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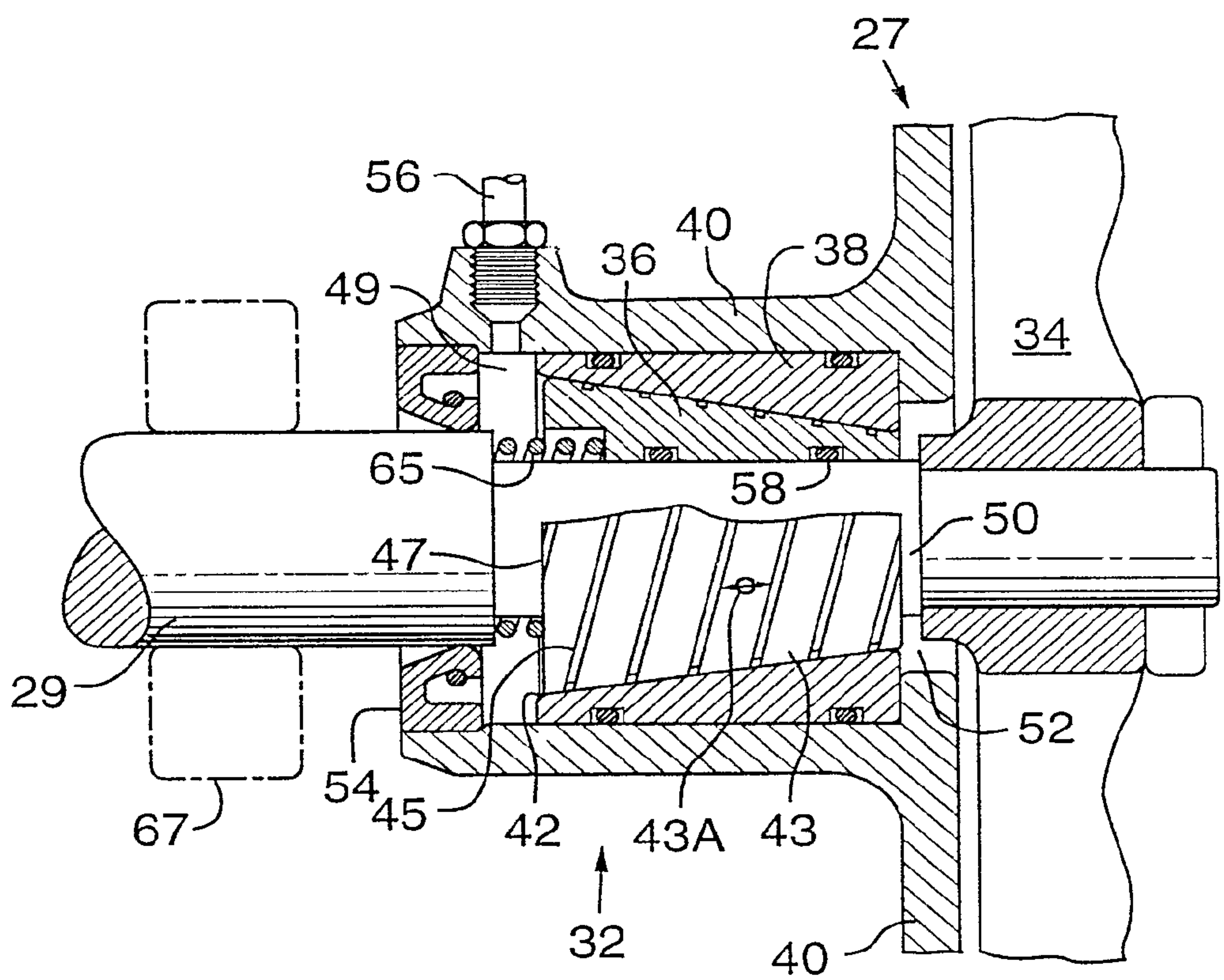

(57) Abrégé/Abstract:
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(54) Title: SEAL/BEARING ASSEMBLY

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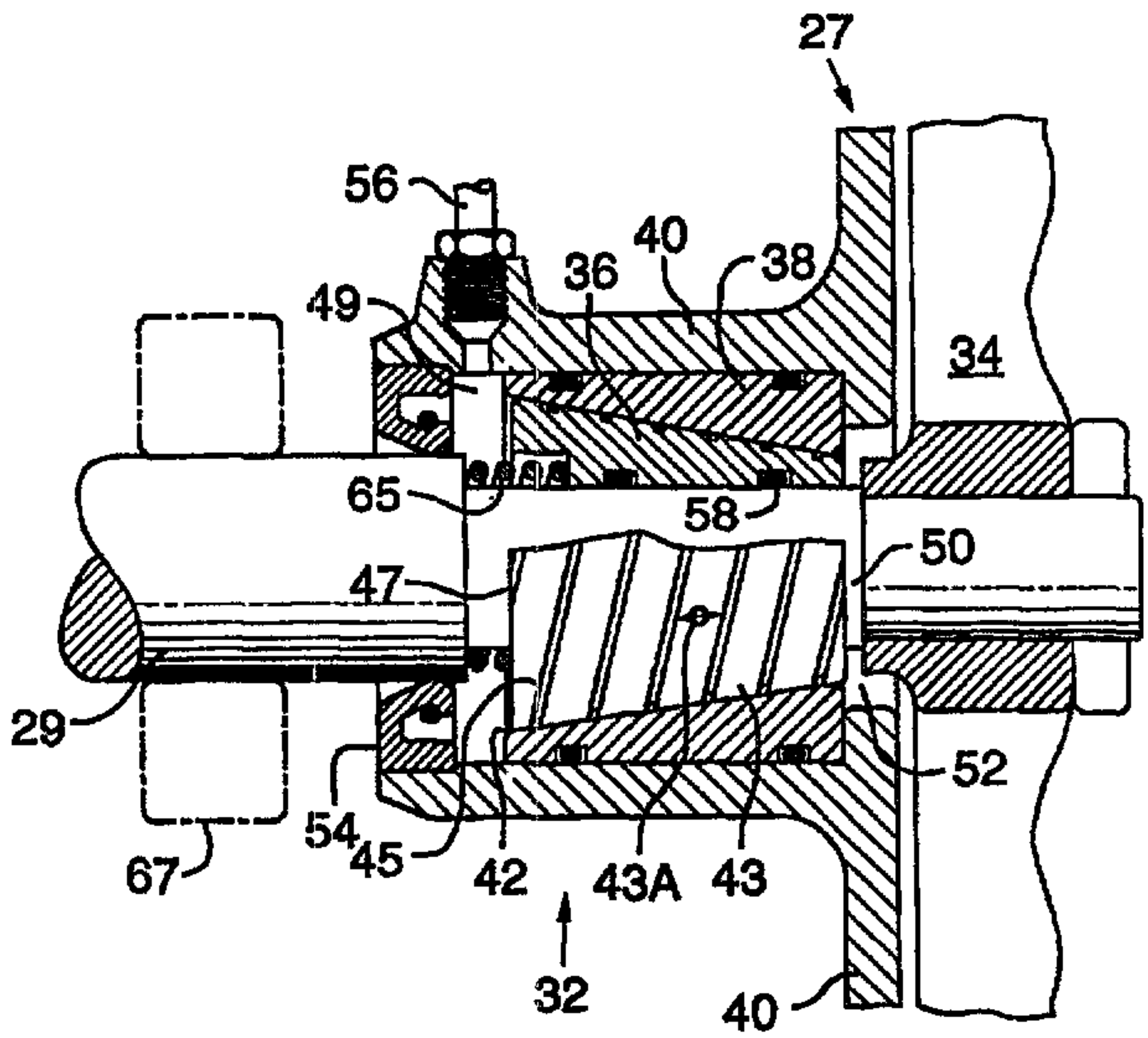
For fitment into the stuffing box of a centrifugal pump. A spiral-groove is cut on the outer surface of the rotor of a pair of tapered bearing surfaces. The groove is 0.02 mm deep, 2 mm wide, and its circumferential length is 50 cm. Barrier-liquid is fed to an entry-mouth of the spiral-groove, and groove generates a pressure in the stuffing box high enough to overcome process pressure. The barrier-liquid may be water. The interface between the bearing surfaces is sealed from the process-fluid and from the outside.



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This invention relates to seals and bearings for rotary shafts, and is particularly suitable for use in motor-driven centrifugal pumps.

Background of the Invention

The requirement is increasing for seals to be so constructed that the seal cannot fail in such a way as to release toxic fluids into the environment. Apart from that new requirement, there is the common need in seal design that the seal last for as long a service life as possible without failure; that, when a seal starts to leak, the seal resists leakage flow and does not burst wide open; that the seal be inexpensive to repair - as to the seal itself, the amount of dismantling needed to expose the seal, and the time the equipment is out of service. Versions of the seal-bearing assembly of the invention address these various needs, as will be described.

General Features of the Invention

An aim of the invention is to provide a means for generating a large pressure in the stuffing box of a centrifugal pump, i.e. a larger pressure than the pressure of the process-fluid.

It is an aim of the invention to achieve this high pressure without resorting to the use of viscous and lubricious barrier-liquids, but to achieve a high pressure even when using water as the barrier-liquid.

It is an aim of the invention to provide a means for controlling the pressure in the stuffing box, and for controlling that pressure in relation to the process-fluid pressure, and in relation to the outside environment pressure.

In one aspect of the invention there is provided apparatus for a rotating shaft, comprising a stator component, and a rotor component adapted for rotation about the axis of rotation of the shaft, wherein the rotor and the stator components are formed with complementary tapered bearing-surfaces coaxially disposed about the axis; the bearing-surfaces of the rotor and stator components are so arranged as to sweep each other in a hydrodynamic-bearing relationship, over an area termed the bearing area, upon rotation of the rotor; one of the bearing-surfaces is formed with a continuous spiral groove, which extends in a spiral configuration along and around the bearing-surface, over the bearing area; the spiral-groove comprises

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several turns extending over the bearing-surface, the arrangement thereof being such as to leave lands between adjacent turns of the spiral-groove; the apparatus is so structured that the spiral-groove has an entry-mouth and an exit-mouth; the apparatus is so structured as to define an entry chamber and an exit chamber, being chambers which are in fluid-conveying-communication with the entry mouth and the exit mouth respectively; the apparatus includes a means for receiving a barrier-liquid from a source of barrier-liquid, and for conveying the barrier-liquid to the entry chamber; the apparatus is so structured that the fit of the bearing-surfaces ensures the establishment and the continuance, during rotation, of a hydrodynamic film between the bearing surfaces.

In another aspect of the invention there is provided seal assembly apparatus for a rotating shaft having an impeller and mounted for rotation within a housing comprising a stator component and a rotor component adapted for rotation about the axis of rotation of the shaft, characterized in that the apparatus includes the following features, in combination: the stator and rotor components are annularly and coaxially disposed about the shaft axially adjacent to the impeller and have complementary axially tapered surfaces for fitting together in a male-female configuration, the rotor being secured for rotation with the shaft and the stator being secured to the housing; one of the complementary surfaces is formed with a continuous spiral groove which extends in a spiral configuration around the surface, the spiral groove having open entry and exit mouths at opposite axial ends, and the complementary surfaces are configured for positive pumping a barrier fluid for sealing toward the impeller fully across the surface upon rotation of the rotor; and a means for receiving a barrier fluid from a source of barrier fluid, and for reliably conveying the barrier fluid to the open entry mouth of the spiral groove.

The invention provides a combined seal and bearing assembly apparatus, comprising a stator and a rotor adapted for rotation about an axis. The apparatus may be installed in place of the stuffing box of a centrifugal pump.

The rotor and the stator components are formed with complementary bearing-surfaces, which are so arranged as to sweep each other in a hydrodynamic-bearing relationship, over an area termed the bearing

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fluid is repelled away from the seal, and the more the seal contact force is relieved. Examples of such structures are shown in US-3,746,350 (1973, Mayer+) and in US-4,243,230 (1981, Baker+).

Shallow grooves in the shape of scrolls have been provided on seal-faces, and serve to move liquid present at the seal-face in a desired direction. Examples are shown in US-4,290,611 (1981, Sedy), US-5,249,812 (1993, Volden+) and US-Re.34,319 (1993, Boutin+).

The use of very small radial clearances between rotating components, to promote flow in a desired manner, is shown in US-5,372,730 (1994, Warner+).

DETAILED DESCRIPTION OF PREFERRED EMBODIMENTS

By way of further explanation of the invention, exemplary embodiments of the invention will now be described with reference to the accompanying drawings, in which:

Fig 1 is a side-elevation of a pump-motor installation;
Fig 2 is a cross-sectioned side-elevation of an apparatus suitable for use in the installation of Fig 1;
Fig 3 is a view similar to Fig 2 of another apparatus;
Fig 4 is a view of a component suitable for use in the apparatus;
Fig 5 is a view of another apparatus suitable for use in the installation;
Fig 6 is a view of another suitable apparatus;
Fig 7 is a circuit diagram;
Fig 8 is a view of another suitable apparatus;
Fig 9 is a view of another suitable apparatus;
Fig 10 is a view of another suitable apparatus;
Fig 11 is a view of an apparatus, shown installed with other structures;
Fig 12 is a view of another type of apparatus;
Fig 13 is a view of another component suitable for use in the apparatus.

The apparatuses shown in the accompanying drawings and described below are examples which embody the invention. It should be noted that the scope of the invention is defined by the accompanying claims, and not necessarily by specific features of exemplary embodiments.

Fig 1 shows a pump-motor installation. An electric motor 20 is mounted on a base frame 23, on which are mounted also a bearing box 25 and a centrifugal pump 27. The armature of the motor is connected to the pump drive shaft 29 via a coupling 30.

The rotary shaft 29 is sealed at the stuffing box 32. It is with the design of the stuffing box that the invention is concerned.

Fig 2 shows a version of the invention, as applied to the pump-stuffing-box situation. A pump impeller 34 is keyed to the shaft 29. A rotary inner sleeve 36 is also keyed to the shaft 29. A static outer sleeve 38 is keyed to the housing 40 of the pump 27.

The inside surface 42 of the outer sleeve 38 is plain. The outside surface 43 of the rotary sleeve 36 is formed with a (single) spiral groove 45. The left-hand mouth 47 of the groove 45 opens into an entry-chamber 49. The right-hand mouth 50 of the groove 45 opens into an exit-chamber 52.

The entry-chamber 49 is sealed to the outside by means of a conventional rubber lip-type seal 54. The entry-chamber 49 is connected to a source of barrier-liquid via a port 56. The exit-chamber 52 is open to the inside of the impeller housing 40, i.e is open to the process fluid being pumped by the pump 27.

The inner and outer, rotor and stator, sleeves are sealed to the shaft and housing respectively by means of O-rings 58.

In use, the shaft 29 is driven into rotation by the electric motor, whereby the sleeve 36 rotates. The groove 45 therefore rotates, and its spiral configuration means that a molecule of liquid positioned in the groove will be carried along the groove. The direction of rotation of the motor 20 is arranged to be such that the molecule is carried to the right (in the orientation of the drawings); that is to say, from the entry-chamber 49 towards the exit-chamber 52.

The action of the spiral groove 45 in driving the molecules of barrier-liquid to the right serves to create a higher pressure of the barrier-liquid in the exit-chamber 52 than in the entry-chamber 49. In fact, the pressure of the barrier-liquid in the entry-chamber, where the barrier-liquid is picked up by the groove, may be at atmospheric pressure or thereabouts; the pressure in the exit chamber, where the barrier-liquid exits from the groove, can, by the action of the spiral groove, exceed the pressure of the process-fluid.

As such, the pair of sleeves 36,38 serves as a very effective seal against leakage of process-fluid out through the stuffing box 32. So long as the shaft 29 remains turning, and a supply of barrier-liquid is present at the port 56, the exit-chamber 52 will be filled with pressurised barrier-liquid.

Even if the process-fluid should be dead-headed (that is to say, the outlet pipe conveying the process-fluid away from the pump is blocked), the sleeves 36,38, with the spiral groove, still serve to pump barrier-liquid into the exit-chamber 52.

If the inlet pipe to the pump should be blocked, whereby the interior or impeller chamber of the pump is subject to vacuum, and cavitates, the barrier liquid is still pumped through into the exit chamber. By contrast, in some conventional designs, cavitation in the pump intake line can lead to liquids being sucked out of the stuffing box, to cavitation in the stuffing box, and to the consequent degradation of the sealing capability of the stuffing box.

The barrier-liquid can easily be kept clean (e.g by filtering), and the barrier liquid is forced out of the exit-mouth 50 of the groove 45. Therefore, the sleeves 36,38 and the surfaces 42,43, are kept free of any grit that may be present in the process-fluid. Process-fluid is often dirty, and can contain grit or other harmful materials, but the dirt is filtered out before the process-fluid can enter the groove or the surfaces 42,43.

The barrier-liquid fills the groove 45, and also forms a hydro-dynamic film that resides between the inner surface 42 of the outer sleeve 38 and the outer surface 43 of the inner sleeve 38; or at least between the surface 42 and the lands 43A of the surface 43 lying between the turns of the groove 45.

The barrier-liquid is supplied to the entry-chamber 49, and is drawn into the entry-mouth 47 of the groove. As a result, no part of the bearing area, i.e of the swept areas of the surfaces 42,43, is dry. Indeed, no part of the bearing area is dry even when the pump is cavitating (i.e sucking on no process-fluid).

The barrier-liquid is supplied to the port 56 at, or around, atmospheric pressure. The barrier-liquid does not have to be pre-pressurised outside the stuffing-box to a pressure higher than the process pressure: the action of the spiral-groove 45 serves to pressurise the barrier-liquid. In fact, the port 56 may be connected to a supply-pipe which simply dips into a reservoir of barrier-liquid, and the level of the liquid in the reservoir may be below the level of the port, whereby the barrier-liquid is drawn into the entry-chamber 49 at a pressure slightly below atmospheric. (The lip-seal 54 would then have to be selected on the basis of having to support a slight vacuum.)

The limitation to the action of the rotating spiral-groove as shown in Fig 2 is that when the pressure of the process-fluid gets too large, the barrier-liquid cannot emerge from the exit-mouth 40 of the groove 45.

Also, if the pressure of the process-fluid is higher than the pressure generated by the groove, the process fluid will tend to enter the interface between the lands 43A and the surface 42, and form its own hydro-dynamic film. This film may then break down, or be blown out, especially if the process-fluid has low viscosity or lubricity.

The smaller the clearance gap between the surfaces 42,43, the higher the pressure it takes to displace the hydro-dynamic film. In the case of cylindrical (i.e not tapered) surfaces 42,43, it is not practically possible to reduce the clearance between those surfaces to below about 0.1 mm, or even more if heavy service is expected.

With a gap like that, and using lubricating oil as the barrier-liquid, it has sometimes been found possible for the spiral-groove to drive liquid into the exit-chamber at pressures measured in many tens of psi, e.g 80 or 100 psi (100 psi = 690 kN/m^2). Such pressures can be enough to ensure that no leakage of process-fluid can occur, even with the pump dead-headed.

However, when the barrier-liquid has a viscosity less than that of lubricating oil, it is found that only much lower pressures can be supported when the clearance is as large as 0.1 mm. When water, for example, is used as the barrier-liquid, it is found that the pressure in the exit-chamber cannot be made to rise more than about 2 or 3 psi above that in the entry-chamber. This is not enough to guard against the leakage of process-fluid into the spiral groove and into the bearing interface.

In the configuration shown in Fig 2, the barrier-liquid, upon emerging into the exit-chamber 52, enters the process-fluid. Therefore, in Fig 2, the barrier-liquid has to be a liquid of a type that can be tolerated in the process-fluid. Although the quantity of the barrier-liquid that enters the process-fluid is small, when compared with the flow rate of the process-fluid being pumped through the pump, in many cases the process-fluid is such that lubricating oil, or other viscous liquid, cannot be allowed to enter the process-fluid even in tiny traces.

In many industrial pump installations, the process-fluid being pumped through the pump is water, or a water-based liquid. Lubricating oil (even small traces thereof) often cannot be tolerated in the water. But, with a clearance between the bearing surfaces as large as 0.1 mm, water itself is almost useless as the barrier-liquid. In order to build up worthwhile pressures in the exit-chamber 52 when water is the barrier-liquid, much smaller clearances than 0.1 mm would be required. But cylindrical surfaces arranged in a male-female bearing configuration, even when made with high precision, cannot, in practice, be run with clearances smaller than that. Cylindrical surfaces may be permissible in the case where traces of lubricating oil can be tolerated in the process-fluid, but that arises in only a very small proportion of pump installations. They are of little use when the barrier-liquid is water.

In Fig 2, the (plain) stator-sleeve 38 is fixed into the housing, but the rotor-sleeve 36 is free to slide axially on the shaft 29 (though keyed to rotate in unison with the shaft). A spring 65 urges the rotor-sleeve into the taper.

The tapered surfaces 42,43 are lapped together as a matched pair, so that the fit between them is very good over the whole swept interface, or bearing area, of the surfaces.

In Fig 2, during rotation a hydro-dynamic film establishes itself between the tapered surfaces. The hydro-dynamic film can be as thin or as thick as it requires to be, as dictated by the viscosity of the barrier-liquid, the process pressure, the speed of rotation, the dimensions of the groove, etc. It has been found that even when the barrier-liquid is water, which has only very small viscosity and lubricity characteristics, a film can establish itself between the surfaces 42,43, and a sizeable pressure can be developed in the exit-chamber 52.

In fact, it has been found that pressures of 60 psi, or higher, can easily be achieved in the exit-chamber, with the pump dead-headed, when the barrier-liquid is water. It is also found, during normal rotation, that the hydro-dynamic film is of such robustness that, after long periods of running, there is no evidence of direct contact between the surfaces 42,43.

However, direct touching contact between the surfaces 42,43 cannot be ruled out, and the sleeves should be made of such material as will accommodate occasional touching without seizing, smearing, pick-up, etc. One sleeve may be of cast iron and the other of bronze, for example. Or, plastic bearing materials may be used, such as PTFE.

In fact, even with water, the hydro-dynamic film that establishes itself in the interface between the surfaces 42,43, though very thin, is nevertheless found to be strong enough that the interface may serve as an actual journal bearing for the impeller shaft 29.

Fig 2 shows a notional bearing 67, which is housed in the bearing box 25, and which is, in a conventional installation, a considerable distance behind the impeller. That is to say, the impeller 34 is mounted on the end of a long overhang of the shaft. As such, the impeller can be susceptible to vibration of a troublesome period and amplitude.

It may be considered that the sleeves 36,38 might serve as a bearing assistant: that is to say, that the bearing formed by the sleeves might serve to assist the bearing 67 by dampening out some of the excessive vibrations. However, it has been found that the bearing 67 can actually be dispensed with in most cases. Even when the barrier-liquid is water, the pump shaft and the impeller are adequately supported by the bearing formed by the sleeves.

One reason for this excellent support is that the bearing is so close to the impeller. In Fig 2, there need be no provision for a separate stuffing-box-seal, whereby the bearing would have to be spaced from the impeller at least by the width of the seal: in Fig 2, the stuffing-box-seal and the bearing are one and the same.

Even if the pump cavitates on the intake, any out-of-balance loadings and other abusive vibration-inducing situations that might occur do not cause vibratory excursions of the shaft and impeller, because the bearing is so close to the impeller. The shaft runs smoothly and evenly under conditions that would be expected to cause a conventional pump to shake its stuffing box seal into leakage.

The spiral groove in the tapered sleeve serves to drive the barrier-liquid along the groove from the entry-mouth to the exit-mouth. It will be understood that the distance travelled by the barrier-liquid per turn of the groove will vary because of the taper: the hoop-length of a turn of the groove at the thin end of the taper is less than the hoop-length of a turn at the thick end.

To allow for this, the designer should provide that the groove becomes slightly larger in cross-sectional area towards the thin end of the taper, to compensate for the reduced hoop-length. Fig 4 shows how the groove 70 may be cut slightly more deeply at the thin end.

The sleeves and the groove, and the direction of rotation of the motor, should be set up so that the groove drives the barrier-liquid towards the impeller. In doing so, as shown in Fig 3, the designer may arrange the sleeves 72,73 to drive the barrier-liquid towards the thick end of the taper. Again, the groove 70 should be a little larger in area (cut a little deeper) at the thin end to make up for the reduction in hoop-length.

In another variation (not shown) the rotor sleeve may be the outer sleeve and the stator the inner sleeve. In that case, the groove would have to be placed on the inside surface of the outer-sleeve, which is much harder to manufacture than putting the groove on an outer surface. However, sometimes, to achieve the best hydro-dynamic effects, the designer may prefer to provide grooves (either spirals or rings) on the inside surface of an outer, stator, sleeve such as sleeve 38 in Fig 2.

It may be desired to protect the system against leakage of the process-fluid even when the motor should stop rotating. The system of Fig 2, for example, would allow leakage of the process-fluid back up the spiral

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groove if the rotation should stop.

To guard against leakage when the motor is stopped, as shown in Fig 5, a lip-seal 74 is provided between the exit-chamber 52 and the interior or impeller chamber of the pump housing 40.

During normal running, the pressure developed in the exit-chamber 52 exceeds the pressure developed in the impeller chamber, and the seal-lip is blown open, allowing the barrier-liquid to flow into the impeller chamber. When rotation stops, the now-higher pressure in the impeller chamber forces the lip-seal to close, thus preventing fluid in the impeller chamber from entering the exit-chamber.

Fig 6 illustrates another way in which the components can be made. Here, the rotor sleeve 76 is formed unitarily with the impeller 78. The outer, stator, sleeve 80 is spring loaded into the taper. It may be noted that only a simple plain diameter 79 need be provided in the stuffing box housing, without a shoulder at the impeller end.

As an alternative to Fig 6 (not shown), the stator sleeve may be built into the stuffing box housing. However, the sleeves being tapered, it is not advisable to have both the rotor surface built into the impeller and the stator surface built into the housing, since one of the surfaces must be free to slide axially if the benefit of the taper is to be realised.

In the assemblies illustrated, provided the barrier-liquid has adequate viscosity and lubricity, the set-up can be expected to give a highly reliable seal against all conditions of the process-fluid likely to be encountered in a practical installation. However, it is important to note that if the supply of barrier-liquid should fail, the entry-mouth of the groove will run dry, and then the whole bearing area between the surfaces may run dry, or parts of that area may run dry. If that happens, the pressure in the exit-chamber 52 will fall, perhaps allowing process-fluid to enter the spiral groove and even leak out (to the left) of the stuffing box seals. Once the surfaces run dry, or partly dry, their service lifetime is much reduced.

Therefore, the designer should see to it that the entry-chamber 49 never runs dry of barrier-liquid.

In suitable cases, the barrier-liquid can be taken from the process-fluid. This can be done, of course, if the process-fluid is lubricating oil, but some other types of liquids are suitable also.

It may be considered that where the process-fluid is lubricating oil it would not matter if the entry-chamber 49 (Fig 2) should run dry, because then the process-fluid would enter the groove 45 from the exit end, and would lubricate the bearing interface. However, if the entry-chamber were to run dry, even when the process-fluid is lubricating oil, the process-fluid could not be expected to reach and to wet the whole swept area of the surfaces, whereby dry-touching of the surfaces could not be ruled out. By ensuring the entry-chamber 49 is kept filled, the designer ensures the hydro-dynamic film is kept intact, and the surfaces do not touch.

Fig 7 shows a system for ensuring the entry-chamber is kept filled with barrier-liquid. In this case, the barrier-liquid is derived from the process-fluid via pipe 83, and also from a separate supply source 85. The supply is controlled by valves, as shown, which can be manually or automatically controlled, as required.

The barrier-liquid passes through a temperature controller 86 and through a filter 87. It is important that the barrier-liquid be filtered clean, since grains of grit trapped between the bearing surfaces would spoil the self-sustaining character of the hydro-dynamic film.

The pressure of the barrier-liquid in the entry-chamber 49 is controlled by a pressure regulator 89. Alternatively, it may be arranged that pressure in the entry-chamber is controlled by drawing the barrier-liquid from a level of the liquid. The height of the level determines the head or pressure.

It has been the case in the designs described so far that the barrier-liquid can be allowed to leak into the process-fluid, and indeed that the process-fluid is the source of the barrier-liquid. However, although that is the simplest set-up mechanically to arrange, its applicability is not so common. Mostly, it is required that the process fluid not be diluted by the barrier-liquid, and also the process-fluid is not suitable for use as the barrier-liquid.

Fig 7 also shows means for controlling the barrier-liquid in the case where the barrier-liquid is to be kept separate from the process-fluid. When the two are separate, additives may be added to the barrier-liquid, as desired, especially where the barrier-liquid is water-based, to enhance its properties of viscosity and lubricity. The barrier-liquid is re-circulated by drawing the liquid from the exit-chamber via pipe 82, while maintaining the pressure in the exit-chamber to a desired value by means of regulator 84.

Back-to-back lip-seals may be arranged between the impeller chamber and the exit-chamber, in place of the single lip-seal 74 (Fig 5). Similarly, back-to-back lip-seals may be positioned at the other end of the tapered sleeves, to seal the entry-chamber 49 (Fig 2) from leakage both inwards and outwards.

Lip-seals can only support pressure-differences of a few psi. Therefore, it is important to protect the seals from large pressure differentials. This is done by the pressure regulators as shown, preferably automatically. The pressure in the exit-chamber 52 is compared with the pressure in the impeller chamber 49, and the exit-pressure is regulated such that the pressure in the exit-chamber is a few psi more than the pressure in the impeller-chamber. Then, if the seals should fail, the process fluid cannot leak into the stuffing box. In the case where leakage of the process-fluid into the barrier-liquid is less important than leakage of the barrier-liquid into the process-fluid, the pressure differential may be arranged the other way round, i.e the pressure in the exit-chamber 94 is then kept a few psi lower than the pressure in the impeller chamber 92. In the circuit of Fig 7, the two pressures may be (automatically) compared, and the difference-computation used to regulate, at 84, the pressure in the exit-chamber.

The pressure in the exit-chamber is generated by the action of the spiral-groove (there is no external source of pressure), and of course the designer should see to it that the pressure-generating capability of the spiral-groove is adequate -- given the viscosity of the barrier-liquid, the speed of rotation, and the rest of the parameters. If the spiral-groove is only capable of delivering, say, 60 psi, then if the process-pressure might rise to 100 psi (e.g during dead-heading), the seals 90 would fail.

Similarly, lip-seals 49 should not be subjected to more than a few psi, and again a pressure regulator 69 controls the pressure of the entry-chamber. (As mentioned previously, pressure in the entry-chamber may alternatively be controlled by drawing the barrier-liquid from a controlled head level.)

When these precautions are taken, the barrier-liquid circulates around a circuit that is quite separate from the process-fluid. The separated barrier-liquid is filtered, and its temperature and other properties are controlled.

It may be noted that the barrier-liquid derived via pipe 82, and being circulated around the circuit, is the whole of the liquid passing through the spiral-groove. In previous designs where a (pressurised-externally) barrier-liquid has been separately-circulated, the circulation has been on a by-pass basis. In Fig 9, the whole of the barrier-liquid that passes along the spiral-groove passes into the pipe 82 and is re-circulated.

As mentioned, elastomeric lip-seals can only support a few psi, and, when that is not good enough, or when lip-seals are not suitable for other reasons, mechanical seals may be substituted.

Fig 8 shows an example of a stuffing-box with two mechanical seals 105,107. The seals are located either end of a matched pair of tapered sleeves 108,109, a spiral-groove being formed on the outer surface of the inner, rotor, sleeve 109. The inner sleeve 109 is keyed at 120 for rotation with a shaft 123, and can slide along the shaft. Entry- and exit-chambers 125,127 are created by the arrangement of the components, and pipes 128,129 convey barrier- liquid through the chambers and through the spiral-groove.

It will be understood that the sleeves and seals as shown in Fig 8 comprise a cartridge that can be made as a convenient sub-assembly, which is suitable for fitment, as an integrated unit, into a stuffing-box housing 130.

Fig 9 shows another arrangement that uses mechanical seals. In this case, the tapered interface of the sleeves faces the other way: the extra space inside what is now the thick end of the inner sleeve 132 (i.e the right-hand end in Fig 9) is used to accommodate one of the mechanical seals 134. This allows the overall length of that seal and the sleeves together to be kept to a minimum. Insofar as the pair of sleeves counts as a journal bearing for the rotating shaft 135, it is important that the overhang of the shaft and impeller, beyond the bearing, be as small as possible. Placing the seal 134 inside the sleeve 132 assists this. It is not so important that the other mechanical seal 136 be short in the axial direction.

As show in Fig 9, another chamber 137 may be provided outside the entry-chamber 147. The barrier-liquid is used to lubricate the bearings 138 in a bearing box. Often, however, this will not be appropriate, and the arrangement of Fig 10 would be preferred.

The seal 139 is mounted on a drive-sleeve 140, which is tightened onto the shaft 135 by means of clamp-screws 141. The right-hand end of the drive-sleeve 140 is formed with drive-teeth 142, which engage corresponding drive-slots in the inner sleeve 143.

The matched pair of tapered sleeves and the seal 134 form a first cartridge sub-assembly, which is suitable for fitment into the housing 144; the drive-sleeve 140 and the seal 139 form another cartridge sub-assembly, which is clamped to the shaft 135 and, by means of a cover 145, bolted to the housing 144.

Again, Fig 7 illustrates the circuit for supplying barrier-liquid to the entry-chamber 147 (Fig 9) and recovering the barrier-liquid from the exit-chamber 149. As shown, this circuit is passive (i.e no energy input) (except that provision may be made for the liquid to be cooled/heated). It may be arranged that the pressure of the process-fluid is monitored, and compared with the pressure in the exit-chamber, to ensure that the seal 134 is not subjected to an abusive pressure differential.

It may be noted that the circulation of barrier-liquid between the sleeves and through the entry- and exit-chambers, serves also to flush the mechanical bearings of any dirt and debris that might build up in the chambers. Conventional mechanical seals are often provided with flush and drain facilities to clean out debris: such facilities are present automatically in the present case without the need for supplied energy, and virtually for nothing.

It may be noted that the inner sleeve 132;143 (Fig 9;Fig 10) is free to slide axially on the shaft 135, and that the spring 150 (Fig 10) urges the inner sleeve to the left, i.e more deeply into the taper. However, the characteristics of the spring 150 are selected primarily on the basis of the requirements of the mechanical seal 134. It might be considered that the force with which the tapered sleeves are pushed together would need to be tightly controlled between close limits. However, this is not the case. The hydro-dynamic film that forms between the tapered surfaces is very robust. Once the film is established between the surfaces, an increase in the force urging the surfaces together has little effect in making the surfaces actually move towards each other, while the force required to physically break through the film and close the surfaces together into touching contact is considerable. Thus, the film-filled gap between the tapered surfaces is self-setting and self-sustaining to a large extent, even though the force pushing the surfaces together may vary, or may be set by the requirements of the mechanical seals.

The axially-movable sleeve 132 is subject to the pressure in the exit-chamber 149, which, like the spring 150, serves also to urge the sleeve 132 more deeply into the taper.

It has been mentioned that the pair of sleeves with the spiral groove serves as a journal bearing for the impeller shaft. In the case of conventional stuffing-box-type pumps, when the requirements of the shaft bearings (located in the bearing box 25) were being determined, one of the key factors in the computations was the length of the overhang by which the impeller extended out beyond the bearing. This overhang determined the period and amplitude of vibrations that might be encountered, and which the bearing had to contain. In the present case, however, this overhang is virtually zero (less than a diameter of the shaft). Therefore, when the shaft bearing is formed by sleeves located very close to the impeller, the loading on the bearing is considerably less than the loading normally encountered with pump-bearings that have to cater for overhanging shafts.

Thus, the new design not only eliminates the need for a bearing box (such as 25), along with its need for lubrication etc, but the new design puts considerably less loading and usage demands on the bearing itself.

Given that in conventional designs the nearest bearing to the impeller can be, typically, 15 or 20 cm away from the impeller, it is not difficult, with the present design, to make a large improvement. The designer should space the bearing interface surfaces axially along the shaft preferably within no more than about 1 diameter of the shaft, from the impeller.

Naturally, the designer should see to it that provision is made for axial thrust forces on the pump shaft to be supported, and it will often be convenient to provide a thrust bearing outside the pump housing, and between the pump housing and the coupling (such as 30) for that purpose. (Couplings usually cannot transmit axial forces.)

Fig 11 shows an arrangement in which not only is the bearing box 25 eliminated, but also eliminated is the coupling 30 between the motor shaft and the pump shaft. The shaft 152 serves both as the armature of the electric motor 154 and the drive shaft of the pump impeller 156. The housing of the pump and the housing of the motor may be bolted together as one unit, the accurately-machined spigots at 158 serving to ensure alignment. Because there is no coupling, axial thrust forces on the shaft may be supported by a thrust bearing 160 actually in the motor housing.

Fig 12 shows a matched pair of tapered sleeves arranged in a seal/bearing configuration, where there is no through-shaft. The rotor 163 is formed as a stub.

The outer sleeve 164 is keyed at 165 to the stationary housing 167. The outer sleeve can float axially (vertically) within the housing, and is pressed upwards by springs 167.

The tapered surface of the rotor sleeve 163 is provided with a spiral-groove 169. By the action of the spiral-groove, when the sleeve 163 is rotated, barrier-liquid supplied to the entry-chamber 170 is forced down to the exit-chamber 172, and a hydro-dynamic lubrication film is established between the tapered surfaces.

The upper end of the entry-chamber 170 is sealed by a sealing interface 174 between the rotor and the stator sleeves. There is very little pressure on this sealing interface since liquid in the entry-chamber 170 is being drawn into the entry-mouth of the spiral-groove 169.

Fig 13 shows a rotor sleeve 174 of the type used in the designs as described. In tests on sleeves, the following performances were noted.

Male-tapered Sleeve No 1 (running in a plain female sleeve):
- included angle of taper = 20 degrees;
- diameter x = 47.6 mm;
- diameter y = 31.8 mm
- length z = 44.4 mm;
- spiral groove = single-start, cut 2.03 mm wide x 0.20 mm deep; - groove pitch = 6.35 mm turn-to-turn;
- width of land between turns = 4.32 mm

Test 1:
- liquid = water at 18 degC (viscosity = 1.15 centi-stokes)
- speed of rotation = 1750 rpm;

Result: generated pressure = 70 psi,
flow rate = 2.2 litres/hr

Test 2:
- liquid = water at 18 degC
- speed of rotation = 3500 rpm;

Result: generated pressure = 100 psi,
flow rate = 4.5 litres/hr

Test 3:
- liquid = water at 18 degC
- speed of rotation = 1100 rpm;

Result: generated pressure = 40 psi,
flow rate = 1.5 litres/hr

Test 4:
- liquid = SAE 30 min. oil at 18 degC, viscosity 50 c-stokes
- speed of rotation = 1750 rpm;

Result: generated pressure = 300 psi
flow rate = 3.0 litres per hour.

Sleeve no 2 (same as sleeve no 1, except that length z reduced to 38.0 mm by machining off the thin end) (running in plain female sleeve).

Test 5:
- liquid = water at 18 degC
- speed of rotation = 1750 rpm;

Result: generated pressure = 60 psi.

Sleeve no 3 (same as sleeve no 1, except that length z reduced to 31.6 mm by machining off the thin end) (running in plain female sleeve).

Test 6:

- liquid = water at 18 degC
- speed of rotation = 1750 rpm;

Result: generated pressure = 50 psi.

Sleeve No 4 (same as sleeve no 1 except that two grooves are cut, each pitched 12.7 mm, width of land between adjacent turns = 4.32 mm) (running in a plain female sleeve):

Test 7:

- liquid = water at 18 degC
- speed of rotation = 1750 rpm;

Result: generated pressure = 32 psi,
flow rate = 5.4 litres/hr

Test 8:

- liquid = water at 93 degC (viscosity = 1.13 centi-stokes)
- speed of rotation = 1750 rpm;

Result: generated pressure = 64 psi

Sleeve No 5 (same as sleeve no 1, single groove, except that groove depth increased to 0.25 mm deep) (running in a plain female sleeve):

Test 9:

- liquid = water at 18 degC
- speed of rotation = 1750 rpm;

Result: generated pressure = 60 psi,
flow rate = 3.0 litres/hr

Sleeve No 6 (same as sleeve no 1, single groove, except that groove depth increased to 0.30 mm deep) (running in a plain female sleeve):

Test 10:

- liquid = water at 18 degC
- speed of rotation = 1750 rpm;

Result: generated pressure = 45 psi,
flow rate = 4.5 litres/hr

Sleeve No 7 (same as sleeve no 1, single groove, except that groove depth increased to 0.35 mm deep) (running in a plain female sleeve):

Test 11:

- liquid = water at 18 degC
- speed of rotation = 1750 rpm;

Result: generated pressure = 20 psi,
flow rate = 6.7 litres/hr

Test 12:

- liquid = molasses at 18 degC
- speed of rotation = 1750 rpm;

Result: generated pressure = 500+> psi,

Tests 1,2,3 show the extent to which increased speed of rotation results in both more pressure and more flow from the spiral-groove.

Test 4 compared with Test 1 shows the much greater pressure available when oil is used as the liquid, in place of water.

Tests 5,6 compared with Test 1: each 6.4 mm reduction in length shows a 10 psi fall in pressure availability.

Test 7 compared with Test 1 shows that the pressure is halved and the flow rate is doubled when two grooves are present, as compared with one groove.

Test 8 shows that when the water isa almost boiling there is roughly 8% drop of pressure capability.

Tests 9,10,11 show the drop-off in pressure as the spiral groove is cut deeper. When the liquid is water, a groove depth of more than about 0.50 mm generates hardly any pressure at all.

Test 12 shows that when the liquid is molasses, a groove depth of 0.50 mm produced more than 500 psi. With molasses, however, little impression was made on the molasses when the grooves were less than 0.30 mm deep.

The tests also showed that viscous liquids such as oil were much more tolerant of changes in speed, grove-depth, etc, than water. Nevertheless, it is clear that the groove as described has a high performance in producing pressure and volume flow, even when the liquid is water.

In most cases, the requirement, when creating a pressure in the barrier-liquid in the stuffing-box, is that a high pressure be achieved. In fact, often, the requirement is that the barrier-liquid be at a higher pressure than the process-fluid.

Theoretically, there would be no need for a high volumetric flow-rate, so long as the pressure is generated. In fact, it can be said that a high flow-rate would be a disadvantage, especially if the flow leaks through into the process-fluid.

However, it is important, when creating a pressure in the barrier-liquid in the stuffing-box, that the means for creating the pressure is robust enough to be able to create that pressure even though the barrier-liquid is circulating vigorously through the stuffing-box, and even though the stuffing-box seals may be leaking.

In this connection, the attempts to create pressure by using the scroll markings as shown in US-4,290,611, may be reviewed. These scroll markings may have a potential to generate pressure, but only if the flow is at the zero or mere-trace level. Pressure is created only while there is no leakage. What can happen is, once a measurable magnitude of leakage starts to appear at the seal, for whatever reason, the pressure-generating capability plummets, and the seal immediately opens and permits copious leakage.

That is not what is wanted. The mechanism for creating the pressure should be versatile and robust enough to maintain the barrier-liquid pressure, even though the seals may be leaking. It is of little practical use if the process-fluid can burst straight through into the stuffing-box as soon as a tiny leak starts to develop in the seal between the process-fluid and the stuffing-box.

In the designs as described, this desired degree of flexible robustness of pressure-generation can be obtained. Provided the parameters of groove size, etc, are properly tailored to the speed of rotation and the viscosity of the barrier-liquid, the barrier-liquid will force its way out inexorably and continuously from the exit-mouth of the spiral-groove.

It is, in most cases of centrifugal pump installations, simple enough for the designer to ensure that enough pressure is available from the spiral-groove to overcome any pressure that might be present in the process-fluid, and to do so even when the spiral-groove is producing flow rates, of the magnitude of a few litres per hour.

It has been found that this level of performance can be achieved even though the barrier-liquid is water.

The designer should make the spiral-groove to the right dimensions to achieve the desired pressure and flow.

The groove should not be cut deeper than about 0.4 mm, given that its width is around 2 mm. In general, the total or aggregate cross-sectional area of the spiral-groove(s) should not be more than about 1 sq mm, and preferably should be no more than about 0.5 sq mm.

The groove should not be too small. A small groove might still be able to produce the pressure, but would not be capable of delivering adequate volumetric flow-rate at that pressure. A groove cross-sectional area of about 0.3 sq mm is a minimum below which the flow-rate of barrier-liquid would be inadequate for most cases.

That is for water: the groove may be around twice the area when the liquid is oil.

It may be noted that the spiral-groove as described can provide not only high pressure but can at the same time provide a good flow rate at that pressure. This has not previously been achieved in a pump stuffing box situation.

The volume of one turn of the groove is typically about 0.06 milli-litres: it may be calculated that, when liquid flows from the exit mouth of the groove at the rate of 3 litres per hour (as typically occurs), this corresponds to a delivery of about 0.03 milli-litres of liquid per revolution, or about half the volume of one turn of the groove per revolution.

The total length of the groove between entry-mouth and exit-mouth is typically at least 50 or 70 cm. 30 cm should be regarded as the minimum. If the total length of the groove is too short, pressure cannot be developed.

A key factor in the use of the invention lies in the establishment of the hydro-dynamic film between the surfaces. The running clearance or gap between the surfaces should be small enough, and the lands between adjacent turns of the spiral-groove should be wide enough, to ensure that the film is robust and secure. A land-width of 4 mm between turns has been found satisfactory. 2 mm should be regarded as a minimum.

As described herein, when the surfaces are tapered, and one of the tapered surfaces is formed on a sleeve that moves axially, the designer can take it that the surface-to-surface gap, which determines the film thickness, can be very small. The smaller the gap, and the wider the land, the greater the pressure difference that can be supported between the turns of the groove.

The longer the spiral-groove, the greater the final pressure that can be achieved. It will be noted also that, the longer the groove, the more the groove resists back-leakage, and the larger the groove can be as to its cross-sectional area, without compromising the pressure at the exit-mouth, and hence the more volume can be moved along the groove.

It is recognised that the dimensional envelope of a typical stuffing box installation is such that the size of spiral-groove that can readily be accommodated therein can produce a more-than-adequate combination of pressure and volumetric flow rate through the groove.

The configuration of the spiral-groove means that even if the mechanical seals should fail, and even if the motor should stop rotating, fluid can only leak through the groove at a very low flow-rate. The groove may be 1 sq mm in area and 50 cm long, through which leakage will inevitably be slow.

As mentioned, sometimes the designer must provide that no leakage of process-fluid can occur, even if the mechanical seal between the exit-chamber and the impeller-chamber should fail (assuming the motor keeps running). In this case, the designer should provide for the pressure in the exit-chamber to be regulated to a pressure slightly higher than the pressure in the impeller-chamber. The pressure-regulators are provided outside the stuffing-box housing, and coupled to the chambers by pipes, as described.

Preferably, the engineer should provide for the automatic regulation of the pressure in the exit-chamber to a value that is just higher than the pressure of the process-fluid in the impeller chamber. The engineer thereby ensures that, if the seal should leak, barrier-liquid will flow from the exit-chamber into the impeller chamber, rather than that process-fluid will flow from the impeller chamber into the exit-chamber. So long as the pressure in the exit-chamber is kept higher than the pressure of the process-liquid, the process-liquid cannot leak into the exit-chamber, and thence to the outside.

The engineer may decide, alternatively, to regulate the pressure in the exit-chamber to a smaller pressure than the pressure in the impeller chamber, if that is more appropriate. The main point about setting the pressures is that the pressure in the exit-chamber should be set to a value which does not differ by very much, whether higher or lower, from the pressure in the impeller chamber.

Similarly, the designer can provide that the pressure differential is also kept small, across the mechanical seal between the entry-chamber and the outside. The brunt of the total pressure differential between the process fluid and the atmosphere is then taken by the spiral-groove which runs between the entry-chamber and the exit-chamber. That is to say, the spiral-groove is used to keep the pressure-differentials across the seals to a minimum. The smaller the pressure differential across a mechanical-rub type seal, the longer the life expectancy of the seal.

The engineer may regulate the pressure in the exit-chamber to a level that is just below, or just above, process-pressure. Although the process-fluid pressure may then be high, the differential on the mechanical seal is low. This of course is good for the mechanical seal, but also, the fact that the pressure in the exit-chamber is high means that the interface pressure between the tapered surfaces is also high, which is good for the surfaces as a running bearing interface.

When the pressure of the process-fluid is high, i.e more than a few psi, even though the differential pressure on the seal may be low, the prudent engineer prefers that the seal to the process-chamber should be of the mechanical type, not the elastomeric lip type. On the other hand, the seal between the entry-chamber and the outside environment can often be safely provided as an elastomeric lip type. Mechanical seals are generally much more expensive than elastomeric lip seals.

The designer should see to it that the supply of barrier-liquid to the entry-mouth of the groove does not run dry. However, this is not too demanding a requirement in the normal industrial pump environment. Besides, the supply side of barrier-liquid need not be pressurised, since the spiral-groove will draw barrier-liquid in from a (slight) vacuum (negative head) if necessary.

The barrier-liquid should be kept clean. If dirt were to appear between the tapered surfaces, that could affect the ability of the hydro-dynamic film to maintain the correct gap between the surfaces. However, it should be pointed out that, in tests, the spiral-groove in the tapered surfaces actually in itself served to clean particles of grit from the surfaces.

It may be surmised that the reason for this self-cleaning capability is that dirt particles tend to congregate, not within the hydrodynamic film, but rather in the spiral-groove. The velocity of liquid moving along the groove then tends to flush the dirt to the exit end of the groove. The lands between the turns of the spiral-groove are narrow enough that the liquid in the films, in the lands, is able to wash into the grooves. The lands should not be more than about 8 mm wide, from this standpoint.

Selection of the materials form which the rotor and stator sleeves are made is important. This is true even though, once the hydrodynamic film is established there is theoretically no contact between the surfaces, because occasional touching contact is inevitable.

The component having the surface in which the spiral-groove is cut should be of a harder material than that on which the plain surface is formed. Then, if any wear should then occur, it is the plain surface that will be eroded, leaving the groove intact. In fact, a little wear is beneficial, insofar as it produces buffing of the surfaces, and enhances their intimacy of fit.

Suitable combinations are that the grooved, male, rotor be of stainless steel, coated with about 0.1 mm of hard chrome, or a ceramic bearing material. (Stainless steel without a coating would not be suitable, as it tends to smear.) The plain, female, stator may be of carbon (graphite), PTFE, or one of the (many) composite materials developed for prolonged running in contact with hard metal.

The designer has to provide some means for urging the tapered surfaces together. This may take the form of a mechanical spring, or provision may be made for the barrier-liquid pressure, or the process-fluid pressure, to act on the axially-movable sleeve in such a way as to urge the surfaces together. (If no axial-constraint at all were provided, the surfaces would just move apart, and the hydro-dynamic film could not develop.)

In assessing just how that is to be done, the designer should take note of which is most important: leakage of barrier-liquid into process-fluid, or leakage of process-fluid into barrier-liquid. When pumping a drinkable liquid, for example, it is important that the drinkable liquid be free of traces of the barrier-liquid, but the drinkable liquid is not toxic and so it does not matter so much if a little of the drinkable liquid leaks out through the stuffing-box. In another case, the process-fluid may be a toxic liquid, or one that is carcinogenic in tiny traces, and in that case dilution of the toxic liquid by the barrier-liquid is preferable to leakage of the toxic liquid into and through the stuffing-box.

The designer may arrange that the pressure of the process fluid is the main agent for forcing the tapered surfaces together, or the pressure of the barrier-liquid. Or, the components may be arranged so that the movable sleeve is neutral to one of, or both, pressures, and its axial force is determined by a mechanical spring. Or, some suitable combination of pressure-exposures and mechanical springs may be provided.

As the pressure of the barrier-fluid increases, the hydro-dynamic film increases its tendency to drive the tapered surfaces apart. Therefore, it is generally preferable for the axially-movable sleeve to be urged more strongly into the taper as the process pressure increases.

If there were too little resistance to the movable sleeve moving away, the pressure could not build up properly. Therefore, the designer should provide that the axial forces acting on the movable sleeve are large enough to hold the tapered surfaces together to allow the desired pressures to be achieved.

It does not matter so much if the tapered surfaces are pressed together more tightly than is necessary, because the hydro-dynamic film is very robust. On the other hand, the sleeves should not be pressed together so hard that the film actually breaks down, and allows the tapered surfaces to touch mechanically. Even if the surfaces could, in such a case, tolerate being run together, tests show that forced contact between the surfaces causes a drop in the pressure in the exit-chamber.

The angle of the taper should be neither too large nor too small. The steeper the angle, the more force is required to hold the tapered surfaces together, to allow pressure to develop. The angle becomes too steep when the force needed to hold the sleeves together is too large to be conveniently provided, or conveniently controlled.

Another problem arises when the taper angle is too steep. Inasmuch as the tapered sleeves are serving as a journal bearing for the impeller shaft, a journal load on the tapered surfaces naturally induces an axial loading between the surfaces. This induced axial force tends to drive the movable sleeve out of the taper. The steeper the taper angle, the greater the induced axial loading on the movable sleeve. If the angle were too steep, it may then happen that such means as are provided to resist the axial movement of the movable sleeve would compromise the movable sleeve's ability to settle itself into the most favourable position for the hydro-dynamic film to develop. However, provided the taper angle is not steep, the induced axial forces on the movable sleeve, due to the sleeves' serving as a journal bearing, may be ignored.

It is pointed out again that, because the overhang of the impeller beyond the sleeves is so small, the journal bearing loads are also small. Journal loads on impellers can also be reduced by balancing the process-fluid outlets from the pump chamber. When the main journal loads were caused by vibrations due to the long overhang, it was often not worth it to balance the outlet pressures, but when the bearing is very close to the impeller, the journal forces, both from output imbalance and from vibration, can be reduced to very small levels.

Based on the above considerations, a taper angle of between 10 and 30 degrees inclusive (i.e between 5 and 15 degrees half-angle) has been found to give good results, with 20 degrees as the preferred value.

The maximum included angle that could be made to work properly according to the invention is about 60 degrees. Above that, the axial forces induced on the movable sleeve cannot be properly controlled.

The taper angle should not, on the other hand, be too small. If the included angle of the taper is too small, manufacture of the tapered surfaces, by lapping them together, can become difficult, because the surfaces may tend to lock up. Also, if the tapered surfaces should become dry, and the coefficient of friction between them thereby increases, lock-up may again occur. Therefore, preferably the angle of the taper should not be less than the self-locking angle. The self-locking angle may be determined from the coefficient of friction between the two sleeves. For metals like cast-iron and bronze, the included angle should not be less than about 7 degrees.

In the designs as described, the spiral-groove has been formed on the rotor sleeve, not on the stator sleeve, and this is preferred. It is also preferred that the rotor sleeve be the male sleeve, since then the groove is cut on an outwards-facing surface; the inside-facing surface of the female sleeve is left plain.

It is contemplated that the spiral-groove may be formed on the tapered surface of the stator, leaving plain the tapered surface of the rotor. It is also contemplated that grooves may be cut on both the rotor and the stator.

In the invention, the hydro-dynamic film should be robust and secure. Unless the liquid is very lubricious and viscous, the preference would be for the groove to be cut in only one of the surfaces, while the other is left plain. Preferably it is the surface in the stator that is left plain, and preferably it is the female surface that is left plain. If grooves are cut in both surfaces, that might tend to break up the film. However, in cases where the barrier-liquid is, or might be, dirty, and the liquid is adequately viscous and oily, grooves on both surfaces may be preferred.

The expression "psi" as used herein is a measure of hydraulic pressure. In SI units, 1 $N/m^2 = 1.45 \times 10^{-4}$ psi.

The embodiments of the invention in which an exclusive property or privilege is claimed are defined as follows:

1. Apparatus for a rotating shaft, comprising a stator component, and a rotor component adapted for rotation about the axis of rotation of the shaft, wherein:

the rotor and the stator components are formed with complementary tapered bearing-surfaces coaxially disposed about the axis;

the bearing-surfaces of the rotor and stator components are so arranged as to sweep each other in a hydrodynamic-bearing relationship, over an area termed the bearing area, upon rotation of the rotor;

one of the bearing-surfaces is formed with a continuous spiral groove, which extends in a spiral configuration along and around the bearing-surface, over the bearing area;

the spiral-groove comprises several turns extending over the bearing-surface, the arrangement thereof being such as to leave lands between adjacent turns of the spiral-groove;

the apparatus is so structured that the spiral-groove has an entry-mouth and an exit-mouth;

the apparatus is so structured as to define an entry chamber and an exit chamber, being chambers which are in fluid-conveying-communication with the entry mouth and the exit mouth respectively;

the apparatus includes a means for receiving a barrier-liquid from a source of barrier-liquid, and for conveying the barrier-liquid to the entry chamber;

the apparatus is so structured that the fit of the bearing-surfaces ensures the establishment and the continuance, during rotation, of a hydrodynamic film between the bearing surfaces.

2. Apparatus of claim 1, wherein the complementary bearing-surfaces lie fitted together in a male-female configuration over the bearing area.

3. Apparatus of claim 2, wherein the bearing-surfaces are arranged in a conical taper configuration.

4. Apparatus of claim 1, 2 or 3, wherein the apparatus includes a means for guiding one of the rotor or stator components for axial movement relative to the other, and the apparatus includes a means for urging that component axially, in the sense to urge the tapered surfaces together.

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5. Apparatus of claim 4, wherein the apparatus is suitable for installation in a process-fluid-transfer machine, which defines a process-chamber containing a process-fluid under pressure, and in which a rotating shaft of the machine extends through the apparatus; and the rotor and stator fit around the rotary shaft.

6. Apparatus of claim 5, wherein the means for urging is an area of the component which is so arranged as to be exposed, when the apparatus is installed in the machine, to process-fluid pressure, and which is so configured that the higher the process-fluid pressure, the greater the force urging the tapered surfaces together.

7. Apparatus of claim 4, wherein the means for urging is an area of the component exposed to barrier-liquid pressure.

8. Apparatus of claim 4, wherein the means for urging is a mechanical spring.

9. Apparatus of claim 1, 2, 3, 4, 5, 6, 7 or 8, wherein:

the apparatus includes a first seal, being a seal of the surface-rubbing kind in which a means is included for resiliently urging sealing-surfaces of the seal together into rubbing, sealing, contact;

and the first seal is so located as to seal and separate the entry-chamber from the outside environment.

10. Apparatus of claim 9, wherein:

the apparatus includes a second seal, being a seal of the said surface-rubbing kind;

and the second seal is so located, when the apparatus is installed in the machine, as to seal and separate the exit-chamber from the process-chamber.

11. Apparatus of claim 10, wherein the first seal is a flexible-lip type seal, and the second seal is a mechanical seal, wherein surfaces of relatively hard material are resiliently rubbed together.

12. Apparatus of claim 5, in combination with the machine, wherein:

the machine is a centrifugal pump which has an impeller mounted on the shaft;

the apparatus is so located with respect to the machine that the bearing surface lies axially-spaced from the impeller by no more than 1 diameter of the shaft.

13. Apparatus of any one of claims 1 to 12, wherein the included angle of taper of the tapered surfaces is between 7 degrees and 30 degrees.

14. Apparatus of any one of claims 1 to 13, wherein the spiral-groove extends in a continuous, regular uninterrupted, open-ended helix, over the bearing surface.

15. Apparatus of any one of claims 1 to 13, wherein the spiral-groove is of the single-start type.

16. Apparatus of any one of claims 1 to 15, wherein the overall length of spiral-groove is at least 30 cm.

17. Apparatus of any one of claims 1 to 16, wherein the width of plain land between adjacent turns of the spiral-groove is at least 2 mm.

18. Apparatus of any one of claims 1 to 17, wherein the aggregate of all the land widths over the bearing area is at least half of the average of all the diameters of the bearing surfaces.

19. Apparatus of any one of claims 1 to 18, wherein the cross-sectional area of the spiral-groove is less than 1 sq mm.

20. Apparatus of any one of claims 1 to 19, wherein the cross-sectional area of the spiral-groove is more than 0.3 sq mm.

21. Apparatus of any one of claims 1 to 20, wherein the spiral-groove is on the rotor, and the surface of the stator is plain and ungrooved.

22. Apparatus of claim 21, wherein the rotor is male.

23. Apparatus of any one of claims 1 to 22, wherein the apparatus includes a pressure regulation system, which is effective to regulate the pressure in the exit-chamber and the pressure in the entry-chamber.

24. Apparatus of claim 5, wherein the apparatus includes a means for measuring the pressure of the process-fluid, and a pressure regulation system effective to regulate the pressure of the exit-chamber to a proportionate value close to pressure of the process-fluid.

25. Apparatus of any one of claims 1 to 24, further including means for conveying barrier-liquid away from the exit chamber and wherein the means for conveying barrier-liquid to the entry-chamber, and the means for conveying barrier-liquid away from the exit-chamber, are connected together in such a manner as to convey barrier-liquid present in the exit-chamber back into the entry-chamber, whereby the barrier-liquid circulates and re-circulates through the spiral groove.

26. Seal assembly apparatus for rotating shaft having an impeller and mounted for rotation within a housing comprising a stator component and a rotor component adapted for rotation about the axis of rotation of the shaft, characterized in that the apparatus includes the following features, in combination:

the stator and rotor components are annularly and coaxially disposed about the shaft axially adjacent to the impeller and have complementary axially tapered surfaces for fitting together in a male-female configuration, the rotor being secured for rotation with the shaft and the stator being secured to the housing;

one of the complementary surfaces is formed with a continuous spiral groove which extends in a spiral configuration around the surface, the spiral groove having open entry and exit mouths at opposite axial ends, and the complementary surfaces are configured for positively pumping a barrier-fluid for sealing toward the impeller fully across the surface upon rotation of the rotor;

and a means for receiving a barrier fluid from a source of barrier fluid, and for reliably conveying the barrier-fluid to the open entry mouth of the spiral groove.

27. The seal assembly apparatus of claim 26, wherein said complementary surfaces are so configured as to sweep each other in a hydrodynamic-bearing relationship.

28. The seal assembly apparatus of claim 27, wherein the spiral-groove is on the complementary surface of the rotor component and the complementary surface of the stator is plain and ungrooved.

29. The seal assembly apparatus of claim 28, wherein said rotor component is male.

30. The seal assembly apparatus of claim 26, 27, 28 or 29, wherein an angle of axial taper of each said complementary tapered surface is between 7° and 30°.

31. The seal assembly apparatus of claim 26, wherein one of said rotor component and stator component is disposed for axial movement relative to said shaft, and further including an arrangement for axially urging that one against the other such that said complementary surfaces are kept in a tight running clearance relationship.

32. The seal assembly apparatus of claim 31, wherein said arrangement is a mechanical spring.

33. The seal assembly apparatus of any one of claims 26 to 32, wherein the barrier-fluid is water-based.

34. The seal assembly apparatus of any one of claims 26 to 33, wherein the barrier-fluid is drawn from the process-fluid being acted upon by said impeller.

35. The seal assembly apparatus of any one of claims 26 to 33, wherein the barrier fluid is separate from process fluid being acted upon by said impeller.

36. The seal assembly apparatus of any one of claims 26 to 35, including in said housing an exit chamber in fluid communication with the exit mouth of said spiral groove, said exit chamber receiving barrier fluid discharged under pressure from said exit mouth.

37. The seal assembly apparatus of claim 36, wherein said exit chamber is separated from a process chamber in which said impeller rotates by a seal subassembly.

38. The seal assembly apparatus of claim 37, including a pressure regulation system for regulating the pressure of the barrier-fluid in said exit chamber to a proportionate value close to pressure of process fluid in said process chamber.

pressure of the exit-chamber to a proportionate value close to pressure of the process-fluid

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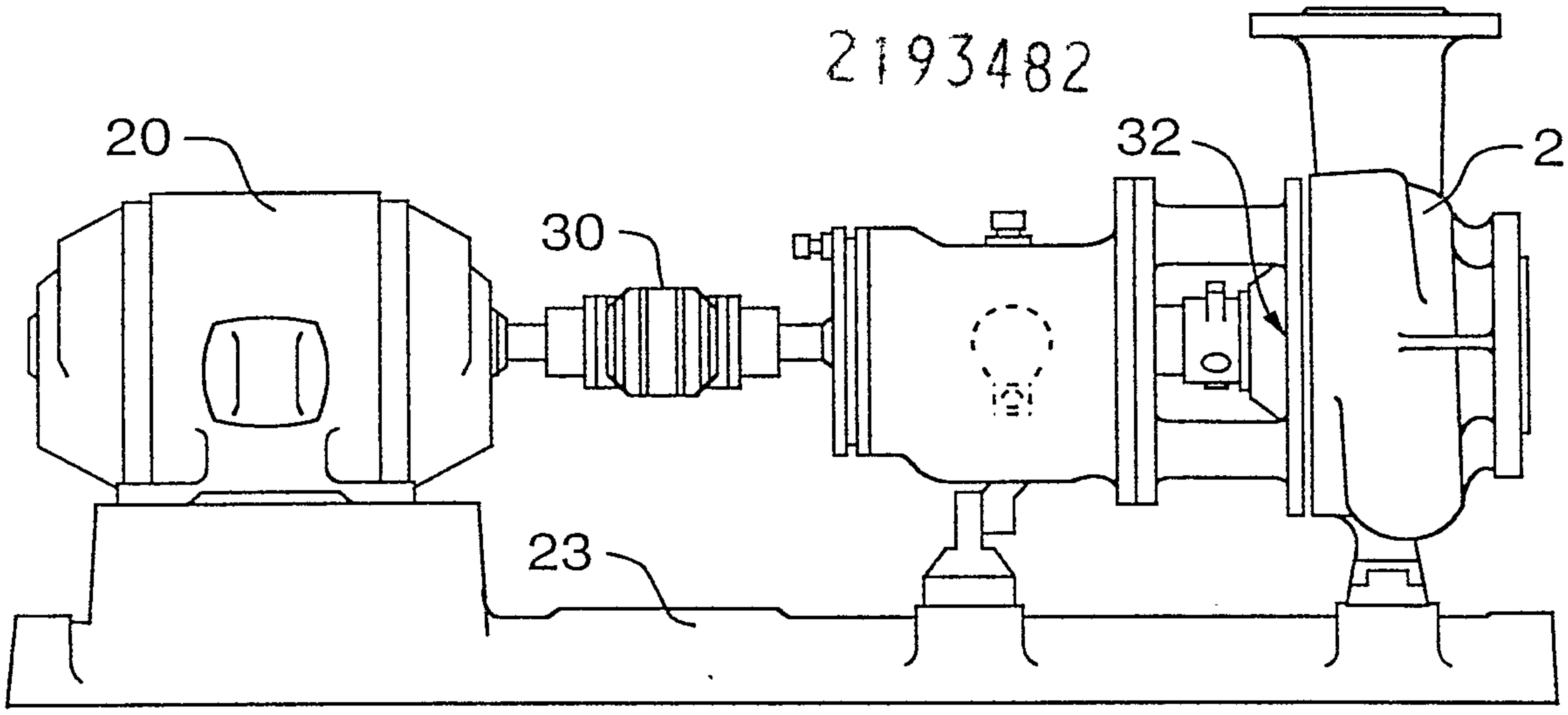

FIG.1.

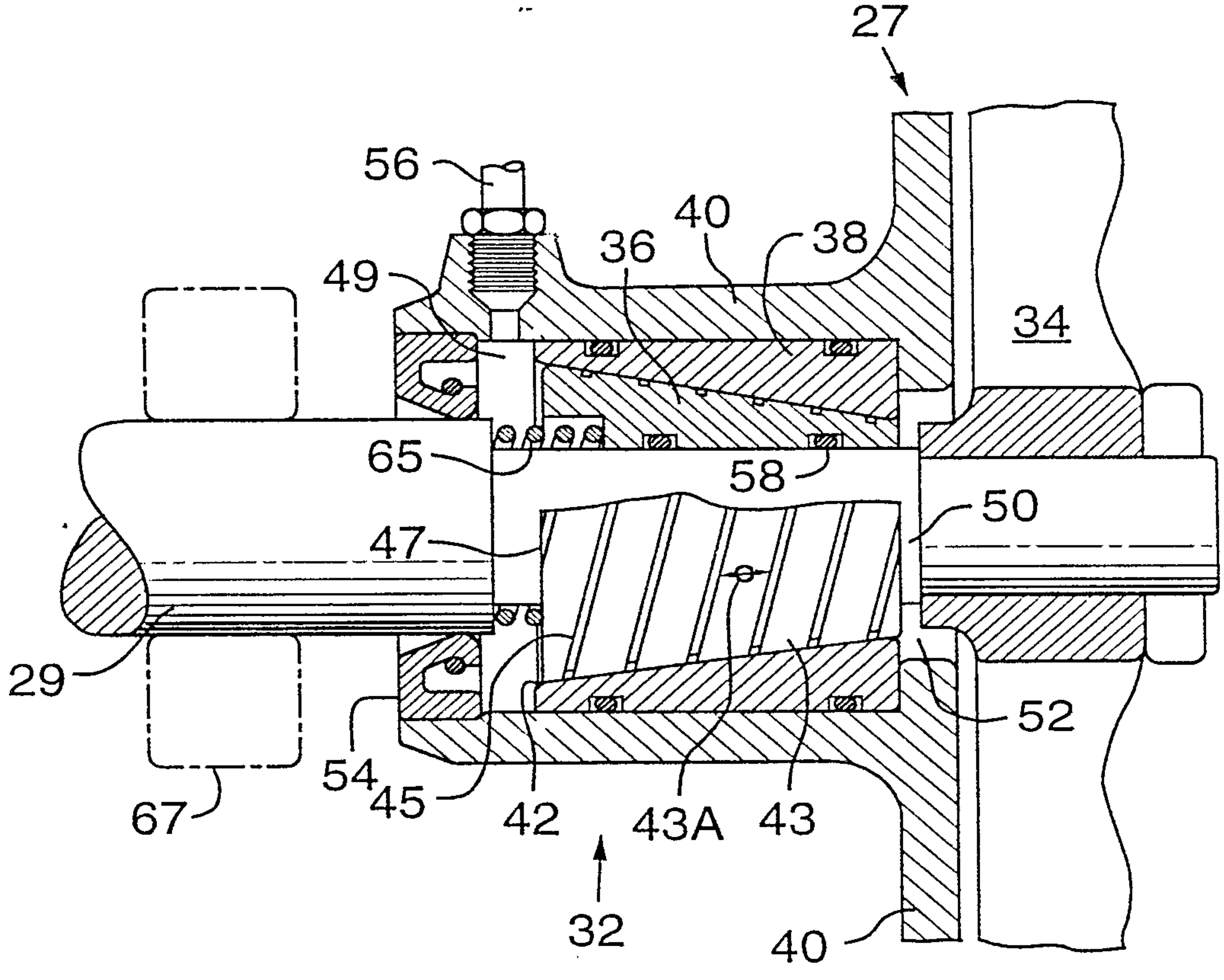

FIG.2.

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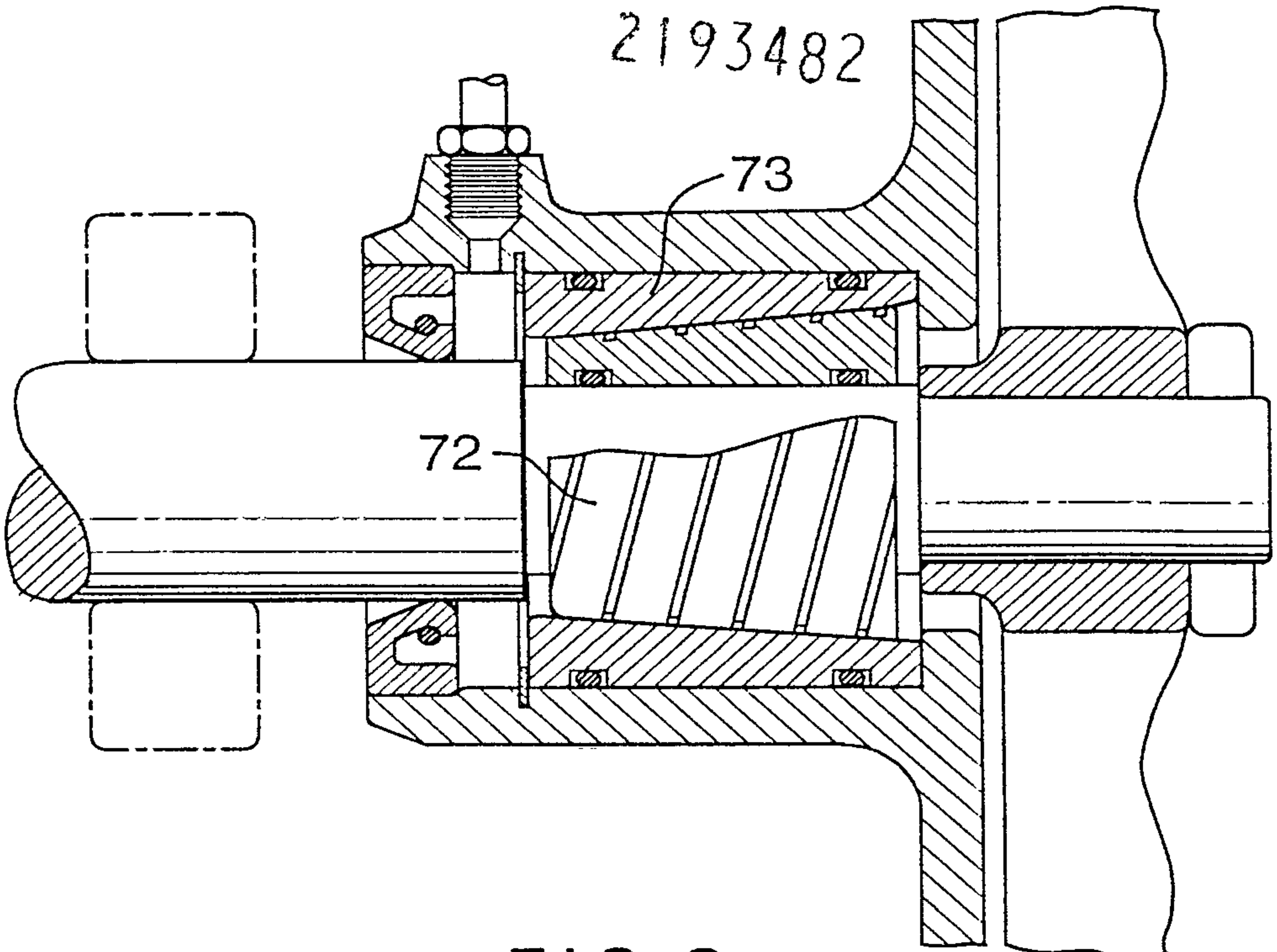

FIG.3.

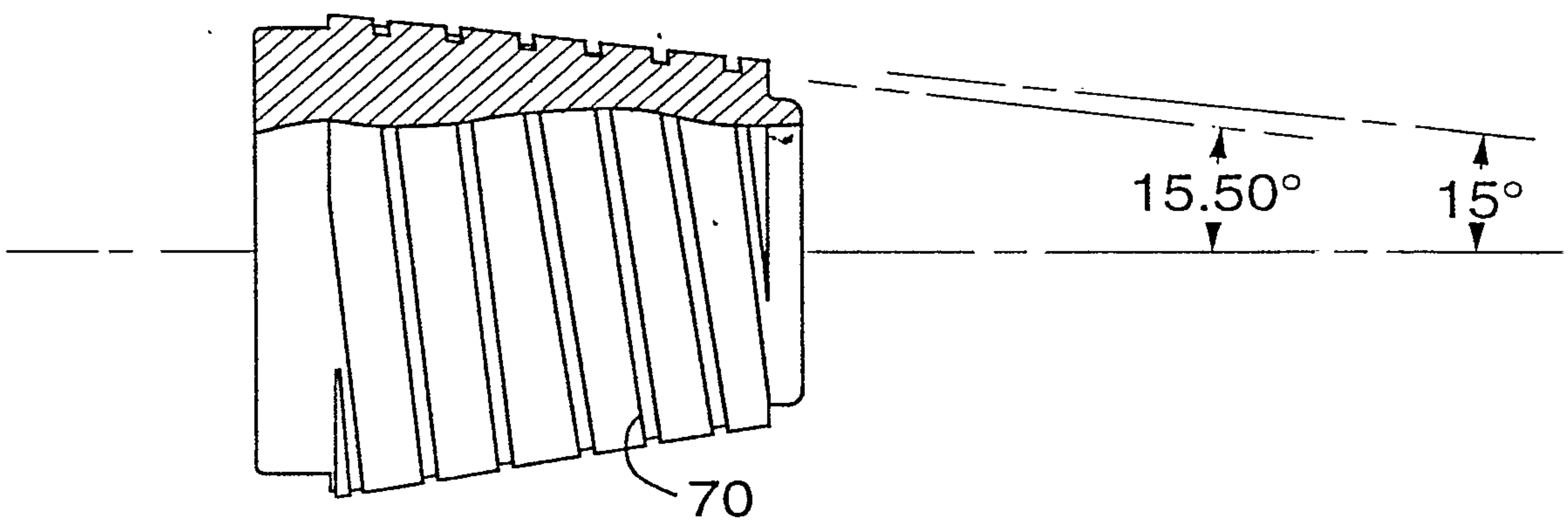

FIG.4.

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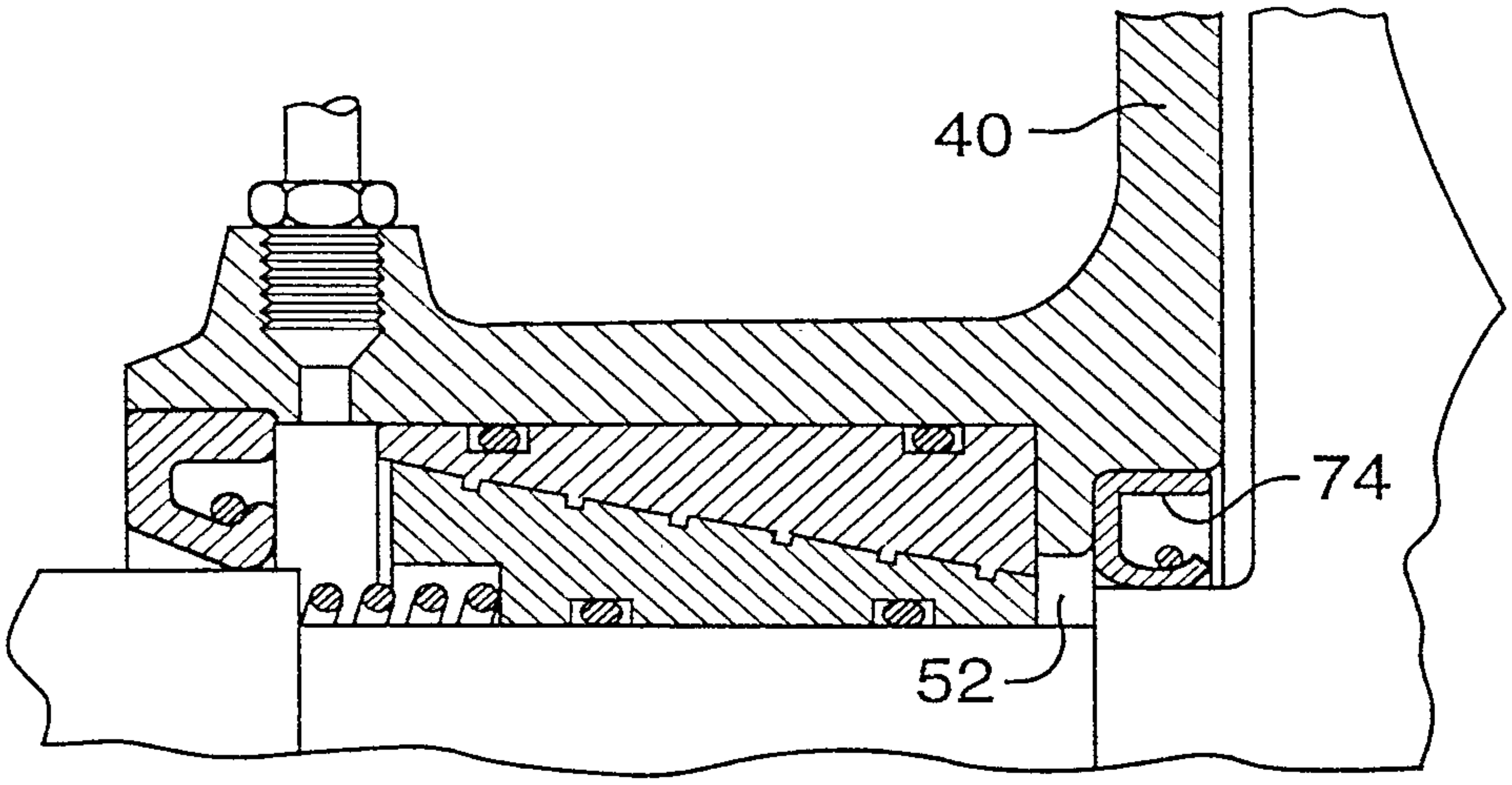


FIG.5.

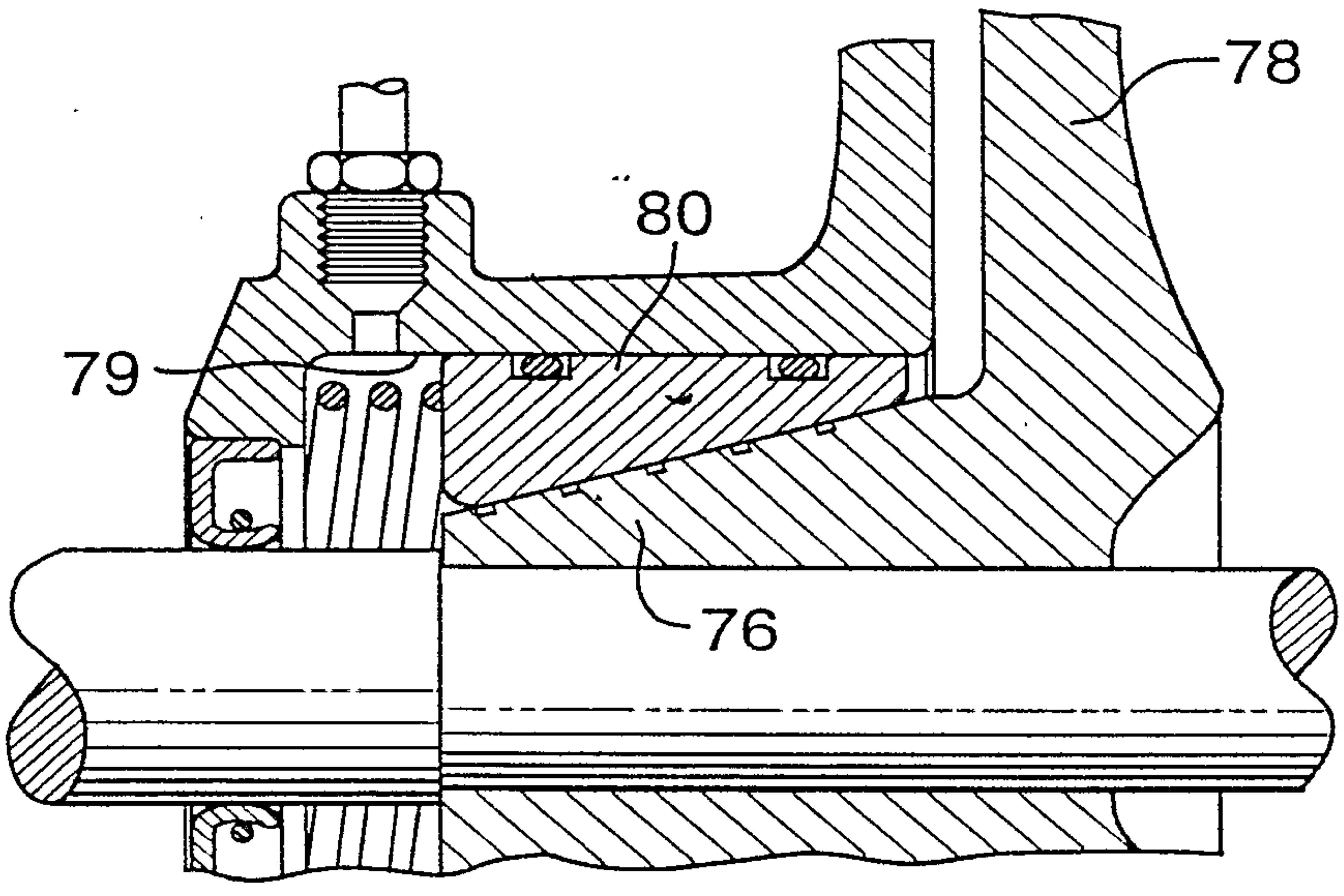
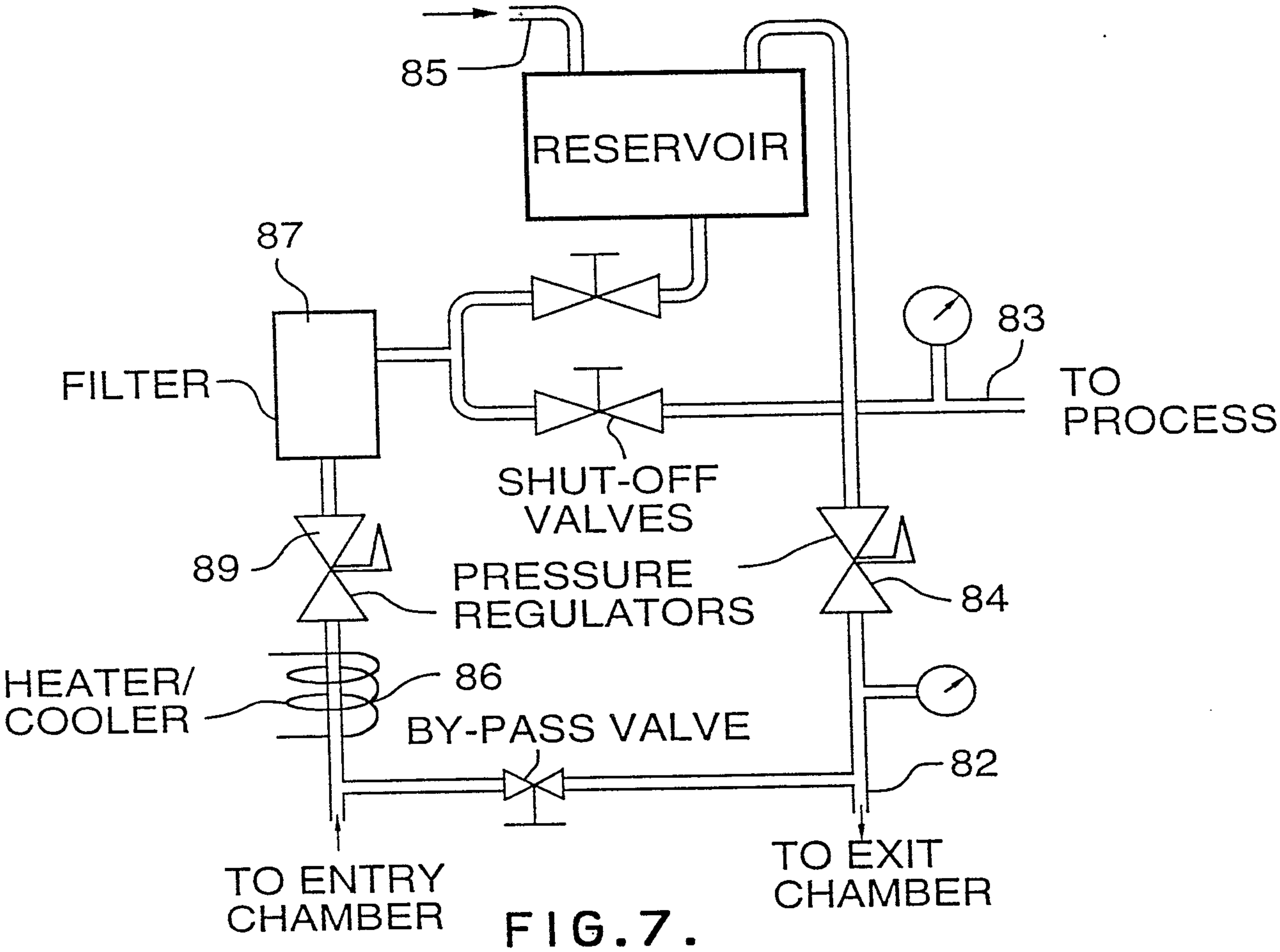


FIG.6.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RESERVOIR
87
FILTER
83
TO PROCESS
SHUT-OFF VALVES
89
PRESSURE REGULATORS
84
HEATER/ COOLER
86
BY-PASS VALVE
82
TO ENTRY CHAMBER
TO EXIT CHAMBER

FIG.7.

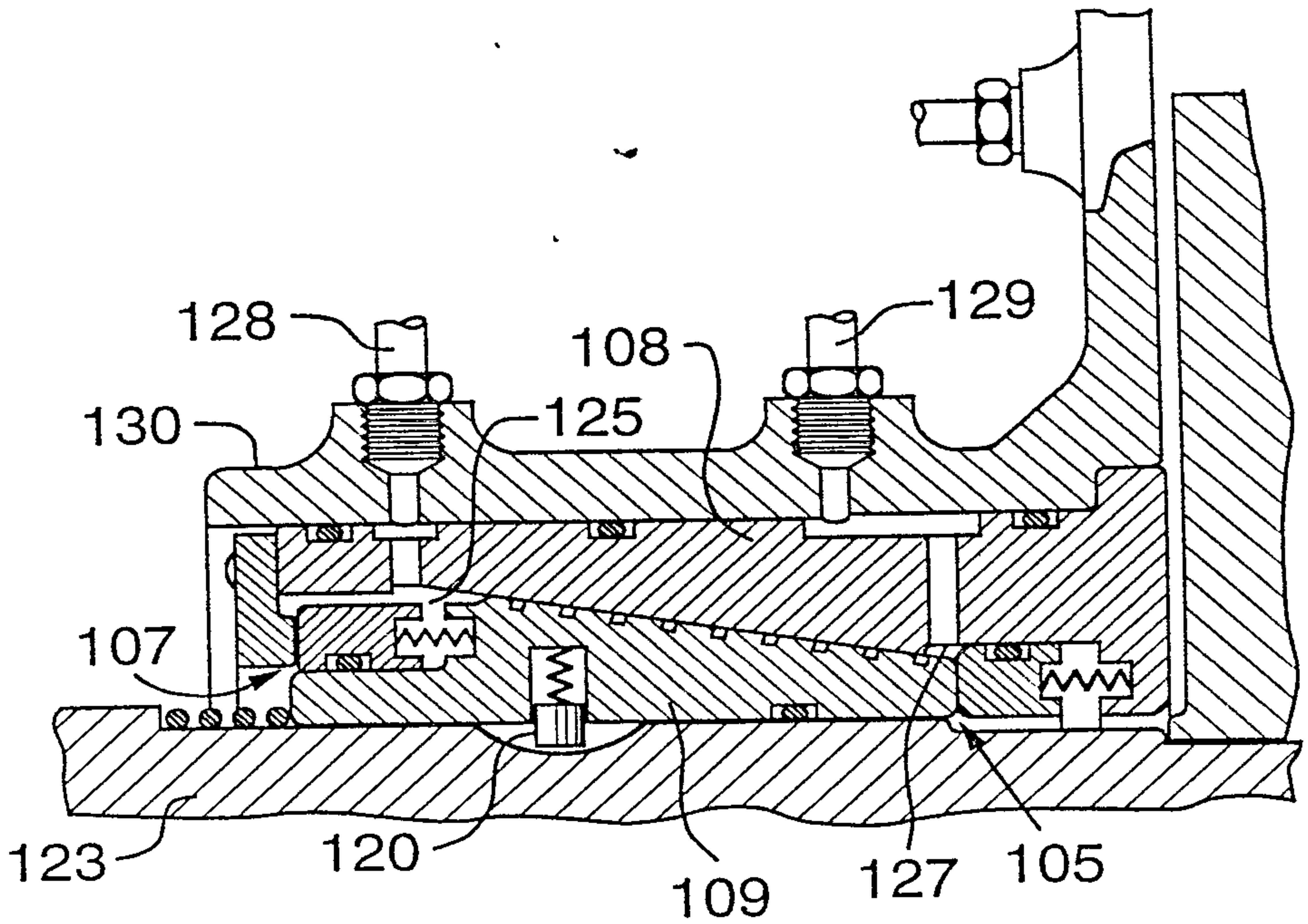
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FIG.8.

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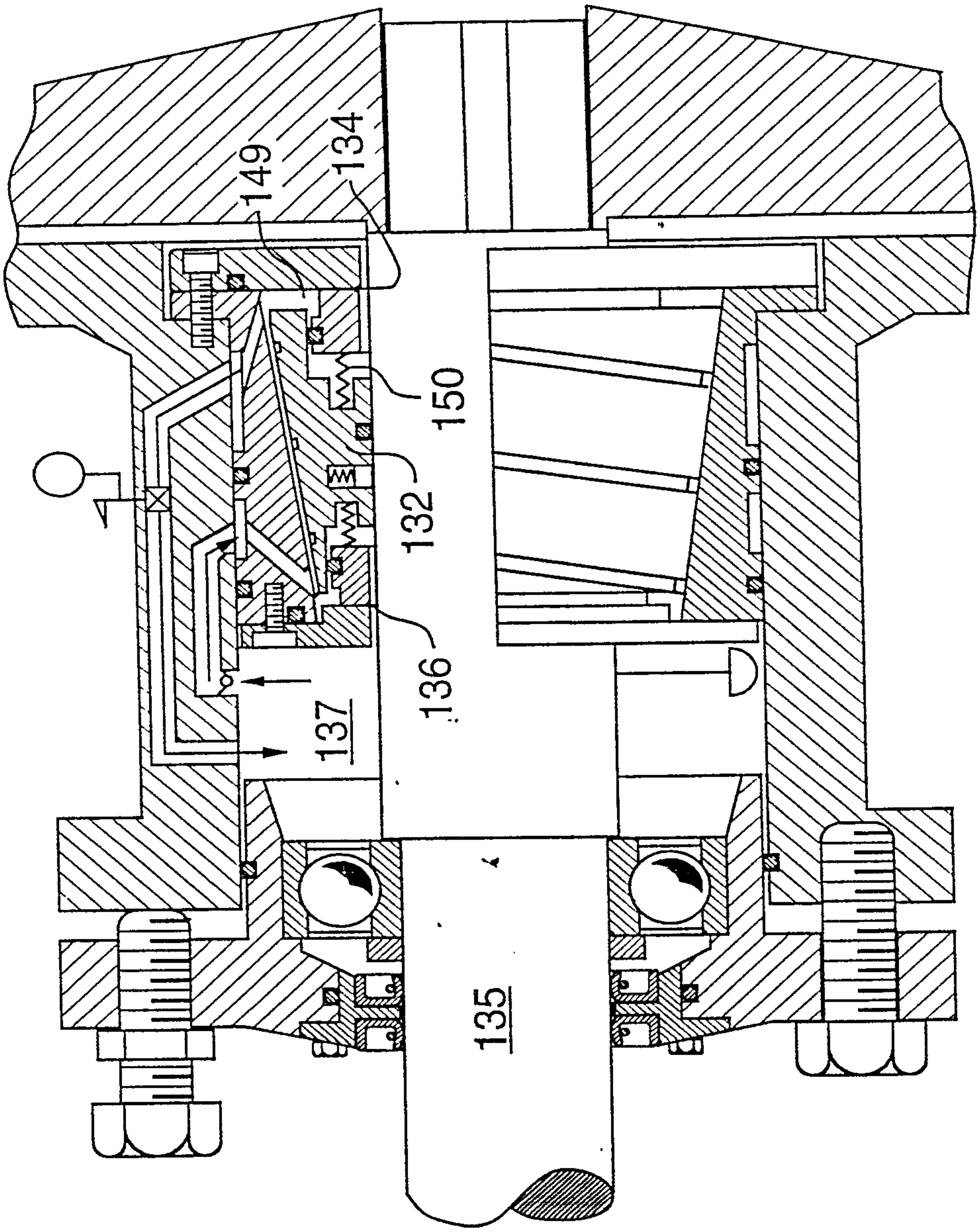

FIG.9.

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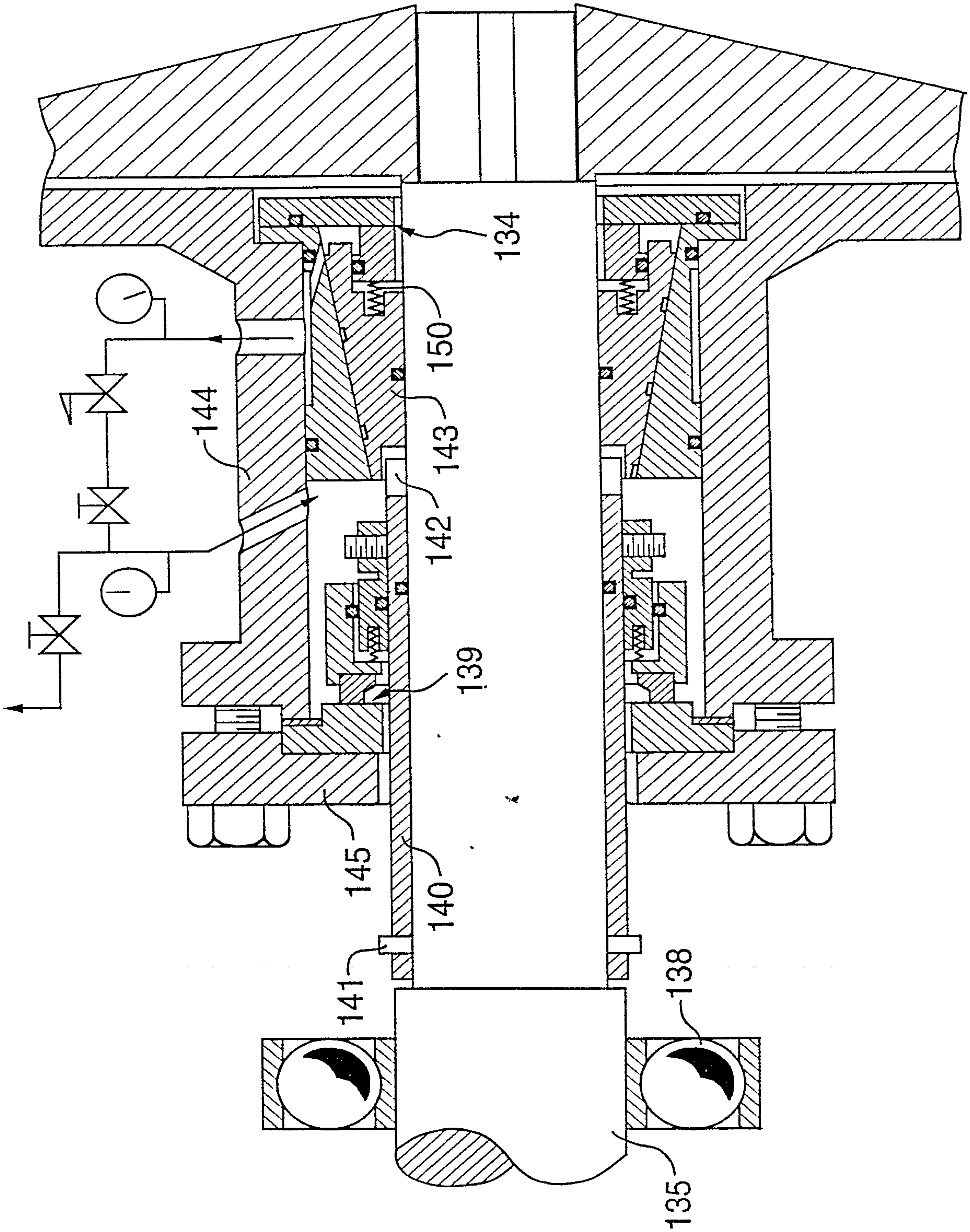

FIG.10.

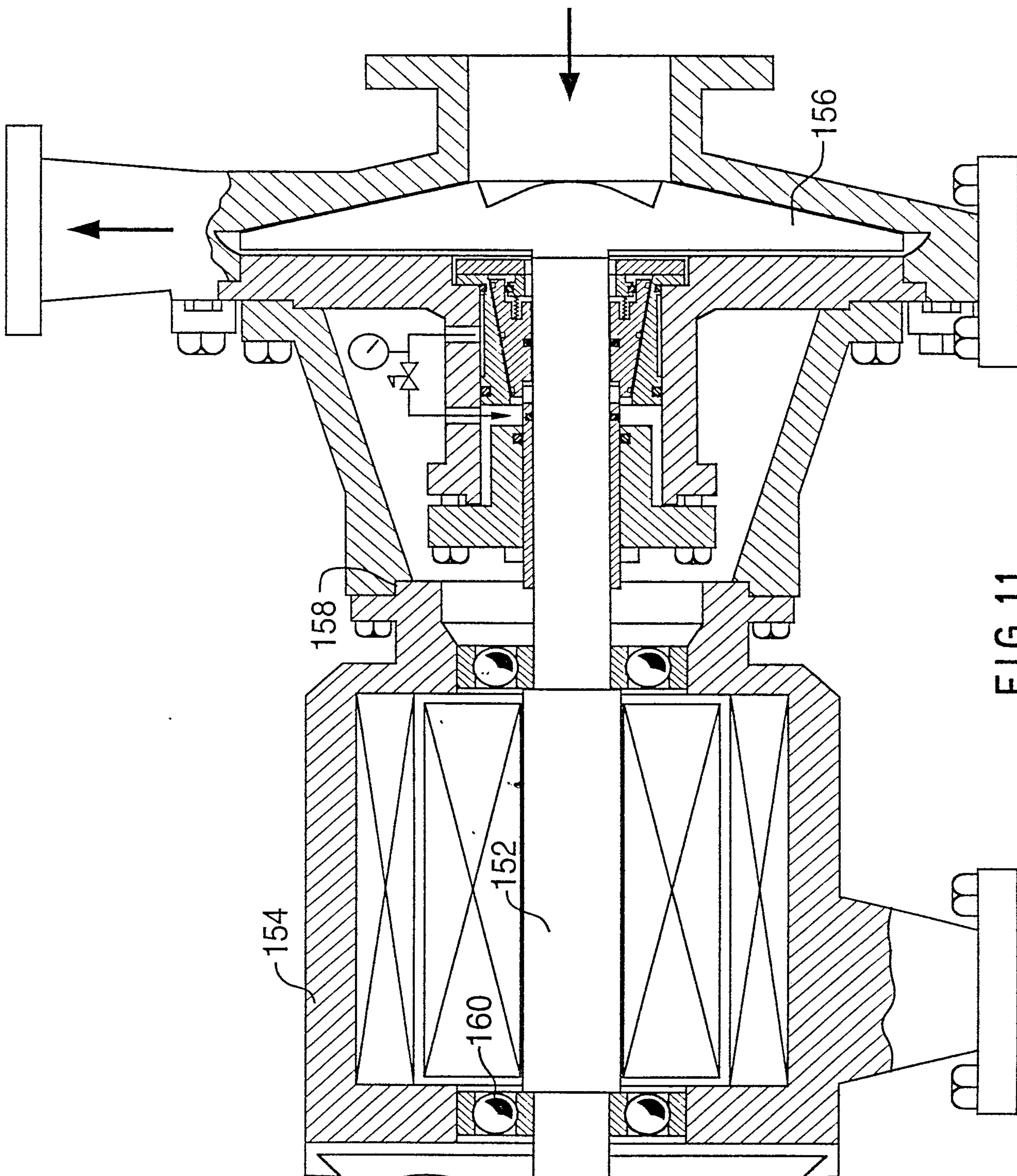

FIG.11.

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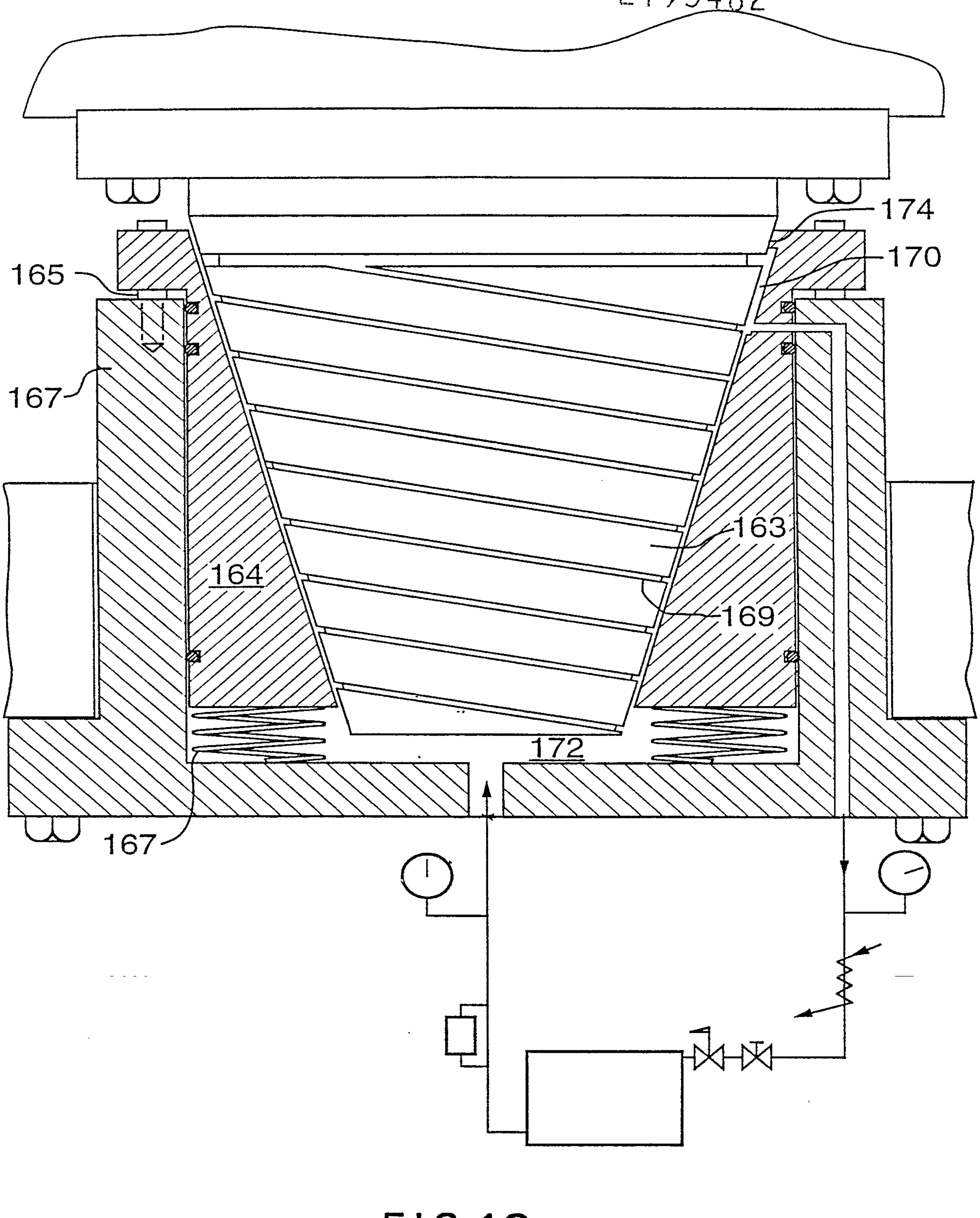


FIG.12.

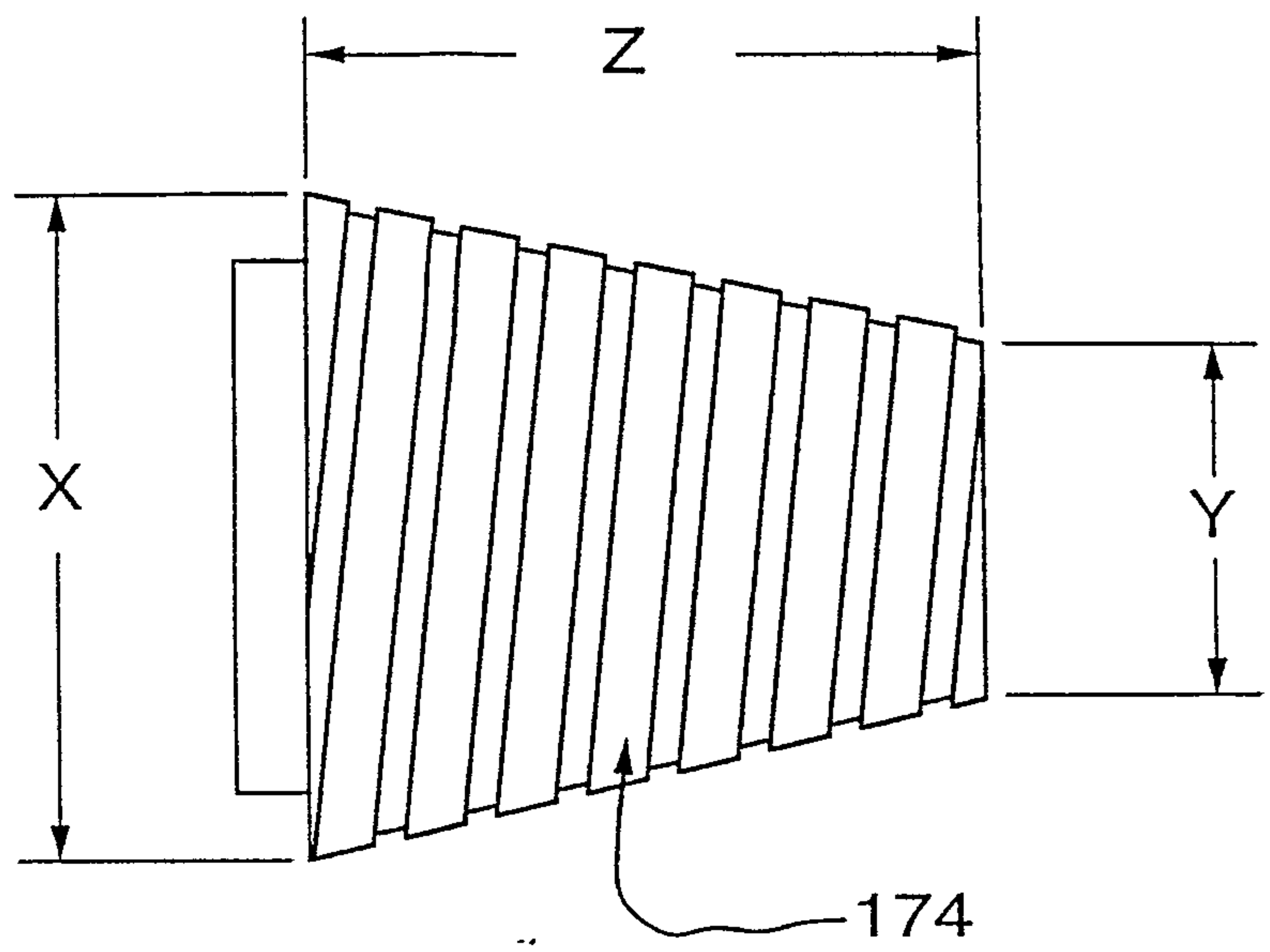
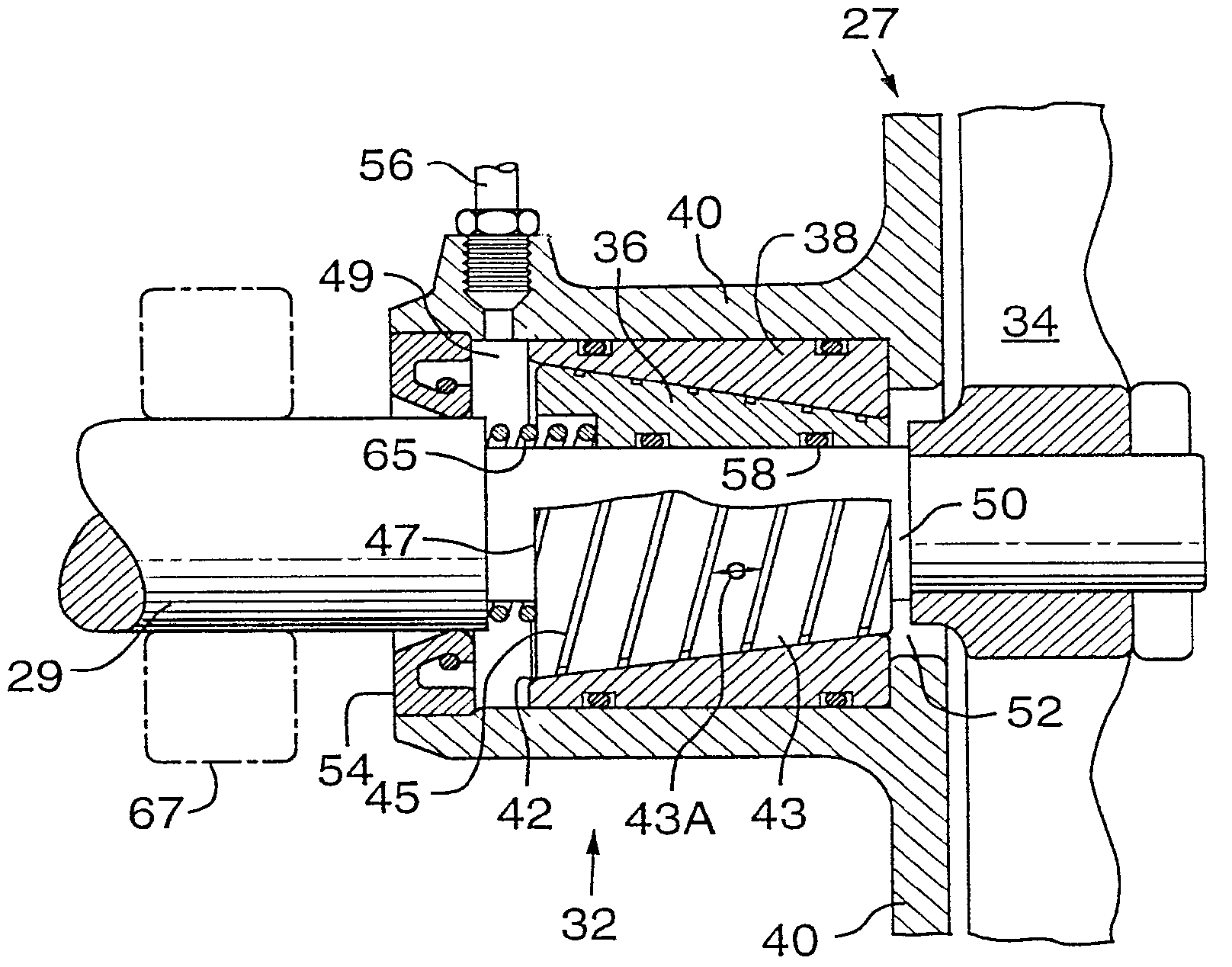


FIG.13.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