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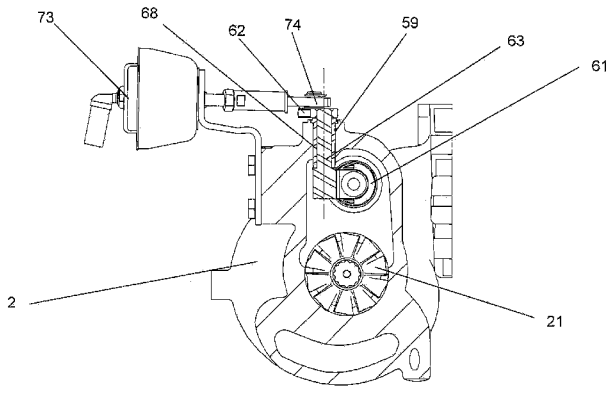


Fig. 1 Prior Art

(57) Abstract: The propensity for gas leakage around a shaft, which extends through a bore which connects volumes of differing pressures, e.g., a turbocharger turbine housing and the ambient air, is minimized with the addition of a pair of seal rings biased by a gas pressure to provide a continuous gas and soot seal.

WO 2012/125387 A2

GAS PRESSURE BIASED SEALING METHOD FOR AN ACTUATING SHAFT

FIELD OF THE INVENTION

This invention addresses the need for an improved shaft sealing design for turbocharger shafts which pass through the walls of the turbocharger housing.

BACKGROUND OF THE INVENTION

Turbochargers are a type of forced induction system. They deliver air, at greater density than would be possible in the normally aspirated configuration, to the engine intake, allowing more fuel to be combusted, thus boosting the engine's horsepower without significantly increasing engine weight. A smaller turbocharged engine, replacing a normally aspirated engine of a larger physical size, will reduce the mass and can reduce the aerodynamic frontal area of the vehicle.

Turbochargers use the exhaust flow from the engine exhaust manifold to drive a turbine wheel (21), which is located in the turbocharger housing (2). Once the exhaust gas has passed through the turbine wheel and the turbine wheel has extracted energy from the exhaust gas, the spent exhaust gas exits the turbocharger turbine housing through the exducer and is ducted to the vehicle downpipe and usually to after-treatment devices such as catalytic converters, particulate traps, and NO_x traps.

In a wastegated turbocharger, the turbine volute is fluidly connected to the turbine exducer by a bypass duct. Flow through the bypass duct is controlled by a wastegate valve (61). Because the inlet of the bypass duct is on the inlet side of the volute, which is upstream of the turbine wheel, and the outlet of the bypass duct is on the exducer side of the volute, which is downstream of the turbine wheel, flow through the bypass duct, when in the bypass mode, bypasses the turbine wheel, thus not powering the turbine wheel. To operate the wastegate, an actuating or control force must be transmitted from outside the turbine housing, through the turbine housing, to the wastegate valve inside the turbine housing. A wastegate pivot shaft extends through the turbine housing. Outside the turbine housing an actuator (73) is connected to a wastegate arm (62) via a linkage (74), and the wastegate arm (62) is connected to the wastegate pivot shaft (63). Inside the turbine housing, the pivot shaft (63) is connected to the wastegate valve (61). Actuating force from the actuator is translated into rotation of the pivot shaft (63), moving the wastegate valve (61) inside of the turbine housing. The wastegate pivot shaft rotates in a cylindrical bushing (68), or directly contacts the turbine housing. Because an annular clearance exists between

the shaft and the bore of the bushing, in which the shaft is located, an escape of hot, toxic exhaust gas and soot from the pressurized turbine housing is possible through this clearance.

Turbine housings experience great temperature gradients and temperature flux. The outside of the turbine housing faces ambient air temperature while the volute surfaces contact exhaust gases ranging from 740°C to 1050°C depending on the fuel used in the engine. It is essential that the actuator, via the translated motions described above, be able to control the wastegate to thereby control flow to the turbine wheel in an accurate, repeatable, non-jamming manner.

A variable turbine geometry (VTG) mechanism is used not only to control the flow of exhaust gas to the turbine wheel but also to control the turbine back pressure required to drive EGR exhaust gas, against a pressure gradient, into the compressor system to be re-admitted into the combustion chamber. The back pressure within the turbine system can be in the region of up to 500kPa. This high pressure inside the turbine stage can result in the escape of exhaust gas to the atmosphere through any apertures or gaps. Passage of exhaust gas through these apertures is usually accompanied by black soot residue on the exit side of the gas escape path. This soot deposit is unwanted from a cosmetic standpoint. This makes exhaust leaks a particularly sensitive concern in vehicles such as ambulances and buses. From an emissions standpoint, the gases which escape from the turbine stage are not captured and treated by the engine/vehicle aftertreatment systems.

Typically some of the leakage of gas and soot through the annulus formed by a shaft rotating within a cylindrical bore was tolerated since the end faces of the bushing are usually in contact with either the inboard flange of the valve arm or the outboard flange or surface of the driving arm of the wastegate control mechanism, thus blocking the leakage some of the time.

Seal means such as seal rings, sometimes also called piston rings, are commonly used within a turbocharger to create a seal between the static bearing housing and the dynamic rotating assembly (i.e., turbine wheel, compressor wheel, and shaft) to control the passage of oil and gas from the bearing housing to both compressor and turbine stages and vice versa. BorgWarner has had seal rings for this purpose in production since at least 1954 when the first turbochargers were mass produced. For a shaft with a seal ring boss of 19mm diameter, rotating at 150,000 RPM, the relative rubbing speed between the seal ring cheek and the side wall of the seal ring groove is of the order of 149,225 mm/sec.

Seal rings, of the variety which are used as described above, are sometimes used as a sealing device for relatively slowly rotating shafts (as compared to the 150,000 RPM

turbocharger rotating assembly seals). These slowly rotating shafts move in rotational speeds of the order of 15 RPM which equates to a relative rubbing speed of 7 to 8mm/sec.

Seal rings, as used in turbochargers, create a seal optimally by contacting part of the side wall of the seal ring against one side wall of the seal ring groove and contacting the outside diameter of the seal ring against the inside diameter of the bore in which the shaft resides. In order for the ring to be assembled to the shaft and then the shaft and ring be assembled into a bore, the depth of the seal ring groove must be such that the ring can collapse in outside diameter (and thus effective circumference and inside diameter) so that the outside diameter of the seal ring can assume approximately the inner diameter of the bore in which it operates. Fig. 2A depicts a seal ring (80) in the naturally expanded condition, albeit assembled to the shaft by forcibly expanding the ring over the diameter of the shaft (63) and then allowing the ring to relax into the groove. As the shaft, with the ring assembled on it, is pushed into the bore of the bushing (68), a chamfer (69) compresses the ring until the outside diameter of the ring can slide in the inside diameter (70) of the bushing. The now-compressed ring seals against the inside diameter of the bushing at any axial position of the shaft.

In this condition, as depicted in Fig. 3, the seal ring (80) can axially reside at any axial position within the confines of the ring groove, the seal ring groove being defined as: the volume between the radial elements of the outside diameter of the shaft (86) and the diameter of the floor (82) of the seal ring groove; and the distance between the inner (83) and outer (81) walls of the seal ring groove. With this definition of the seal ring groove, it can be seen that there always exists a volume under the ring, (i.e., between the inside diameter (84) of the compressed piston ring, and the diameter of the floor (82) of the seal ring groove. There also can exist a volume between the inner wall (83) of the seal ring groove and the proximate wall of the seal ring. On the opposite side of the seal ring groove, there can also exist a volume between the outer wall (81) of the seal ring groove and the proximate wall of the seal ring. Fig. 3 depicts a condition in which the seal ring (80) is somewhat centered between the inner and outer walls (83 and 81) of the seal ring groove, thus allowing passage of gas and soot (86) around the seal ring. Since the axial position of the seal ring is controlled by the friction between the inner diameter of the bore in the bushing, moved by any contact with a side wall of a groove, an optimized sealing condition only exists when the seal ring sidewall is in direct contact with a seal ring groove side wall. In any other axial condition, the leakage path depicted in Fig. 3 exists.

There are a number of patents which teach designs to reduce this leakage by modifying the pressure differential across a plurality of seal rings by introducing a pressure or vacuum between the rings, but potential leakage always exists unless the rings are in direct contact with the side wall(s) of the groove.

Thus it can be seen that there is a need for a design to produce a complete gas seal for "slowly rotating" wastegate and VTG pivot shafts in turbochargers.

SUMMARY OF THE INVENTION

The present invention solves the above problems by introducing gas pressure in between first and second seal rings for an actuator shaft in a turbocharger, the gas pressure forcing a plurality of seal rings to make sealing contact in order to provide a continuous gas and soot seal between a chamber internally pressurized with exhaust gas and soot and the environment outside.

BRIEF DESCRIPTION OF THE DRAWINGS

The present invention is illustrated by way of example and not limitation in the accompanying drawings in which like reference numbers indicate similar parts, and in which:

Fig. 1 depicts a the section for a typical wastegate turbocharger;
Figs. 2A, B depict two sections showing seal ring compression;
Fig. 3 depicts a section view showing gas leakage passage;
Fig. 4 depicts a section view of a the first embodiment of the invention; and
Fig. 5 depicts a magnified view of the first embodiment of the invention.

DETAILED DESCRIPTION OF THE INVENTION

Gas and soot leakage from within a turbocharger to the ambient clean air surrounding a turbocharger is not permitted by engine manufacturers. Turbocharger manufacturers have been using piston rings, or seal rings, to seal gases and oil from communicating between the bearing housing cavity and either or both turbine and compressor stages ever since turbochargers were first in mass production in Diesel engines in the 1950s. So the engineering and application of such a seal is logical for any gas or material seal in less demanding locations on a turbocharger.

A section through a typical assembled seal ring, as depicted in Figs. 2 and 3 viewed perpendicular to the axis of the shaft on which it is assembled, has a narrow rectangular cross section which is partially disposed in either an annular groove in the shaft, or in the

bore in which the shaft rotates, both methods providing a level of sealing between the shaft and its bore. Axially, the seal ring is preferably positioned towards one of the side walls of the seal ring groove; however, there is no mechanical means to effect such a biasing. The groove is typically rectangular in section with the radial depth of the groove greater than the length of corresponding side of the seal ring. In the case of the seal ring groove being in the bore, this allows assembly of the seal ring in the seal ring groove by expansion of the seal ring, thus allowing the mating shaft to pass through the bore of the component. In the case of the seal ring groove being in the shaft, this allows assembly of the seal ring in the seal ring groove in the shaft by contraction of the seal ring, thus allowing the mating shaft and contracted seal ring to pass through the bore of the component until the seal ring is allowed to expand in its assembled configuration, as depicted in Figs. 2A and 2B. The width of the rectangular groove is ideally close to the width of the seal ring to provide optimum sealing. Typically, the closer the widths of the groove and seal ring, the better the sealing capability, but the greater the propensity for the seal ring to seize in the groove.

The design of the seal ring in Fig. 2B is such that the approximate diameter of the relaxed shape of the ring at rest is greater than the diameter of the bore (70) into which it is assembled so, in the assembled state, the spring force of the contracted ring forces the outwards facing surface of the partial circumference of the seal ring against the inwards facing surface of the bore in which it is located. The resulting seal is a contact seal which prevents escape of exhaust gas not only through the cylindrical interface between the bore (70) and the ring (80) but also through the radial interface between the groove wall (81) and the ring (80). This method of sealing is thus different from a “purge seal” where gas under pressure must be constantly be supplied to form an “air dam” to prevent transition of exhaust gas across the zone of high pressure. Since there is less leakage of pressure medium in accordance with the present invention, the pressure medium can be supplied by static means such as a pressure accumulator, or may be supplied by an active means such as a small pneumatic pump or via coupling to the compressor outlet.

Because of the hostile thermal and chemical environment, the pivot shaft is typically not fitted directly to a bore machined directly in the turbine housing, but more often to a stationary bushing or bearing (68) located in a bore in the turbine housing (2). This is in order to better match thermal coefficients of expansion (to maintain close clearances) and to inhibit the galling potential, which is severe, between the material of the pivot shaft and the material of the turbine housing. The bushing is typically axially constrained by a pin (59) through a bore perpendicular to the axis of the bushing, piercing both the outside diameter

of the bushing and the bore in the turbine housing, thus constraining the bushing in the turbine housing.

In the inventive configuration, using a plurality of seal rings, each seal ring mounted in its seal ring groove, on a low speed wastegate or VTG pivot shaft, the inventors developed a design using at least two seal rings, with one ring on each side an annulus defined between the (at least) two rings, into which is introduced a gas under pressure. Each seal ring thus having one side face proximal to the gas filled annulus and one side face distal to the gas filled annulus, in which the seal rings are axially forced apart by the gas pressure introduced into the annulus to create direct contact between an annular distal side face on each of the seal rings and an annular contacting side face on each of the seal ring grooves.

The pressure of the gas introduced into the annulus must exceed the pressure inside the turbine housing in order to force the inner sealing ring against the side of the groove. The pressure required therefore depends on the pressure of the gas inside the turbine housing for a given application, and can be easily determined.

In the first embodiment of the invention, as depicted in Fig. 4 and presented as a magnified view in Fig. 5, two seal rings (80) are disposed around a rotatable pivot shaft (63), and axially located by the side walls of the seal ring grooves into which they are fitted.

Gas (91) under pressure is ducted to a port (90) which fluidly connects the pressurized gas to an annular volume bound radially by the outside diameter of the pivot shaft (63) and the inside diameter of the bore (70) in the bushing (68), and bound axially by the proximal faces of the inner and outer seal rings (80). The gas pressure, applied between the seal rings, forces the seal rings axially apart until the outwards facing surfaces of the seal rings (80) contact the complementary annular sealing surfaces (64 and 66) of the seal ring grooves of the shaft (63), while in circumferential contact with the inwards facing surface of the bore (70) of the bushing (68), thus providing gas and soot sealing around the pivot shaft and the bore into which it is mounted. This inventive seal provides a gas and soot seal between the inside of the turbocharger and the environment external to the turbocharger.

In a variation to the first embodiment of the invention, a plurality of seal rings are installed in an appropriately wider groove in place of the single seal ring per groove as in the first embodiment of the invention.

Thus it can be seen that in the inventive solutions, gas pressure is used to physically move the seal rings until they contact the complementary annular sealing surfaces of their

seal ring grooves blocking passage of exhaust gas and soot to the environment external to the turbocharger.

Now that the invention has been described,

I CLAIM:

1. A turbocharger with

a turbine in a turbocharger housing (2),

a device within the turbocharger housing,

an actuator (73) located outside the turbocharger for actuating the device within the turbocharger housing,

a pivot shaft (63) rotatably mounted in a bore (70) extending through the turbocharger housing for transmitting an actuating movement from the actuating mechanism to the device,

first and second sealing rings (80) provided in at least one circumferential groove in at least one of said pivot shaft and said bore, and

a means for introducing sufficient pressure between said sealing rings to urge the sealing rings axially apart, under operating conditions, each against a respective sealing groove wall (81, 83).

2. The turbocharger as in claim 1, wherein both sealing rings are provided in the same groove.

3. The turbocharger as in claim 1, wherein each sealing ring is provided in a separate groove.

4. The turbocharger as in claim 1, wherein at least one circumferential groove is provided in the pivot shaft.

5. The turbocharger as in claim 1, wherein at least one circumferential groove is provided in the bore.

6. The turbocharger as in claim 5, wherein the bore is provided in a bushing, and wherein said bushing is seated in said turbocharger housing.

7. The turbocharger as in claim 1, wherein said device is a wastegate valve (61).

8. The turbocharger as in claim 1, wherein said turbocharger is provided with a variable geometry turbine (VTG) mechanism, and wherein said device is the VTG mechanism.

9. A method for operating turbocharger with a turbine in a turbocharger housing (2), a device within the turbocharger housing, an actuator (73) located outside the turbocharger for actuating the device within the turbocharger housing, a pivot shaft (63) rotatably mounted in a bore (70) extending through the turbocharger housing for transmitting an actuating movement from the actuating mechanism to the device, first and second sealing rings (80) provided in at least one circumferential groove in at least one of said pivot shaft and said bore, said method comprising:

introducing sufficient pressure between said sealing rings to urge the sealing rings axially apart, under operating conditions, each against a respective sealing groove wall (81, 83).

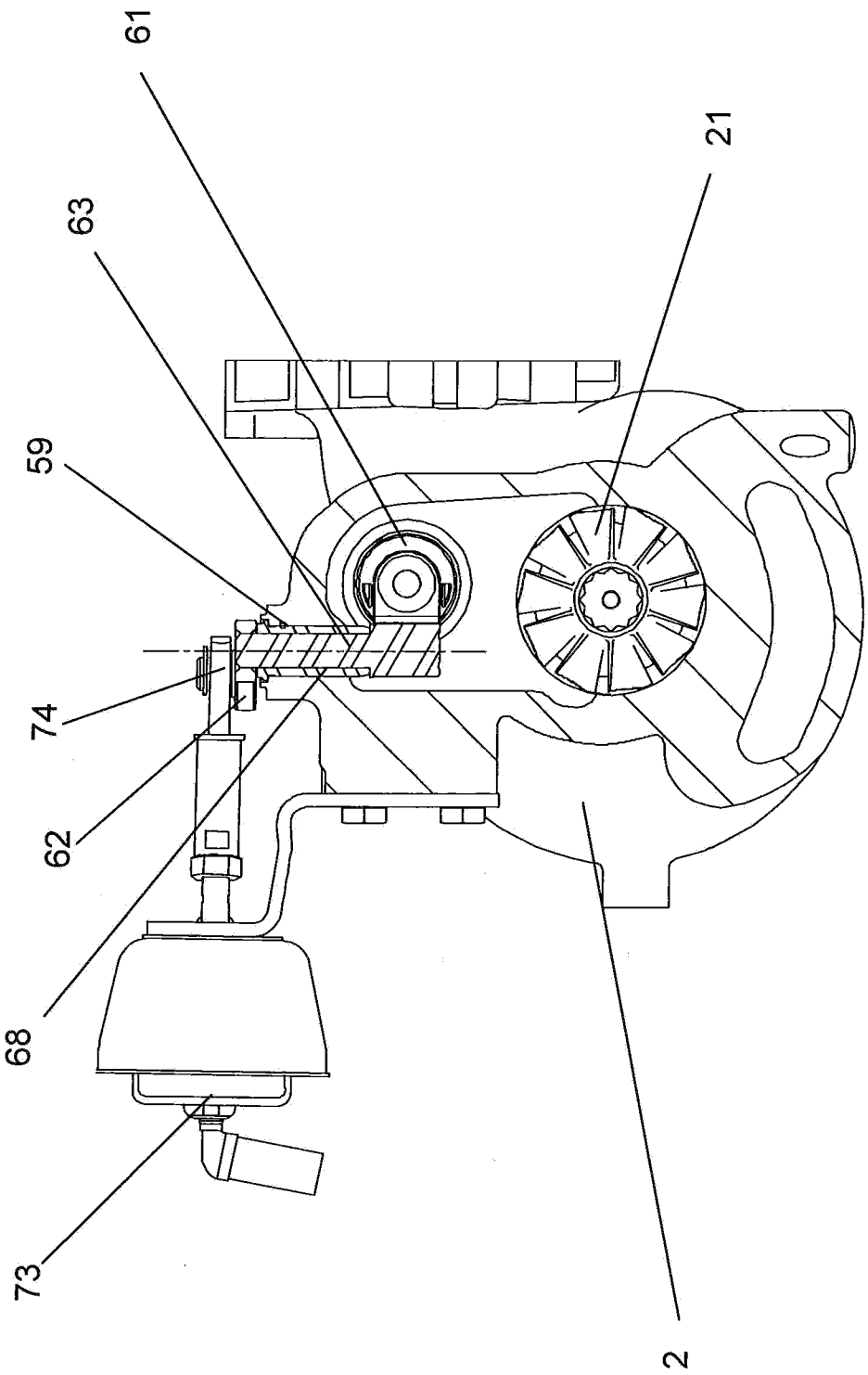


Fig. 1 Prior Art

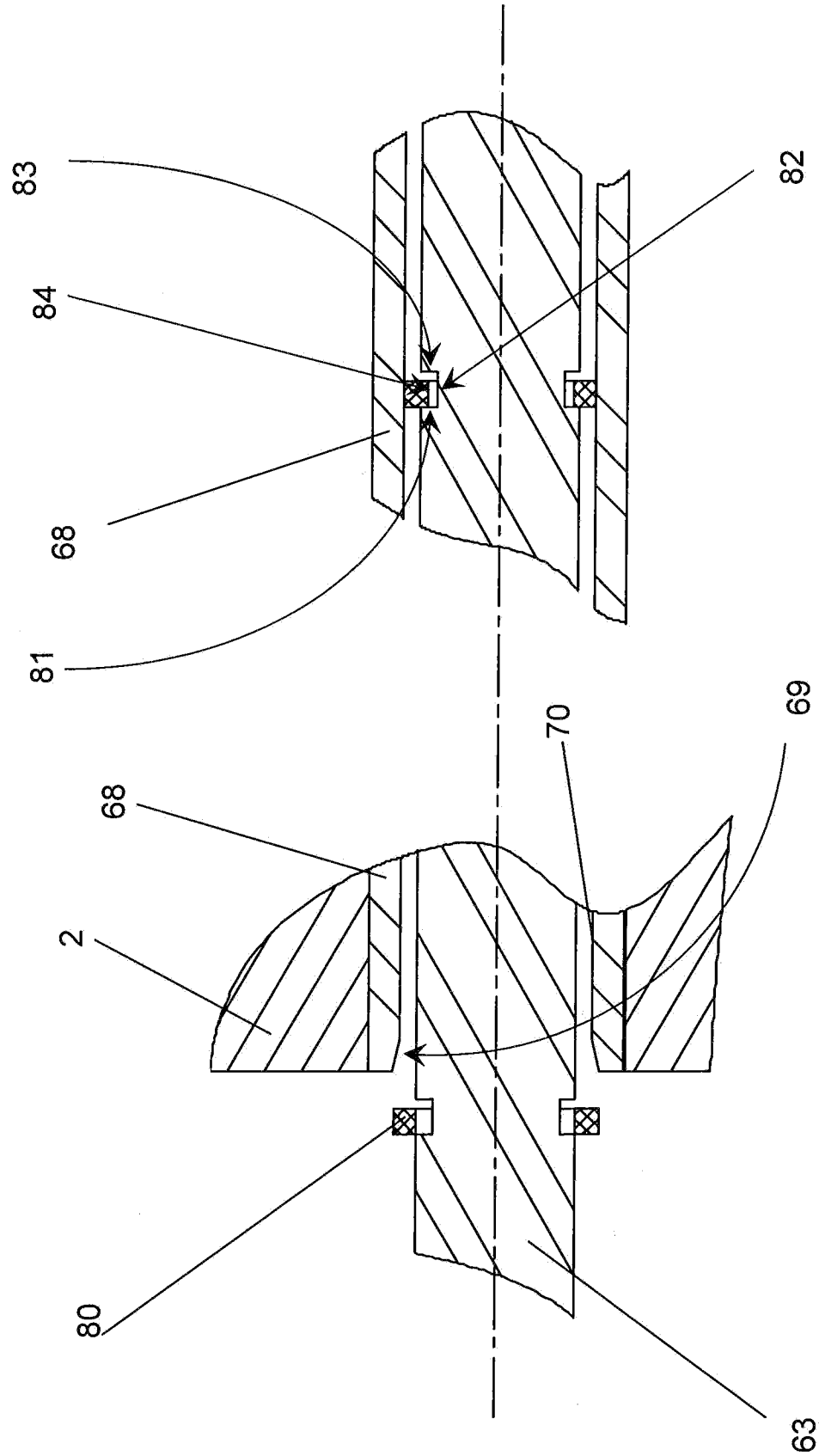


Fig. 2B Prior Art

Fig. 2A Prior Art

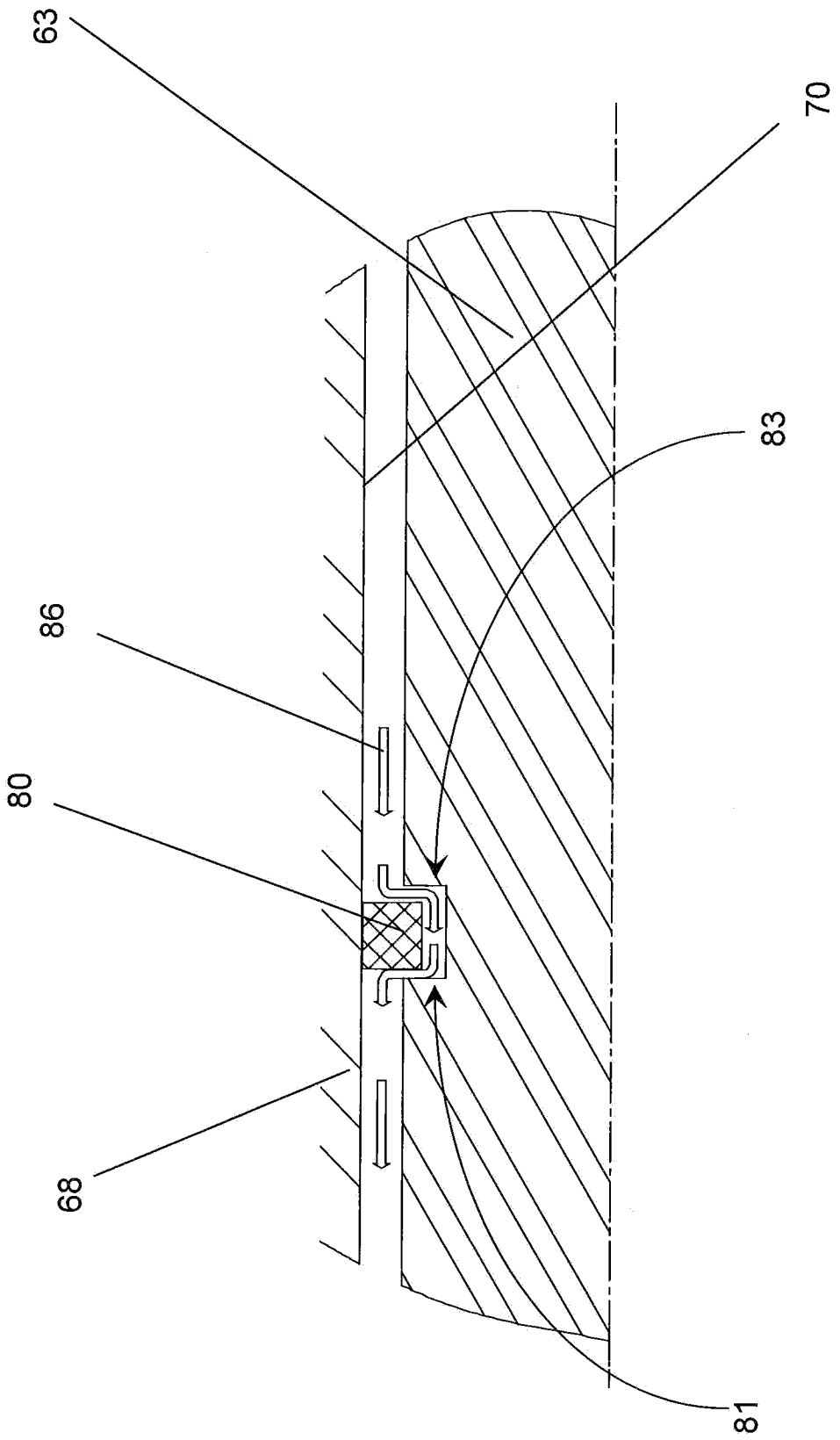


Fig. 3 Prior Art

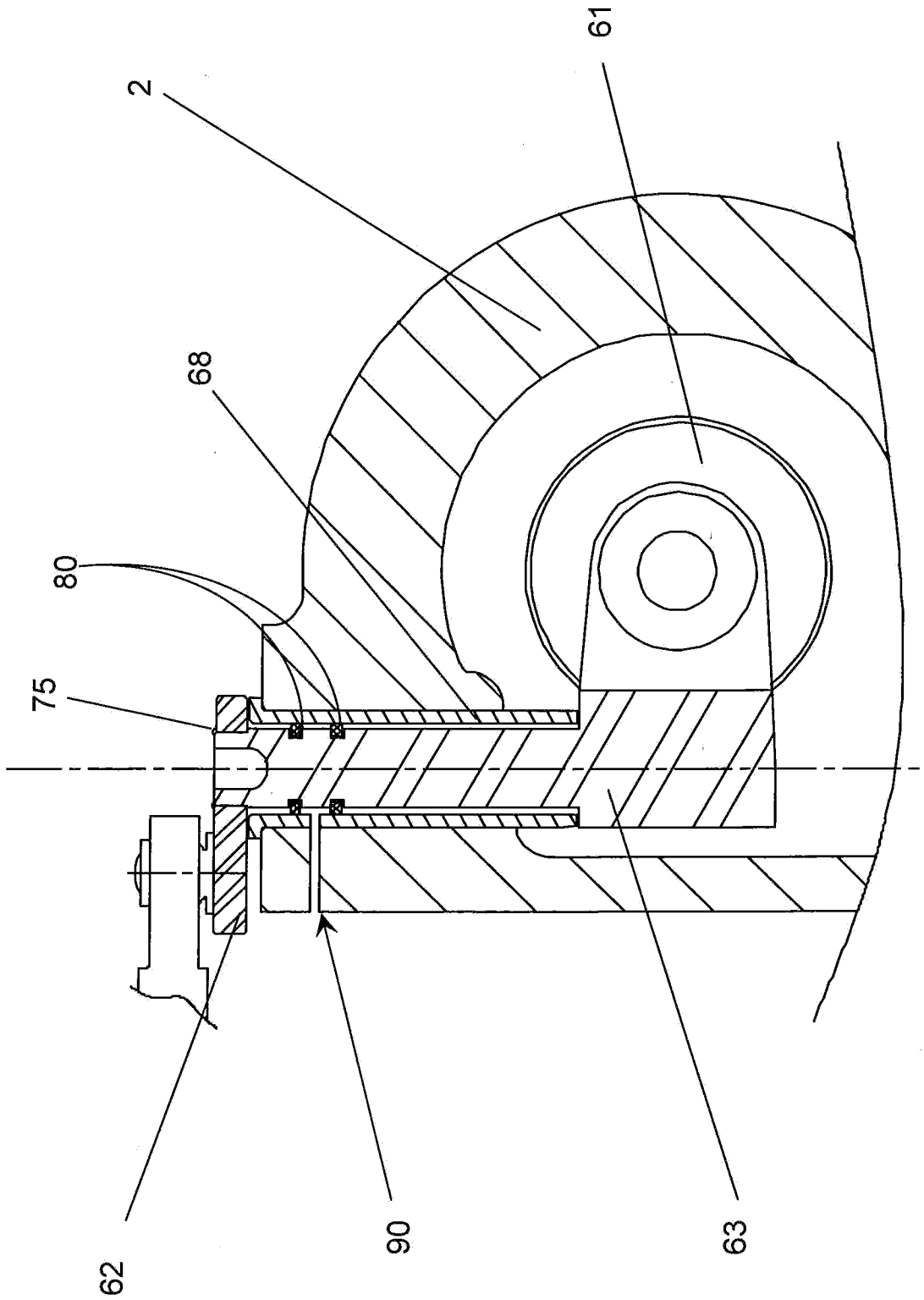


Fig. 4

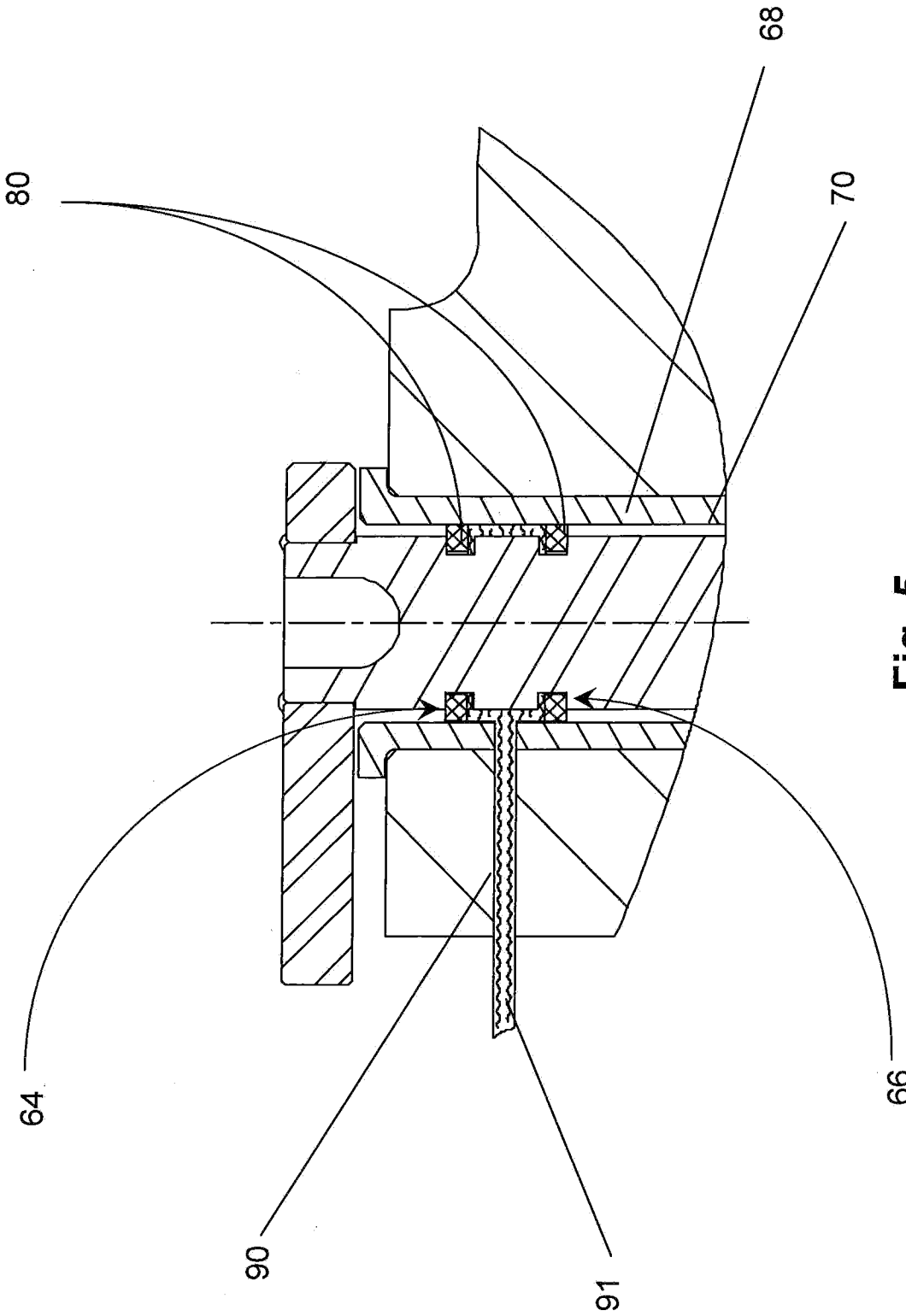


Fig. 5