel additive slurry to the nozzles of a fluid fuel burning system, as used for instance in gas turbine powerplants.

In recent years, as a greater proportion of the available fuel oil supply has been obtained from certain oil fields, increasing trouble has been encountered with a characteristic type of corrosion of combustion chamber parts and other metals subjected to high temperatures and combustion products, such as the tubes in steam power boilers. This has come to be popularly referred to as "vanadium corrosion," since it appears to occur principally with petroleum products made from crudes containing a high proportion of vanadium compounds. While the exact mechanism of this corrosion has not yet been fully ascertained, research has shown that it can be reduced or, at least in some cases, entirely prevented by supplying to the combustion chamber certain compounds which appear to have an influence on the slagging properties of the insoluble ash resulting from combustion of the oil. Unfortunately, it is not economically feasible to use additives that are soluble in the fuel oil. Therefore, various ways have been tried for furnishing insoluble substances to the combustion process, as for instance by feeding a solid wire or rod directly into the flame in the combustion space. Such expedients involve cumbersome and often troublesome mechanical equipment, and make it difficult to properly proportion the additive to the rate of fuel flow.

The most convenient way to properly supply these additive compounds in correct proportion has appeared to be by providing a liquid "slurry," consisting of a suitable liquid vehicle in which the solid additive compounds are suspended in finely divided, almost colloidal, form. But no matter how small these insoluble particles are divided, they still have an extremely abrasive effect on any moving parts contacted in the pump used to deliver the slurry into the fuel stream. Since the fuel nozzle supply pressure may be as high as 600 to 800 p. s. i., the pump required to force the slurry into the fuel line must have quite close clearances, meaning that any abrasive particles in the liquid passing through the pump will have an extremely serious effect on the operation and life of the pump.

Accordingly, an object of the present invention is to provide improved apparatus for delivering under pressure an abrasive particle containing

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slurry to the fuel stream of a liquid fuel combustion system, so arranged that abrasive particles from the slurry cannot enter the high pressure pump.

Another object is to provide a slurry pumping system of the type described capable of operating at high speeds, so the pumping equipment required will be comparatively small and therefore adaptable to compact portable gas turbine powerplants, such as those used in locomotives.

A still further object is to provide a slurry pumping system capable of providing very accurate control of the proportion of slurry relative to the rate of fuel flow.

Other objects and advantages will become apparent from the following description taken in connection with the accompanying drawings, in which Fig. 1 is a diagrammatic representation of a slurry pumping system in accordance with the invention, Fig. 2 is an enlarged detail view of a special spring-loaded check valve through which the slurry is received, and Fig. 3 is an enlarged detail view of a somewhat similar spring-loaded discharge valve.

Generally, the invention is practiced by providing a pair of specially arranged check-valves in the slurry supply conduit and causing these valves to be operated in proper sequence and the slurry to be forced into the discharge line by a high pressure pump acting in a separate circuit which contains only a light hydrocarbon oil, such as diesel fuel oil.

Referring now more particularly to Fig. 1, this improved pumping system is shown as comprising a suitable reservoir 1 for the fuel additive slurry, connected by an inlet conduit 2 to one end of a check-valve casing 3. This check-valve unit includes a spring-loaded suction valve indicated generally at 4, and a quite similar spring-loaded discharge valve shown generally at 5. The slurry is discharged at high pressure through a conduit 6 at the top of casing 3.

The "charging oil system" comprises a reservoir 7 for the charging oil, which may conveniently be diesel fuel, connected by an inlet conduit 8 to the suction port 9b of a high pressure "valve actuator pump" indicated generally at 9, the discharge port of which communicates by way of conduit 10 with the midportion of casing 3. The charging oil supply conduit 8 is provided with a "charge oil pump" 11, while the slurry supply conduit 2 is provided with a "slurry charge pump" 12, the functions of which will be seen more particularly hereinafter.

Looking now more closely at the check valve

unit 3, it will be seen that the casing has a central longitudinal bore 13 in the bottom of which is the suction valve unit 4, separated by a tubular spacer member 14 from the discharge valve unit 5. It is to be noted that the spacer 14 defines a central chamber 15 of significant volume, which communicates with the charging oil inlet port 16. The upper end of bore 13 is closed by a suitable threaded plug 17, which may be provided with a fluid seal in the form of an O-ring 18. The lower end of plug 17 has a plurality of radially extending arms 19, the spaces between which provide free communication of liquid to the discharge port 20.

Referring now more specifically to the construction of the suction valve 4, it will be seen in Fig. 1 that this valve comprises a generally cylindrical housing member 21, the external surface of which is provided with a fluid seal in the form of an O-ring 22, and the upper end of which is engaged by the lower end portion 14a of the spacer cylinder 14. Housing 21 defines a central bore having a reduced diameter portion 23 and an enlarged portion forming a cylindrical chamber 24. The movable flow control member comprises a plunger member having a lower piston portion 25 and a larger diameter upper end portion 26 adapted to receive the free end of a suitable biasing coil spring 27. As will be clear in Fig. 1, the upper end of coil spring 27 seats against the end portion 14a of spacer 14.

Further details of the suction valve 4 may be better seen by reference to the enlarged detail view of Fig. 2. It will be observed that, whereas Fig. 1 shows the suction valve plunger in its lowermost or closed position, Fig. 2 shows it elevated against the bias of spring 27, at what will be referred to herein as the "cutoff" point. The significance of this will be seen hereinafter.

It will also be seen in Fig. 2 that the piston portion 25 of the check-valve plunger is provided with several circumferentially spaced grooves or "flutes" 25a, which do not extend the full length of the piston 25. With this arrangement, it will be apparent that piston 25 establishes communication between the inlet chamber 23 and the outlet chamber 24 whenever the plunger is above the cutoff position shown in Fig. 2, and that communication between these chambers is interrupted whenever the plunger is below this cutoff position. It will be seen that this cutoff point is the position at which the extreme upper ends of the grooves 25a become aligned with the annular shoulder identified 21a. It will also be seen that only the extreme upper portion of piston 25, identified 25b, is truly a "piston" in the sense that it will completely occupy the bore 23 so as to effect a fluid displacement when it moves in the bore.

The lowermost position of the plunger is determined by engagement of the beveled annular seat member 26a with the cooperating conical surface 21b of check valve housing 21. It will now be observed that the piston portion 25b has an "effective displacement" corresponding to a stroke of a length identified by the dimension arrow D in Fig. 2.

The extreme upper end of plunger portion 26 has a number of circumferentially spaced upwardly projecting bosses 26b which serve the dual function of centering the coil spring 27 and forming a stop defining the upper limiting position of the plunger member. It will be seen in Fig. 1 that upward movement of the plunger is limited by engagement of these abutment members 26b with the adjacent end portion 14a of the spacer cylinder. It will also be observed in Fig. 1 that, in this upper limiting position, the spaces between the abutment members 26b continue to provide free communication between the chamber 24 and the passage 14b through the spacer cylinder end portion 14a.

The discharge check valve assembly 5 is quite similar in construction to the lower unit 4, comprising the housing 28 having an O-ring seal 29 and defining a lower cylinder portion 30 and an enlarged upper chamber 31, with a conical seat portion 32 therebetween. The upper end of housing 28 is engaged by the abutting end members 19 of plug 17. The discharge valve plunger comprises the lower piston portion 33, the beveled seat member 34 adapted to sealingly engage the housing seat portion 32, and an upper end portion 35 having a plurality of axially extending abutment members 36, the structure and function of which is analogous to that described in connection with the abutments 26b of the suction valve plunger. It will be apparent in Fig. 1 that the abutments 36 engage the radially projecting arm portions 19 of plug 17 to determine the uppermost position of the plunger, so that free communication is at all times provided for liquid to flow from chamber 31 to the discharge port 20. The coil spring 37 is centered by the abutment members 36.

Referring still to Figs. 2 and 3, attention is particularly directed to the sole distinguishing feature between the discharge valve 5 and the suction valve 4. This lies in the fact that the grooves 33a in the piston portion 33 do extend the full length of the piston, all the way to the annular seat portion 34. This means that there is no true "piston" portion corresponding to that identified 25b in Fig. 2, so that the inlet chamber portion 30 is in free communication with the outlet chamber 31 whenever the plunger is elevated above the position shown in Fig. 3, that is, whenever the seat member 34 moves away from the conical seat 32. Thus the discharge valve 5 is simply a check-valve, having no piston portion corresponding to that identified 25b in Fig. 2. The significance of this very important distinction between the suction and discharge valves will be seen from the description of the method of operation hereinafter.

The function of the charge pump 12 is to insure that the slurry is supplied to check valve unit 3 at a positive pressure, in order to prevent evaporation of any readily volatile components of the slurry, which might otherwise cause "cavitation" with consequent improper actuation of the check-valves. Cooperating with the slurry charging pump 12 is a constant discharge pressure regulating valve indicated generally at 12a. This may, for instance, be adjusted so the pressure of the slurry supplied to the check-valve unit 3 is maintained constant at a value on the order of 20 pounds per square inch. It will be apparent that pump 12 may be dispensed with in applications where the slurry reservoir 1 can be mounted a sufficient distance above valve 21 that the hydraulic head provides the positive pressure desired in inlet chamber 23.

While the slurry charge pump 12 has been illustrated diagrammatically as being of the well known gear type, it may preferably be a centrifugal or rubber impeller type pump having moving parts relatively insensitive to the action of the abrasive particles in the slurry. Because of the comparatively low discharge pressure of pump 12,

it may be feasible to use a gear type pump if the clearances between the rotors and housing are made sufficiently great, relative to the maximum size of the abrasive particles in the slurry so that excessive wear will not result.

The diesel oil from reservoir 7 is supplied to the high pressure pump 9 by the charging oil pump 11, at a pressure which may be maintained constant at a value on the order of 40 p. s. i. by a constant pressure regulating valve 11a. It is important to note that the discharge pressure of the charging oil pump 11 is at least equal to, and preferably somewhat greater than, the discharge pressure of the slurry charge pump 12. This means that the pressure of the charging oil in conduit 10 will always be sufficiently great that there will be no tendency for the lower pressure slurry to enter conduit 10 and so reach the high pressure pump 9.

The pump 9 serves the dual function of generating the high pressure required to force the slurry through the discharge conduit 6 against the high pressure obtaining in the fuel supply conduit (not shown), and at the same time serving as a "hydraulic actuator" for opening and closing the check-valves 4, 5 in proper sequence. While not material to an understanding of the present invention, it may be noted that pump 9 may advantageously be a modified multiple piston pump of the type used as fuel oil injection pumps in reciprocating diesel engines. Because its most significant function is actuating the check-valves 4, 5, pump 9 will be referred to hereinafter as the "valve actuator pump."

It is to be observed in Fig. 1 that the valve actuator pump piston 9a does not uncover the charging oil inlet port 9b until the piston gets near the bottom of its stroke. The uppermost position of piston 9a is shown in dotted lines at 9c, and the total displacement of the piston is identified by the dimension arrow S.

The operation of the system may now be outlined as follows.

First assume that the valve actuator pump piston 9a is at the bottom of its stroke, as shown in Fig. 1. As the piston moves upwardly, it immediately blocks off the charging oil inlet port 9b, and during the remainder of the stroke, identified s_1, it builds up pressure in the conduit 10 and chamber 15 until the plunger of the discharge valve 5 opens against the bias of spring 37 and the high pressure existing in the conduit 6 and delivers liquid to conduit 6. Meanwhile, of course, the biasing spring 27 and the high pressure in chamber 15 hold the suction valve plunger in the closed condition shown in Fig. 1.

Now when piston 9a begins to descend, the pressure in chamber 15 drops until spring 37 causes the discharge valve 5 to return to the closed position shown in Fig. 1. Continued downward movement of piston 9a, through the remainder of stroke portion s_1, causes the pressure in chamber 15 to drop until the suction valve plunger rises against the bias of spring 27, as aided by the slurry charging pressure produced by pump 12. As piston 9a continues down, slurry flows past the suction valve 4 into the central chamber 15 of the check valve unit 3. As pump piston 9a approaches bottom dead center, it uncovers the charging oil inlet port 9b, during the stroke portion identified s_2. When this happens, the charging oil, being at a pressure on the order of 40 pounds per square inch as compared with the slurry charging pressure of only 20 pounds per square inch, enters the pump 9, raises the pressure in conduit 10 and check valve chamber 15, and helps move the suction valve plunger downward to its seated position, as shown in Fig. 1. Actually, of course, the suction valve plunger moves downwardly under the influence of its own spring 27 as soon as the rate of downward movement of piston 9a becomes insufficient to hold the suction valve plunger open against the bias of spring 27.

In other words, the suction valve plunger moves downwardly from the position shown in Fig. 2 to the closed position shown in Fig. 1 concurrently with the uncovering of the charging oil inlet port 9b by the piston 9a. During this downward movement, the suction piston portion 25b moves through a stroke of the length identified by dimension arrow D in Fig. 2, with the result that a quantity of charging oil is drawn into pump 9 corresponding in volume to the displacement D. Thus the piston portion 25b of the suction valve 4 is in effect a metering device for drawing into the valve actuator pump 9 a measured quantity of charging oil from the reservoir 7. If now the valve actuator piston 9a again moves upwardly, the charging oil in the pump again causes the pressure in the check-valve chamber 15 to build up and open the discharge valve 5.

It will be seen that this process results in a measured quantity of charging oil being drawn into the check-valve chamber 15 on each stroke of the valve actuating pump 9. The result is there is a small but substantially continuous net average flow of charging oil from reservoir 7 through the conduit 10 and into the check-valve unit 3. This comparatively small flow prevents any abrasive containing slurry from moving upstream into the pump 9. Thus the high pressure pump 9 always contains only clean charging oil.

When the system is first started, it is of course necessary to make sure that the entire charging oil circuit and the check valve unit are full of clean charging oil. This could, for instance, be accomplished by running the charging oil pump 11 and valve actuator pump 9 with the gear pump 12 stationary, so there will be no slurry charging pressure tending to open suction valve 4. Or, suitable shut-off valves could be provided in conduits 2, 10 so that slurry could not get into the check-valve unit during the initial charging process, and so that any abrasive particles remaining in the check-valve unit could not diffuse into the conduit 10 and pump 9 during any shutdown periods.

As has been noted above, the charging oil pressure of 40 p. s. i. is substantially higher than the slurry charging pressure of 20 p. s. i. It should be observed, however, that the system will be operative if these pressures are equal. This is because the biasing spring 27 is adequate to move the suction valve plunger to closed position so as to draw in the charging oil. It is believed of some advantage, however, to have the charging oil pressure somewhat above that of the slurry pressure, in order to more positively close the suction valve and insure against slurry entering the conduit 10.

As also indicated above, the function of the slurry charge pump 12 is merely to supply the slurry to the check-valve unit at a sufficiently high pressure as to prevent cavitation due to vaporization of liquid components of the slurry. It follows that, if the slurry has no substantial tendency to vaporize, or if the slurry reservoir 1 be located at a level somewhat higher than that of

the check-valve unit 3, the slurry pump 12 may be omitted.

As will also be appreciated from the above description of the operation, the design of the suction valve plunger, specifically the length of the stroke D from the cutoff point to the bottom of the plunger stroke, determines the volume of charging oil admitted on each stroke of the valve actuating piston 9a. The rate of flow of charging oil through conduit 10 required to completely prevent the entrance of slurry must of course be determined somewhat by experience. In the application for which the invention was developed, it has been found that introducing charging oil in approximately the proportion of one part to four parts of slurry is satisfactory.

It has also been found important that the volume of the connecting conduit 10 be of a certain minimum value. Specifically, it is found that this volume should be on the order of eight times the displacement of the valve actuating pump piston 9a. This volume may be somewhat greater than the value stated, but if it is made too great then compressibility effects due to foam and entrained gases in the charging oil may make the action of the check-valves erratic. The interconnecting conduit 10 in Fig. 1 is shown broken to indicate that its length is made sufficient that the volume represented by the conduit will be as stated.

The invention provides a slurry pumping system of comparative simplicity, yet capable of extremely high capacity with equipment of reasonable size. Specifically, a pumping system in accordance with the invention is capable of speeds on the order of 500 cycles per minute, as compared with less than 100 per minute with the arrangements known to the prior art. This means that the pumping components can be very much smaller. This is important since, in a fuel burning system such as that in a gas turbine powerplant, the slurry metering system is merely an accessory superimposed on the powerplant for the purpose of avoiding the service difficulties resulting from vanadium corrosion. Such added accessories are particularly irksome if they entail substantial investments in extra equipment occupying valuable space around the powerplant.

While the system has been described in its simplest form, it will be appreciated that in an actual fuel burning system there will be two high pressure pump cylinders 9, each having its associated check-valve unit 3 discharging into a common fuel nozzle supply line 6. By arranging the high pressure pump pistons 9a so that they operate 180° out of phase with each other, and by arranging that the s_1 portion of the upward stroke requires 180° or half of the drive cam travel a substantially smooth continuous discharge into the conduit 6 can be effected. In a gas turbine powerplant having perhaps six combustors, there will accordingly be twelve pump cylinders 9 and twelve check-valve units like that shown at 3. It is for this reason that multiple cylinder diesel engine fuel injection pumps are particularly appropriate for this service, since conventional diesel pumps can be readily modified to serve as the valve actuator pump for a system incorporating the invention.

It will be apparent to those skilled in the art that many other changes and substitution of equivalents may be made. In addition to elimination of the slurry charge pump 12, it may also be noted that the charging oil pump 11 may in some cases be eliminated simply by mounting the charging oil reservoir at a sufficient elevation above the pump 9. In a gas turbine powerplant, the pumps 9, 11, 12 could be driven by suitable gearing from the powerplant shaft rather than by the electric motors shown in Fig. 1. And the precise mechanical design of the check-valve units may take many forms.

It is, of course, intended to cover by the appended claims all such modifications as fall within the true spirit and scope of the invention.

What I claim as new and desire to secure by Letters Patent of the United States is:

1. In pumping apparatus for an abrasive or corrosive liquid, the combination of a multiple check-valve assembly, first suction conduit means for supplying the liquid to one end of the check-valve assembly, second discharge conduit means communicating with the other end of the check-valve assembly, and a third charging liquid conduit communicating with the mid-portion of the check-valve assembly, said check-valve assembly including housing walls defining a central chamber communicating with said charging conduit, a suction valve disposed at the end of the housing to which the first conduit is connected and having a first movable flow control member biased to closed position, the check-valve housing having also a discharge valve disposed in the end of the housing to which the second conduit is connected and having a flow control member biased to closed position, positive displacement means in the housing adapted to effect a predetermined increase in the volume of said central chamber when the pressure therein rises to a preselected value, and valve actuator pump means adapted to force charging liquid into said third conduit at a pressure varying cyclically from a high value sufficient to open the discharge valve and effect movement of said displacement means for transiently effecting enlargement of the volume of said central chamber, to a low value sufficient to effect opening of the suction valve to admit liquid to the central chamber.

2. In pumping apparatus for an abrasive or corrosive liquid, the combination of a multiple check-valve assembly, first suction conduit means for supplying the liquid to one end of the check-valve assembly, second discharge conduit means communicating with the other end of the check-valve assembly, and a third charging liquid conduit communicating with the mid-portion of the check-valve assembly, said check-valve assembly including a housing defining a central chamber communicating with said charging conduit, a suction valve disposed at the end of the housing to which the first conduit is connected and having a movable flow control piston member and spring means biasing the flow control member to closed position, the check-valve assembly having also a discharge valve disposed in the end of the housing to which the second conduit is connected and having a flow control member and means biasing it to closed position, said suction valve piston member being slidably disposed in a cylindrical bore portion of the housing and cooperating therewith to define a cut-off point in the return stroke of the piston dividing the total stroke into a portion in which the first conduit is placed in communication with said central chamber and a second portion in which such communication is interrupted and the piston moves through a positive displacement causing a predetermined quantity of charging liquid to be drawn into the central chamber from said third conduit, and valve actuator pump means adapted to force charging liquid through said

third conduit at a pressure varying cyclically from a high value sufficient to open the discharge valve against the bias of its spring and the back pressure in the second conduit and effect flow of liquid into the discharge conduit, to a low value sufficient to effect opening of the suction valve to admit liquid to said central chamber, and means for supplying charging liquid to the valve actuator pump at a pressure at least equal to the pressure at which liquid is supplied to the suction valve.

3. Pumping apparatus in accordance with claim 2 in which the means for supplying the liquid to the suction check-valve includes pump means adapted to maintain the supply pressure

to the check-valve at a positive pressure to prevent cavitation due to evaporation of volatile constituents of the liquid, and the charging liquid supply means includes a pump for supplying charging liquid to the valve actuator pump at a pressure not less than the suction valve supply pressure.

4. Pumping apparatus in accordance with claim 2 in which the volume of the charging liquid supply conduit between the valve actuator pump and check-valve assembly is on the order of eight times the displacement of the valve actuator pump.

No references cited.