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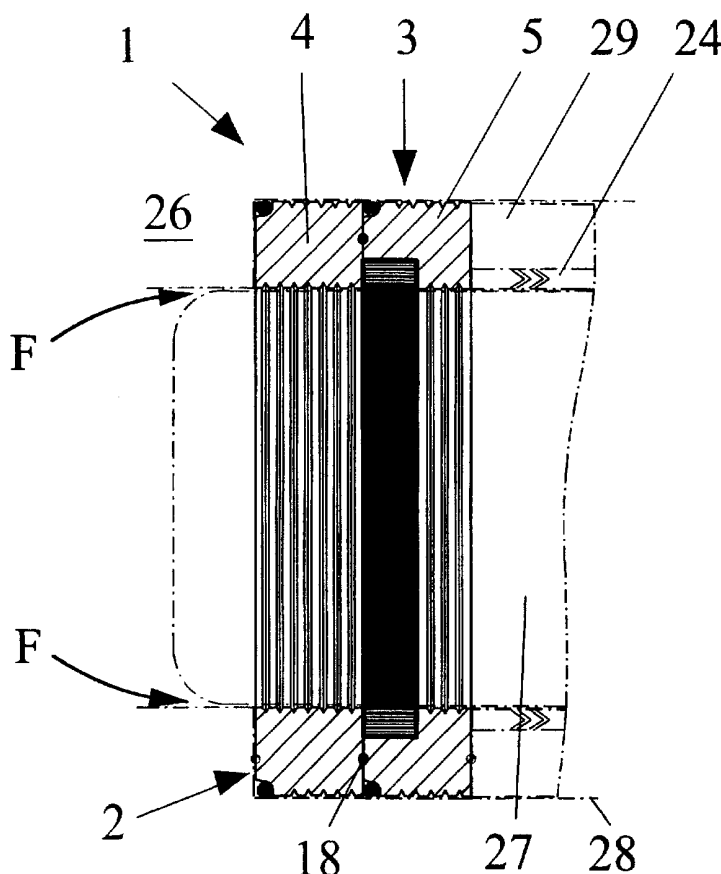
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(54) Title: SEAL ASSEMBLY



(57) Abstract: The present invention relates to a seal assembly (1) for sealing the gap between two relatively moveable parts (26, 27) to prevent fluid leakage from a high pressure side to a low pressure side. Such a seal assembly generally comprises at least one ring seal (2, 3) having a first peripheral surface (8, 9) supported by one (26) of said parts. The at least one seal ring also comprises a second peripheral surface (10, 11) for sealing against said other part (27). According to the invention the seal assembly comprises a primary seal (2) facing the high pressure side and consisting of at least one metal ring (4) having several small peripheral sealing lips (6) provided at its second peripheral surface (10), for performing an initial sealing function and for simultaneously providing a pressure drop thereacross. In addition it comprises a secondary seal (4) provided behind the primary seal, with regard to the high pressure side, and consisting of at least one metal ring (5) being provided with a recess (13) in its second peripheral surface (11), and a first soft sealing ring (14) received in the recess.

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TITLE: SEAL ASSEMBLY**TECHNICAL FIELD**

5 The present invention relates to a seal assembly for sealing the gap between two linearly relatively moveable surfaces, and specifically concerns such a seal assembly for sealing against very high fluid pressures.

BACKGROUND

10 For sealing between two linearly relatively moveable surfaces, such as between the piston and inner cylinder wall of a fluid operated cylinder, conventional ring seals, such as O-rings, lip seals and cup seals, have traditionally been used. For most ordinary purposes such ring seals have been very efficient. However, the traditionally employed materials can only safely perform their sealing function in lower or medium pressure ranges. In applications employing higher pressures or in applications with specific aggressive fluids such conventional seals will
15 degrade very rapidly, and therefore other materials, such as synthetic composite materials, have been suggested for the ring seals used in such environments. Even the synthetic composite materials presently employed in high pressure applications, will not withstand pressures higher than approximately 2000 bar. Above said upper pressure limit ring seals made of synthetic composite materials will easily be damaged due to the high surface pressure
20 that they will be subjected to.

In applications working with high fluid pressures, such as in pressure intensifiers, the high pressure seals often determine the acceptable upper limit for the fluid pressure, and with the presently available seals the practical limit will be set to approximately 2000 bar, as explained
25 above. Although attempts are frequently made to operate pressure intensifiers with higher pressures, such operation suffers from the drawback that the service intervals will be very short. When employed in such higher pressure ranges over 2000 bar the high pressure seals will have a useful life of only a few hours.

30 For all pressure intensifiers it is important that appropriate seals are provided between the high pressure and low pressure sides, to thereby effectively secure the intended increase of the fluid pressure in the high pressure chamber. In applications where different fluids are used on

the high pressure and low pressure sides it is also important to secure that high pressure fluid does not escape past the seals, since it might otherwise mix with the low pressure fluid and adversely affect the efficiency of the intensifier apparatus.

5 In pressure intensifiers working in lower and medium pressure ranges reliable sealing around the periphery of the high pressure piston may be provided using the above discussed traditional ring seals. The problem of fluid leaking past the seals may be taken care of by providing fluid leakage outlets in the cylinder wall, between seals or behind all of the seals, such as disclosed in WO96/41692. Such fluid leakage outlets also serve the purpose of
10 allowing monitoring of the seals and their function.

On the other hand, in the traditional design of pressure intensifiers generating high hydraulic pressures, say in the pressure range of up to approximately 2000 bar, special measures must be taken to prevent unwanted leakage and to increase the useful life of the seals. DE-A-42 06 759
15 describes a high pressure seal provided in the inner end of the high pressure cylinder and intended for sealing the high pressure cylinder from the low pressure cylinder by being biased against the high pressure piston. In this publication it is suggested to compress the high pressure seal axially to thereby obtain a desired radial biasing of the seal against the piston. To achieve this an axially displaceable, spring-biased sleeve is provided in the high pressure
20 cylinder, receiving the high pressure piston. The sleeve is biased towards and directly engages the high pressure seal to radially expand it against the piston in the well known manner. The axial force applied to the seal by the sleeve is also said to be supported and to a certain extent controlled by allowing the high pressure fluid to act directly on the spring end of the sleeve during the forward stroke of the piston.

25 Moreover U.S. Patent No. 4 478 561 discloses a sealing assembly intended to facilitate the correct adjustment of the biasing force of a high pressure seal by providing a packing that may be radially adjusted. In said disclosure it is suggested to perform such radial adjustment of the seal or packing by compressing it by means of an axially adjustable nut.

30 As mentioned above, the sealing of the gap between the high pressure piston and the inner wall of the surrounding high pressure cylinder is a weak point of most pressure intensifiers,

and especially in such an apparatus employed in a high pressure range. The problems encountered in designing such an apparatus may be related first of all to the basic fact that in conventionally designed intensifiers it is not possible to provide a reliable seal between the outer periphery of the high pressure piston and the inner periphery of the high pressure cylinder and to simultaneously obtain an acceptable useful life for the seals. Making the gap between the high pressure piston and the high pressure cylinder large to relieve the seals will inevitably lead to leakage, and biasing the seals heavily against the high pressure piston will unavoidably lead to a premature, increased wear of the seals, reducing their useful life.

Maintaining a desired gap between the high pressure piston and the high pressure cylinder is a complex problem since the cylinder wall of the high pressure cylinder will tend to expand under the influence of the high working pressure. The severity of this problem depends upon the strength of the wall and on the pressure range in which the apparatus is intended to work. In order to strengthen the high pressure cylinder, it has been suggested to pre-stress it by tightly winding wires around the outer surface thereof, see i.a. WO9533928, or by forming the high pressure cylinder of several coaxial casings having pressurized fluid spaces therebetween, GB-A-1 264 497. It should be pointed out though, that in very high pressure ranges of say 5000 - 15000 bar this problem would still exist irrespective of the chosen material, dimension, or pre-stressing of the cylinder. This is due to the fact that the structure of the material will be deformed when exposed to such very high pressures. This problem would become even more apparent when attempting to design an apparatus of the generally described kind so that it becomes slimmer than the traditional design, meaning that its overall dimensions are reduced, since the desire would then be to reduce the thickness of the high pressure cylinder walls. Such an attempt to reduce the wall thickness would increase the expansion of the cylinder walls as the pressure of the high pressure fluid increases. Even with a given pre-stressing of the high pressure seals or of the cylinder itself, the gap between the high pressure piston and cylinder wall will therefore vary with the pressure of said fluid.

RELATED ART

It has been widely accepted for many years to employ labyrinth seals and packings for providing a fluid seal between relatively rotating surfaces, such as between a rotating shaft and a casing. Formerly, labyrinth seals of metallic material were standard seals for many such

applications, but lately labyrinth seals of synthetic material have entered the market. Furthermore, multi-piece labyrinth seals have been most common, that is having at least one ring fixed to the rotating part and at least one ring fixed to the other, stationary part, said rings thereby rotating in relation to each other. Examples of such conventional multi-ring labyrinth seals are disclosed in US Patents Nos. 4 466 620 and 4 706 968. Attempts have also been made to simplify the labyrinth seals and to achieve an efficient sealing with one-piece labyrinth seals being fixed to the stationary part, as is disclosed for instance in US Patent no 5 074 567 and in CH A5 641 883.

Labyrinth seals have to a very great extent been employed in rotating machines in order to prevent i.e. lubricants from escaping from a bearing and to additionally exclude contaminants from entering the bearing and thereby the lubricant from the outside, as is disclosed in the above mentioned US Patents Nos. 4 466 620, 4 706 968 and 5 074 567. Other applications include seals for preventing leakage along the rotating shaft of a compressor, as is disclosed in CH A5 641 883. All such applications involve very moderate pressures.

The effectiveness of the labyrinth seal is based on the fact that centrifugal forces produced by the relative rotation of the two surfaces creates turbulence in the fluid present within the grooves of the seal. This means that the labyrinth seals are traditionally regarded to have an inferior sealing function when the two surfaces are at a standstill or are rotating at low rpm, and to become more effective with increasing rpm. This is specifically the case for labyrinth seals of metallic material, where relatively wide gaps must be maintained between the metallic seal and the rotating part in order to avoid wear on both the seal and on the rotating part.

SUMMARY

The invention overcomes the above discussed problems in an efficient and satisfactory manner.

It is a general object of the present invention to provide an efficient and reliable seal assembly for linearly relatively moveable parts or surfaces. In particular it is desirable to raise the practical upper pressure limit for such a seal, with regard to fluid pressures at which the seal assembly will be operable and will still have an acceptable useful life.

It is a further specific object of the invention to provide an improved pressure intensifier having a high pressure seal that will allow effective operation of the pressure intensifier in very high pressure ranges.

5 It is a further object of the invention to provide an improved method of sealing the gap between linearly relatively moveable parts or surfaces.

Briefly, the invention concerns a seal assembly intended for preventing fluid leakage from a high pressure side to a low pressure side in a fluid operated apparatus comprising two linearly relatively moveable components. Specifically, such a seal assembly serves to prevent fluid
10 from leaking through the gap between the components, and comprises at least one ring seal supported by one of said components and sealing against said other component. The invention suggests providing a combination of a primary seal made of metal and a secondary seal likewise made of metal and carrying one or several "soft seals". In a surface thereof intended
15 for sealing against said other component the primary metal seal is provided with several small sealing lips serving the double purpose of providing an initial sealing function and simultaneously providing a pressure drop across said seal. This means that a significant reduction of the high pressure side fluid pressure will be achieved before the fluid reaches the "soft seal" of the secondary seal, which will therefore only be exposed to a fluid pressure that
20 it will withstand.

In one embodiment of the invention, intended for applications experiencing working pressures exceeding approximately 2000 bar, the primary seal comprises an optional number of separate metal rings, the number of rings being chosen based on the intended maximum pressure for a
25 specific application, such that this pressure may be appropriately reduced before reaching the secondary seal.

In another embodiment of the invention the secondary seal likewise consists of an optional number of metal rings, each carrying a "soft seal". At least some of the secondary seal metal
30 rings are provided with several small sealing lips serving the same purpose as those of the primary seal metal rings, such that the fluid pressure will be further reduced by each such secondary seal metal ring.

In further embodiments of the invention radial seals are provided between each of the primary and secondary seal metal rings, and at least some of the metal rings are provided with small sealing lips also at the surface of the rings being supported in a groove of said one component, thereby also excluding fluid leakage between said one component and the seal assembly.

In yet another embodiment of the invention the metal rings of the primary and secondary seals are formed as an integrated unit, the intended pressure range determining the overall length of the seal assembly as well as the respective lengths of the primary and secondary seal portions of the seal assembly.

A second aspect of the invention relates to an apparatus for generating very high hydraulic fluid pressures in a pressure range above approximately 2000bar, and incorporating a high pressure seal designed in accordance with the above stated general principles of the invention. Said high pressure seal will withstand the very high working fluid pressures of the pressure intensifier and will perform an efficient sealing function.

In one embodiment of this aspect of the invention the inventive seal seals is employed in combination with an arrangement for continuously controlling the gap between the seal and a high pressure piston of the apparatus, so that a condition with a zero-play can be maintained therebetween. This is extremely important for optimizing the effectiveness of such a seal assembly.

A further aspect of the invention relates to a method of sealing the gap between two linearly relatively moveable parts, in accordance with the above stated general principles of the invention.

Preferred further embodiments of the different aspects of the invention are specified in the respective dependent subclaims.

In summary, the present invention provides for the following advantages over the state of the art:

- The seal assembly withstands and works efficiently with significantly higher pressures than conventional seals;
- The useful life of the seal assembly can be significantly increased, and consequently;
- The costs for service and replacement will be drastically lowered, especially in very high pressure ranges;
- The shutdown time for service and repair will be significantly reduced;
- Fluid leakage can be practically excluded for any pressure range; and
- Dimensioning of the seal can be easily adapted to a specified application.

Other advantages offered by the present invention will be readily appreciated upon reading the below detailed description of embodiments of the invention.

BRIEF DESCRIPTION OF THE DRAWINGS

The invention, together with further objects and advantages thereof, may best be understood by making reference to the following description taken together with the accompanying drawings, in which:

Fig. 1 is a side view, in section, illustrating a first embodiment of a high pressure seal assembly according to the present invention,

Fig. 2 is a side view, in section, of the primary seal of the seal assembly illustrated in fig. 1,

Fig. 3A is a side view, in section, illustrating the assembled secondary seal of the seal assembly illustrated in fig. 1,

Fig. 3B is an axial plan view of the secondary seal of fig. 3A,

Fig. 4 is an exploded view of the secondary seal of figs. 3A and 3B with two radial sealing rings,

Fig. 5A illustrates a second embodiment of a seal assembly according to the invention, in an assembled condition,

Fig. 5B is a cross section through a first alternative embodiment of one of the radial seal rings illustrated in figs. 4 and 5A,

Fig. 5C is a cross section through a second alternative embodiment of one of the radial seal rings illustrated in figs. 4 and 5A,

Fig. 5D is an enlarged partial view, in section, illustrating a fixing protrusion of the primary seal illustrated in fig. 5A,

Fig. 6 illustrates a third, alternative embodiment of a seal assembly according to the invention, in an assembled condition,

Fig. 7 illustrates a fourth, alternative embodiment of a seal assembly according to the invention, in an assembled condition,

Fig. 8 is a partial longitudinal section through a pressure intensifier according to the invention equipped with a seal the assembly of the second embodiment,

Fig. 9 is a partial enlarged section of a detail of a pressure intensifier similar to that of fig. 8, but illustrating a modified arrangement for maintaining a minute gap between the seal and the piston, and

Fig. 10 illustrates a fifth, alternative embodiment of a seal assembly according to the invention.

DETAILED DESCRIPTION

The prior art discloses that when setting out to design a fluid seal for linearly moveable parts of a machine working with very high fluid pressure the man skilled in the art has been inclined to try solving the accompanying problems by attempting to optimize the effectiveness of the

above discussed conventional types of ring seals. Specifically such optimization has involved looking for appropriate materials and attempting to improve the technique for biasing the seals against the moving parts.

5 Quite contrary to such an approach, the inventor has chosen to disregard the general design practice within this field and looked for an alternative solution to the problems associated with linear movement and high working pressures. During this work the inventor has obtained astonishing results with a seal assembly configuration containing elements that would traditionally be regarded as quite inappropriate for such applications. Such elements would
10 normally be found in the above related labyrinth seals that are exclusively intended for rotating parts creating the necessary turbulence and that are regarded to be ineffective for sealing against even low pressure fluids at low rotational speeds or at standstill. It is therefore quite surprising that the below described, proposed combination of features leads to the favorable results in the inventive seal assembly that is intended for sealing between linearly
15 moveable parts in high pressure applications. Embodiments of said solution will now be described with reference to the attached drawings.

Figs. 1, 2, 3A, 3B and 4 illustrate a basic embodiment of the seal assembly 1 according to the invention. The assembly is illustrated in an assembled condition in fig. 1. The assembly 1
20 basically consists of a primary or front seal 2 that is provided to the left in fig 1, and a secondary or rear seal 3 that is provided to the right in fig 1. In this specification the terms “front” and “rear” will be used to indicate positions of parts of the seal assembly with relation to the high pressure and low pressure side, respectively of the assembly, meaning that a “front” part is positioned towards the high pressure side of the apparatus compared to a “rear”
25 part.

In the embodiment of the seal assembly 1 illustrated in fig. 1, the primary seal 2 consists of a metal ring 4, preferably of hardened steel. The metal ring 4, which is illustrated separately in fig. 2, has a generally rectangular cross section and is formed with small parallel circumferential sealing lips 19, 6 on its radially outer, or first, and inner, or second, peripheral
30 surfaces 8 and 10, respectively. The circumferential sealing lips 19, 6 are preferably formed by machining parallel grooves in the respective peripheral surfaces prior to hardening of the ring.

The sealing lips 19, 6 perform the double function of providing an initial axial seal against the pressurized fluid on the high pressure side HPS (see fig. 9) and of producing a pressure drop across the ring 4, thereby gradually reducing the fluid pressure towards the secondary or rear seal 3.

Said secondary or rear seal 3, which is illustrated separately in figs. 3A, 3B and 4, consists of a metal ring 5 similar to the primary seal ring, i.e. likewise preferably of hardened steel, but provided with a recess 13 in its inner or second peripheral surface 11 (see fig. 4), adjacent its front side surface 17A. In said recess 13 is positioned a "soft" ring seal 14 that in the illustrated embodiment consists of a glass filled high molecular nylon material, but which, depending upon the specific application, can be manufactured from other synthetic composite materials. In the remaining area of the inner or second peripheral surface 11 as well as in the outer or first peripheral surface 9 the secondary seal metal ring 5 is likewise provided with the small parallel circumferential sealing lips 20, 12. Whereas the main purpose of the small sealing lips 20, 12 is the same as for those of the primary seal 2, i.e. to perform an initial sealing function and to produce a pressure drop, to thereby gradually reduce the pressure in a rearward direction over the secondary seal 3, the specific purpose of the soft seal or seals 14 is to perform a main sealing function for the purpose of stopping fluid from leaking past the seal assembly 1. It is indicated in fig. 1 that a final sealing ring 24 may be provided rearwardly of the secondary seal 3 for providing the final approximately fluid-tight sealing towards the low pressure side LPS (see fig. 9). Said final sealing ring 24 is illustrated as being a synthetic lip seal, but other conventional sealing rings may be employed.

The height of the small sealing lips 6, 12, 19, 20 is dependent partly upon the high pressure side fluid pressure and partly upon the viscosity of the fluid, but will for most high pressure applications be in the order of 0,1-0,2 mm.

The inventive seal assembly 1 is preferably supported in a groove 28 formed in a first 26 of two relatively moveable parts 26, 27. This is illustrated in fig 1 which very schematically indicates that the first 26 of the relatively moveable parts is a cylinder casing and that the other part 27 is a piston or plunger linearly moveable in the cylinder. The first peripheral surface 8, 9 of the front and rear seals 2, 3 faces the bottom of the groove 28 where the

corresponding sealing lips 19, 20 form an axial seal between the seal assembly 1 and said cylinder 26. In the groove 28, behind the secondary seal 3, may be provided a sleeve or bushing 29 supporting the final sealing ring 24. Furthermore, the metal rings 4, 5 of the primary and secondary seals 2, 3 are each provided with a cut-out 21, 22 at the corner thereof formed by the first peripheral surface 8, 9 and the front side surface 16A, 17A facing the high pressure side. A conventional "soft" ring seal 23, such as an O-ring, is provided in each such cutout, serving as a combined radial and axial seal.

The second surfaces 10, 11, the soft seal ring 14 and the final sealing ring 24 form the axial seal between the seal assembly 1 and the moveable piston 27. The metal rings 4, 5 of the front or primary and rear or secondary seals 2, 3 are each provided with a generally semicircular groove 15 in their side surfaces 16B, 17A facing each other, and a ring seal 18 is provided in said facing grooves of the metal rings. In the lower part of fig. 1 it is indicated that similar grooves 15 and ring seals 18 may be provided also in the front side surface 16A of the primary seal metal ring 4, in the rear side surface 17B of the secondary seal metal ring 5 and in the corresponding side surfaces of the groove 28 and the sleeve 29. Said ring seals 18 form an initial radial seal for the seal assembly 1.

In this embodiment the seal assembly 1 is designed primarily for an application with relatively moderate working pressure, with only one metal ring 4, 5 in each of the primary and secondary seals 2, 3, and in such an application the ring seals 18 may be conventional ring seals, such as O-rings. However, the exploded view of fig. 4 illustrates that the ring seal 18 may also be a metal ring seal.

With the seal assembly 1 as described above, the fluid F from the high pressure side will enter the gap between the piston 27 and the primary seal 2 metal ring 4. The inner metal sealing lips 6 do not form an absolute axial seal against the fluid, but will perform an initial sealing function, gradually reducing the pressure of the fluid across the ring 4. As the fluid reaches the secondary seal 3 and its soft sealing ring 14, the pressure will have been reduced to such an extent that the sealing ring 14 will be able to perform its main sealing function, without being degraded by too high pressure. The final fluid-tight seal is then provided by the combination of the small metal sealing lips 12 of the secondary seal 3 metal ring 5 and the final sealing ring 24. High pressure

fluid F will also enter the minute radial gaps between the front side surface 16a of the front metal ring 4 and the corresponding side of the groove 28 in the cylinder 26, and between the front and rear metal rings 4, 5. This radially leaking fluid will be sealed off by the combined action of the radial ring seals 18, and, at the bottom of the groove 28, of the "soft" ring seals 23 and the small outer metal sealing lips 19, 20 on the front and rear metal rings 4, 5. This provides a very effective seal against a fluid of high pressure, and the wear of the seal will be minimized since the pressure will have been reduced significantly before reaching the main leakage preventing seal 14, thereby eliminating both the exposure of the "soft seals" to high surface pressure and the need for heavy radial biasing of said "soft seals".

The above discussed radial ring seals 18 made of metal will also be used in applications with higher pressure, such as in different types of hydraulic cylinders operating with high or very high pressures. Hydraulic cylinders operating with higher fluid pressures, such as the later described pressure intensifier 30 illustrated in fig. 8 and 9, often employ connecting rods extended between front and rear end walls thereof. The connecting rods are provided for clamping the cylinder casing between said end walls, and in such applications the clamping force may be transmitted to the inventive seal 1 provided in the cylinder groove 28, such as by the sleeve 29. In such a case the seal assembly 1 may preferably be provided with an alternative form of radial seal. This is illustrated in figs. 5A-C, of which fig 5A illustrates a further assembled embodiment of the inventive seal assembly 101, designed for applications operating with higher pressure. In this embodiment the secondary seal 103 comprises three metal rings 5 and 5' behind the primary seal 102. Specifically, the last or rearward metal ring 5 corresponds to that of the secondary seal 2 in the first embodiment, whereas the two front metal rings 5' of the secondary seal are each provided with a semicircular groove 15 in each side surface, for receiving a radial ring seal.

Figs. 5b and 5c show two alternative embodiments of the radial ring seal. In both alternatives the radial ring seals 18', 18'' have an elongated shape in the axial direction of the seal assembly 101 before being clamped in the cylinder. In other words the ring seals 18' 18'' have a width larger than the combined depth of the grooves 15 of two adjacent metal rings 4, 4', 5, 5'. The ring seals 18' 18'' consist of a soft metal, such as mild steel, or alternatively a composite material, and are deformable into the grooves 15 by said clamping force applied by connection rods. This provides an excellent radial seal between the two adjacent metal rings 4, 4', 5, 5'. It will be

obvious that generally elongated shapes may be used for the radial ring seals, other than those specifically illustrated herein.

5 The primary seal 102 consists of a metal ring 4' that is identical to that of the first embodiment, except that it is provided with an alternative type of radial seal 25 that likewise takes advantage of said above described clamping force, or another comparative force, and that is illustrated in a larger scale in fig. 5D. Specifically, the metal ring 4' of the primary seal 102 is formed having a circular protrusion 25 on its side 16A' facing the high pressure side, said protrusion having a generally triangular, outwardly pointed cross section. The protrusion 25 is
10 intended to be pressed into a front side surface 28A of the groove 28 of said one part 26 upon assembly, and upon application of said clamping force, to thereby fix said metal ring in position and to provide an efficient radial seal.

15 Figs. 6 and 7 show two further embodiments of the inventive seal assembly 201 and 301, for the purpose of illustrating the modular structure of the seal assembly that makes it possible to design and build a seal adapted to the conditions of every specific application. In the embodiment of fig. 6 the seal assembly 201 is built for a higher pressure range than the previous embodiments, and comprises a primary seal 202 having two metal rings 4' and 4'', of which the front ring 4' is as previously described, whereas the rear ring 4'' has radial ring seals 18, 18' or 18'' provided in both of its side surfaces. The secondary seal 203 on the other hand consists
20 of five consecutive metal rings of which the first four rings 5' are of the kind described in connection with fig. 5A. The last or rearmost ring 5'' lacks the small sealing lips on its inner and outer peripheral surfaces and has been included to exemplify that in certain applications the soft sealing ring 14 of said rearmost metal ring provides the final fluid-tight seal.

25 Fig. 7 illustrates a further embodiment of the inventive seal assembly 301, which is built for an even higher pressure range and where the primary seal 302 comprises a total of three metal rings, namely one front metal ring 4' as previously described, followed by two metal rings 4'' like the rear ring of fig. 6. The secondary seal 303 is identical to that of fig. 6, except that the rearmost metal ring 5 is here provided with the small sealing lips like in the first embodiment
30 of fig. 1, and is followed by the final sealing ring 24. Compared to the embodiment of fig. 6 this embodiment may be designed for an application using a fluid having a lower viscosity.

The seal assembly of the invention may be employed in a wide variety of fluid operated apparatuses comprising two linearly relatively moveable, generally cylindrical components. However at present a preferred application of the inventive seal assembly is in a pressure intensifier operating in high and very high pressure ranges, specifically in pressure ranges above approximately 2000 bar, such as for use in hydroforming equipment. Fig. 8 illustrates an embodiment of a pressure intensifier 30 according to the invention, incorporating a seal assembly as described above. The pressure intensifier 30 of fig. 8 basically consists of a low pressure cylinder 32 provided to the right in fig. 8 and a high pressure cylinder 31 provided to the left in fig 8. It should also be emphasized that the term "low pressure" as used in the description of the pressure intensifier - i.e. "low pressure fluid", "low pressure chamber" etc., refers to a pressure that is low compared to the high pressure output from the intensifier, but that is not necessarily low in other respects or applications.

A rear end wall 34 closes the low pressure cylinder 32 outwardly and a front end wall 35 closes the high pressure cylinder 31 outwardly. The low pressure and high pressure cylinders 32 and 31 are separated by an intermediate wall 37 provided with a central, through bore 37A for sealingly receiving an axially displaceable high pressure piston 43 firmly attached to a low pressure piston 42, as is well known in the art.

The low pressure cylinder 32, which is illustrated with a portion thereof cut away, is clamped between the rear end wall 34 and the intermediate wall 37 by means of first connecting rods 40 being evenly distributed around the outer circumference of the cylinder 32, although only two are visible in fig. 8. The rods 40 are passed through bores (not shown) in the end wall 34 and their threaded front ends are screwed into corresponding threaded blind bores (not shown) in the intermediate wall 37. The likewise threaded rear ends of the rods 40 extend out through corresponding bores in the rear end wall 34 and are engaged by nuts 41 tightened against the end wall 34.

The low pressure cylinder 32 displaceably receives the low pressure piston 42 which is provided with the appropriate seals 42A, such as O-rings, at its outer circumference for sealingly engaging the inner wall of the cylinder 32. By means of the piston 42 the low pressure cylinder 32 is divided into a rear working chamber 32A and a front return chamber

32B. Inlet-outlet openings 49, 50 for low pressure working fluid are provided in the rear wall 34 and in the intermediate wall 37, respectively and serve to supply working fluid to the working chamber 32A and return fluid to the return chamber 32B, respectively.

5 As was mentioned above, the low pressure piston 42 carries a high pressure piston 43 protruding from the front face thereof and extending into the central bore 37A of the intermediate wall 37. Said central bore 37A is provided with a low pressure seal 51 surrounding and sealingly engaging the high pressure piston 43 and positioned in the rear portion of the bore 37A, closest to the low pressure cylinder 32. In the illustrated embodiment the low
10 pressure seal 51 is a multi-lip seal of a synthetic composite material, but other standard seals suitable for the pressures on the low pressure side may likewise be employed. In front of the low pressure seal 51, adjacent the latter, is positioned a distribution ring or bushing 52 communicating with a low pressure fluid inlet 59, through which fluid is introduced into the high pressure cylinder 31. In front of the distribution ring 52 is provided the further bushing
15 29 and the multi-lip seal 24 of fig. 1, serving as a final seal against a high pressure chamber 46. The high pressure chamber 46 is formed in the high pressure cylinder 31 and partly in the bore 37A in the intermediate wall 37, in front of the high pressure piston 43, and by a central bore 35A in the front end wall 35.

20 The high pressure piston 43 is firmly attached to the low pressure piston 42, and may preferably be secured with a press fit in a central opening provided in the low pressure piston. The front wall 35 of the high pressure cylinder 3 is provided with a central high pressure outlet 54 through which a high pressure fluid consumer (not shown) is connected to the portion of the high pressure chamber 46 formed by the bore 35A within the front end wall 35.
25 In the illustrated embodiment the front end wall 35 is provided with an adapter 36 for direct attachment to the consumer by means of bolts. The outer end of the adapter 36 is provided with appropriate seals, ordinary O-rings will be sufficient in most cases, for providing an appropriate sealing against the consumer. Such an embodiment where the pressure intensifier 30 is "docked" directly in the consumer is preferred in many applications with very high
30 pressures, such as in the hydroforming techniques, since it will eliminate the need for hose or pipe connections.

In order to withstand the very high pressures generated in the high pressure chamber 46 the high pressure cylinder 31 is clamped between the front end wall 35 and the intermediate wall 37 by means of second, heavier connecting rods 38 evenly distributed around the outer periphery of the high pressure cylinder 31. Threaded rear ends of the connecting rods are screwed into corresponding threaded blind bores provided in the front face of the intermediate wall 37 and the likewise threaded front ends thereof are extended through corresponding through bores in the front end wall 35 and are engaged by nuts 45 tightened against the end wall 35. The connecting rods 38 carry at least one support ring 39 surrounding and provided in close engagement with the outer periphery of the high pressure cylinder 31 for reasons explained below.

The high pressure cylinder 31, like the low pressure cylinder 32, is illustrated with a mid-portion thereof cut away, and it should be emphasized that one or more additional support rings 39 may be provided along the cut-away portion, the number of support rings 39 required depending i.a. upon the stroke length of the intensifier 30 and upon the strength of the wall of the high pressure cylinder 31.

In a groove or recess 28 of the high pressure cylinder 31 is provided a high pressure seal assembly 101 according to the invention, as is illustrated in greater detail in connection with the modified embodiment of fig. 9. The seal assembly 101 illustrated in the pressure intensifier of the invention basically corresponds to the embodiment of fig. 5A but any of the seal assembly embodiments of the invention may likewise be employed in the pressure intensifier. In the embodiment of fig. 8 the seal assembly 101 is clamped between the front wall 35 and the intermediate wall 37, in the latter case through the bushings 52, 29, by means of the connecting rods 38.

The seal assembly 101 is specifically suitable for sealing against the very high working fluid pressures of pressure intensifiers, such as for use in the hydroforming techniques and in some press equipment and ranging from about 2000 bar upwards. In order to provide its proper sealing function and to protect the soft seal of such a hard seal-soft seal-combination a seal assembly intended for such pressure ranges requires that a minute gap, referred to herein as a "zero play" condition, is established between the metal rings of the seal assembly and the high

pressure piston. Direct contact between the metal rings of the seal assembly 101 and the high pressure piston cannot be totally eliminated but should be minimized in order to minimize resistance and to avoid premature wear. In view of the fact that direct contact between the metal rings and the piston cannot always be avoided, the high pressure piston 27 is preferably hardened and provided with a titanium-nitride surface coating having a Brinell hardness number being approximately 400 higher than that of the metal rings.

In view of the above discussed desire to establish a "zero play" between the metal rings 4', 5, 5' of the seal assembly 101 and the high pressure piston 43 the pressure intensifier 30 of the invention is preferably provided with an arrangement for controlling said minute gap, that is for maintaining the "zero play" under the varying pressure conditions appearing during operation of the intensifier. In the illustrated preferred embodiment the high pressure cylinder 31 of the intensifier 30 specifically consists of separate, coaxial inner and outer casings 68 and 69 respectively. The inner casing 68 is provided closely fitting into the outer casing 69, and a pressure fluid space 60 is provided between the two casings, extending over a major portion of the axial length of the inner casing 68 and around the entire outer periphery thereof. In any event the fluid space should have an axial extension as well as an axial position corresponding to that of the seal assembly 101. In the illustrated embodiment the space 60 is provided in the shape of a ring-shaped recess in the outer periphery of the inner casing 38, but could likewise be formed by such a recess in the inner circumference of the outer casing 69, or even a combination of both. The space 60 preferably ends at a substantially equal distance from both ends of the inner casing 68. The width of the pressure fluid space 60 in the radial direction of the casings is in the order of a few μ .

The outer casing 69 is provided with an inlet channel 61, the inner end of which communicates with the pressure fluid space 60, and the outer end of which opens into the outer periphery of the outer casing 69 and is connected to a fluid line 62. The fluid line 62 supplies pressure fluid to the space 60 from a pressure fluid source. The fluid source may be a pressure intensifier 70 as indicated in fig. 8, or any other pressure fluid source delivering the adequate pressure for a specific application. In the case of the pressure intensifier 70 this may preferably be attached directly to the apparatus. It is also indicated in fig. 8 that in one embodiment the intensifier 70 is connected to a control unit 71 through which the pressure

level of the fluid output from the intensifier 70 is continuously controlled in dependence upon the detected working pressure of the apparatus, symbolized by an input 72 to the control unit 71. In this respect the "working pressure" means the instantaneous pressure in the high pressure chamber 46. This input 72 can therefore be in the form of a signal related to a pressure detected in the high pressure chamber 46, but may likewise be in the form of a signal related to a pressure detected in the working fluid chamber 32A, since the latter pressure is directly proportional to the pressure in the high pressure chamber 46. Conventional means that are not specifically illustrated or described herein will be used for detecting said pressures, and the described control functions may be performed in any suitable manner known within the general field of control techniques.

The inner casing 68 is axially shorter than the outer casing 69 and is generally formed of an inner section forming the inner wall of the high pressure cylinder 31, and an axially longer outer section. Like in the low pressure cylinder, the front and rear ends of the outer section of the inner casing 68 are guided by projections formed on the rear and front end faces of the front and intermediate walls 35 and 37 respectively and sealed with O-rings. In a similar manner the front and rear ends of the outer casing 69 are guided in circular grooves formed on the rear and front end faces of the front and intermediate walls 35 and 37 respectively and sealed with O-rings.

The general operation of the above described pressure intensifier will not be described in detail herein, but for further details of the operation of the pressure intensifier, reference is made to our Swedish Patent Application No. 9904464-6 having the title "Apparatus for generating hydraulic pressure".

During such normal operation of the pressure intensifier 30 the pressure of the fluid in the high pressure chamber 46 will be multiplied, in the present embodiment approximately 20 times, corresponding to the ratio of the areas of the high and low pressure pistons. With a system pressure of 350 bar this will mean an output pressure in the order of 7000 bar. With pressures of this magnitude, and even higher, the outwardly directed forces applied to the high pressure cylinder 31 would tend to expand the cylinder wall, thereby unacceptably increasing the gap between the high pressure piston 43 and the seal assembly 101. According to this

embodiment of the inventive pressure intensifier 30 said tendency will be counteracted by applying fluid pressure to the fluid pressure space 60, between the inner and outer cylinder casings 68, 99, as described above. By regulating the counter-pressure in direct relation to the prevailing pressure in the high pressure chamber 46, the gap between the seal 101 and the piston 43 will be kept at a constant small value securing a proper sealing function for the seal 101. In view of the differential area of the inner cylinder wall and the bottom wall of the pressure fluid space 60, the pressure of the fluid supplied to said space will only be in the order of a few hundred bars. Nevertheless it may be appropriate to provide the support ring or rings 39. Said ring or rings mainly serve to support the outer casing 68 radially. In alternative embodiments other methods of supplying fluid to the fluid space 40 may be employed.

In a further alternative embodiment not specifically illustrated, the high pressure cylinder may be formed by several separate casings, and a pressure fluid space may be provided between one or more adjacent casings. By means of such arrangements it will be possible to further vary the characteristics of the high pressure cylinder wall, for adapting it to different applications with specific requirements.

Fig. 9 illustrates an enlarged detail of a modified embodiment of the pressure intensifier 30 of fig. 8, in which the seal assembly 101 employed as high pressure seal is likewise identical to that illustrated in figs. 5A-D. However, this embodiment employs an alternative configuration of the arrangement for maintaining the very small gap, or rather a zero-play, between the hardened metal seal rings 4', 5, 5' of the seal assembly 101 and the high pressure piston 43, in order to secure an efficient seal and to protect the soft seal 14 of said hard seal-soft seal-combination. In this embodiment the control of said play is effected by providing a fluid space 60' in the wall of the high pressure cylinder 31', surrounding the seal assembly 101 at a relatively small radial distance therefrom and communicating directly with the high pressure chamber 46'. In this manner the gap will be continuously and automatically controlled by the actual fluid pressure prevailing in the high pressure chamber 46'. Specifically, through the fluid space 60' the momentary working fluid pressure will act upon the portion of the high pressure cylinder 31' wall containing the groove 28 and thereby supporting the seal assembly 101. Thus, with the increasing working fluid pressure and the concomitant tendency for radial

outward expansion of the seal 101 a correspondingly increasing counterforce will be applied by the fluid in the fluid space 60', counteracting said radial expansion.

5 Although the high pressure seal of the invention has been described above with sole reference to the modular arrangement where separate metal rings are arranged in a required combination, the invention is not, in its broadest aspect, restricted to this presently preferred embodiment. Fig. 10 illustrates an alternative embodiment of the inventive seal assembly 401, where the primary 402 and secondary 403 seals do not consist of separate metal rings, but are formed as one integrated unit. In this case the seal assembly 401 is adapted to the specific application by varying the width of the integrated primary 402 and secondary 403 seal sections depending upon the magnitude of the fluid pressure on the high pressure side and upon the viscosity of the fluid.

10 It will be understood by those skilled in the art that various modifications and changes may be made to the present invention without departure from the scope thereof, which is defined by the appended claims.

PATENT CLAIMS

1. Seal assembly (1; 101; 201; 301; 401) for sealing the gap between two linearly relatively moveable, generally cylindrical parts (26, 27) to prevent fluid leakage from a high pressure side (HPS) to a low pressure side (LPS), and comprising at least one ring seal (2, 3; 102, 103; 202, 203; 302, 303; 402; 403) having a first peripheral surface (8, 9) supported by one (26) of said parts and a second peripheral surface (10, 11) for sealing against said other part (27), **characterized by** a primary seal (2; 102; 202; 302; 402) facing the high pressure side and consisting of at least one metal ring (4; 4'; 4'') having several small peripheral sealing lips (6) provided at its second peripheral surface (10), for performing an initial sealing function and for simultaneously providing a pressure drop thereacross, and by a secondary seal (3; 103; 203; 303; 403) provided behind the primary seal, with regard to the high pressure side, and consisting of at least one metal ring (5; 5'; 5'') being provided with a recess (13) in its second peripheral surface (11), and a first soft sealing ring (14) received in the recess.

2. Seal assembly (201; 301) according to claim 1 for applications working with a fluid pressure on the high pressure side (HPS) of above approximately 2000bar, **characterized in** that the primary seal (202; 302) consists of several metal rings (4; 4'; 4'') provided next to each other, followed by the secondary seal (3; 103; 203; 303), the number of metal rings (4; 4'; 4'') provided in the primary seal depending upon the magnitude of the fluid pressure on the high pressure side (HPS).

3. Seal assembly (101; 201; 301) according to claims 1 or 2 for applications working with a fluid pressure on the high pressure side (HPS) of above approximately 2000bar, **characterized in** that the secondary seal (103; 203; 303) consists of several metal rings (5; 5'; 5'') provided next to each other, following the primary seal (102; 202; 302), the number of metal rings (5; 5'; 5'') provided in the secondary seal depending upon the magnitude of the fluid pressure on the high pressure side (HPS), and in that at least all of the metal rings (5, 5') of the secondary seal except the last (5''), as seen from the high pressure side, have several small peripheral sealing lips (12) provided at their second peripheral surface (11).

4. Seal assembly (1; 101; 201; 301; 401) according to any of claims 1 - 3, **characterized in** that the metal rings (4, 4', 4''; 5, 5', 5'') of the primary and secondary seals (2, 3; 102, 103; 202, 203; 302, 303; 402; 403) each consist of hardened steel.

5. Seal assembly (1; 101; 201; 301; 401) according to any of claims 1 - 4, **characterized in** that the soft seal (14) provided in the recess (13) of the secondary seal metal rings (5; 5'; 5'') consist of a synthetic composite material.

6. Seal assembly (1; 101; 201; 301) according to any of claims 1-5, **characterized in** that the metal rings (4, 4', 4''; 5; 5'; 5'') of the primary and secondary seals (2, 3; 102, 103; 202, 203; 302, 303) are each provided with a generally semicircular groove (15) in their side surfaces (16A, 16B, 17A, 17B) facing an adjacent metal ring (4, 4', 4'' or 5; 5'; 5'') and in that a second sealing ring (18; 18'; 18'') is provided in said grooves of adjacent metal rings.

7. Seal assembly (1; 101; 201; 301) according to claim 6, **characterized in** that said second sealing ring (-s) (18'; 18'') has a width larger than the combined depth of the grooves (15) of two adjacent metal rings (4, 4', 4'', 5; 5'; 5'') and in that the second sealing ring (-s) (18'; 18'') consists of a soft metal or alternatively a composite material deformable into the grooves to provide a radial seal between the two adjacent metal rings.

8. Seal assembly (1; 101; 201; 301; 401) according to any of claims 1-7, intended to be received in a circumferential groove (28) of said one part (26) with said first peripheral surface (8, 9), **characterized in** that the at least one metal ring (4; 4'; 4'') of the primary seal (2; 102; 202; 302; 402) has several small peripheral sealing lips (19) provided at its first peripheral surface (8), for performing a sealing function against the groove (28) of said one part and for simultaneously providing a pressure drop thereacross.

9. Seal assembly (1; 101; 201; 301; 401) according to claim 8, **characterized in** that at least all of the metal rings (5; 5') of the secondary seal (3) except the last (5''), as seen from the high pressure side (HPS), have several small peripheral sealing lips (20) provided at their first peripheral surface (9), for performing a sealing function against the groove (28) of said one part (26) and for simultaneously providing a pressure drop thereacross.

10. Seal assembly (1; 101; 201; 301; 401) according to claim 8 or 9, **characterized in** that at least some of the metal rings (4, 4', 4'', 5, 5', 5'') of the primary and secondary seals (2, 3; 102, 103; 202, 203; 302, 303; 402; 403) are provided with a cut-out (21, 22) at the corner thereof formed by the first peripheral surface (8, 9) and the side surface (16A, 16A', 17A) facing the high pressure side (HPS) and in that a third sealing ring (23) is provided in each such cut-out.

11. Seal assembly (101; 201; 301; 401) according to claim 8, 9 or 10, **characterized in** that the first metal ring (4') of the primary seal (102; 202; 302; 402) is formed having a circular protrusion (25) on its side (16A') facing the high pressure side (HPS), said protrusion having a generally triangular, outwardly pointed cross section, and intended to be pressed into a side surface (28A) of the groove (28) of said one part (26) upon assembly, to thereby provide a radial seal.

12. Seal assembly (1; 101; 201; 301; 401) according to any of the preceding claims, **characterized in** that the small sealing lips (6, 12, 19 and/or 20) have a height of approximately 0,1-0,2 mm.

13. Seal assembly (1; 101; 201; 301; 401) according to any of the preceding claims, **characterized in** that the first sealing ring (14) consists of a synthetic composite material, preferably a high molecular glass-filled nylon material withstanding high surface pressure.

14. Seal assembly (1; 101; 201) according to any of the preceding claims, **characterized by** a fourth sealing ring (24) provided facing the low pressure side (LPS) behind the last metal ring (5; 5'') of the secondary seal (3; 103; 203), said fourth sealing ring being a synthetic lip seal providing the final approximately fluid-tight sealing towards the low pressure side (LPS).

15. Seal assembly (401) according to any of the preceding claims, **characterized in** that the metal ring (-s) of the primary seal (402) and the metal ring (-s) of the secondary seal (403) are formed as one integrated unit.

16. Seal assembly (401) according to claim 15, **characterized in** that the width of the integrated primary (402) and secondary (403) seals is dependent upon the magnitude of the fluid pressure on the high pressure side (HPS).

5 17. Pressure intensifier (30) having a low pressure piston (42) being linearly displaceable in a low pressure cylinder (32) forming a working chamber (32A) in turn being connected to a low pressure fluid supply, said low pressure piston carrying a high pressure piston (43) being displaceable in a high pressure cylinder (31; 31') forming a high pressure chamber (46; 46') being connected to a low pressure fluid supply and to a high pressure fluid consumer, the
10 inner wall of the high pressure cylinder carrying a high pressure seal assembly (1; 101; 201; 301; 401) for sealingly engaging the high pressure piston to effectively seal fluid in a high pressure side (HPS) from a low pressure side (LPS), **characterized by** in that the high pressure seal assembly consists of a primary seal (2; 102; 202; 302; 402) facing the high pressure side and consisting of at least one metal ring (4; 4'; 4'') having several small
15 peripheral sealing lips (6) provided at an inner peripheral surface (10) thereof, for performing an initial sealing function and for simultaneously providing a pressure drop thereacross, and by a secondary seal (3; 103; 203; 303; 403) provided behind the primary seal, with regard to the high pressure side, and consisting of at least one metal ring (5; 5'; 5'') being provided with a recess (13) in an inner peripheral surface (11) thereof, and a first soft sealing ring (14)
20 received in the recess.

18. Pressure intensifier (30) according to claim 17, for applications working with a fluid pressure on the high pressure side (HPS) of above approximately 2000bar, **characterized in** that the primary seal (202; 302) consists of several metal rings (4; 4'; 4'') provided next to each other, followed by the secondary seal (3; 103; 203; 303), the number of metal rings (4; 4'; 4'') provided in the primary seal depending upon the magnitude of the fluid pressure on the high pressure side (HPS).
25

19. Pressure intensifier (30) according to claims 17 or 18, for applications working with a fluid pressure on the high pressure side (HPS) of above approximately 2000bar **characterized in** that the secondary seal (103; 203; 303) consists of several metal rings (5; 5'; 5'') provided next to each other, following the primary seal (102; 202; 302), the number of metal rings (5;
30

5' 5") provided in the secondary seal depending upon the magnitude of the fluid pressure on the high pressure side (HPS), and in that at least all of the metal rings (5, 5') of the secondary seal except the last (5"), as seen from the high pressure side, have several small peripheral sealing lips (12) provided at their second peripheral surface (11).

5
20. Pressure intensifier (30) according to any of claims 17-19, for applications working with a fluid pressure on the high pressure side (HPS) of above approximately 2000 bar, **characterized in** that a minute gap, i.e. producing a "zero play" condition, exists between the metal rings (4; 4'; 4''; 5; 5'; 5") of the seal assembly (1; 101; 201; 301; 401) and the high pressure piston (43).

10
21. Pressure intensifier (30) according to any of claims 17-20, **characterized in** that the metal rings (4, 4', 4''; 5, 5', 5") of the primary and secondary seals (2, 3; 102, 103; 202, 203; 302, 303; 402; 403) each consist of hardened steel.

15
22. Pressure intensifier (30) according to any of claims 17-21, **characterized in** that the soft seal (14) provided in the recess (13) of the secondary seal metal rings (5; 5'; 5") consist of a synthetic composite material.

20
23. Pressure intensifier (30) according to any of claims 17-22, **characterized in** that the metal rings (4, 4', 4''; 5; 5'; 5") of the primary and secondary seals (2, 3; 102, 103; 202, 203; 302, 303) are each provided with a generally semicircular groove (15) in side surfaces (16A, 16B, 17A, 17B) thereof facing an adjacent metal ring (4, 4', 4'' or 5; 5'; 5") and in that a second sealing ring (18; 18'; 18'') is provided in said grooves of adjacent metal rings.

25
24. Pressure intensifier (30) according to claim 23, **characterized in** that said second sealing ring (-s) (18'; 18'') has a width larger than the combined depth of the grooves (15) of two adjacent metal rings (4, 4', 4'', 5; 5'; 5") and in that the second sealing ring (-s) (18'; 18'') consists of a soft metal or alternatively a composite material deformable into the grooves to provide a radial seal between the two adjacent metal rings.

30

25. Pressure intensifier (30) according to any of claims 17-24, wherein the high pressure seal assembly (1; 101; 201; 301; 401) is received with said first peripheral surface (8, 9) in a circumferential groove (28) in the inner wall of the high pressure cylinder (31; 31'), **characterized in** that the at least one metal ring (4; 4'; 4'') of the primary seal (2; 102; 202; 302; 402) has several small peripheral sealing lips (19) provided at its first peripheral surface (8), for performing a sealing function against the groove (28) of the high pressure cylinder and for simultaneously providing a pressure drop thereacross.

26. Pressure intensifier (30) according to claim 25, **characterized in** that at least all of the metal rings (5; 5') of the secondary seal (3) except the last (5''), as seen from the high pressure side (HPS), have several small peripheral sealing lips (20) provided at their first peripheral surface (9), for performing a sealing function against the groove (28) and for simultaneously providing a pressure drop thereacross.

27. Pressure intensifier (30) according to claim 25 or 26, **characterized in** that at least some of the metal rings (4, 4', 4'', 5, 5', 5'') of the primary and secondary seals (2, 3; 102, 103; 202, 203; 302, 303; 402; 403) are provided with a cut-out (21, 22) at the corner thereof formed by the first peripheral surface (8, 9) and the side surface (16A, 16A', 17A) facing the high pressure side (HPS) and in that a third sealing ring (23) is provided in each such cut-out.

28. Pressure intensifier (30) according to claim 25, 26 or 27, **characterized in** that the first metal ring (4') of the primary seal (102; 202; 302; 402) is formed having a circular protrusion (25) on its side (16A') facing the high pressure side (HPS), said protrusion having a generally triangular, outwardly pointed cross section, and intended to be pressed into a side surface (28A) of the groove (28) in the high pressure cylinder (31; 31') upon assembly, to thereby provide a radial seal.

29. Pressure intensifier (30) according to any of claims 17-28, **characterized in** that the small sealing lips (6, 12, 19 and/or 20) have a height of approximately 0,1-0,2 mm.

30. Pressure intensifier (30) according to any of claims 17-29, **characterized in** that the first sealing ring (14) consists of a synthetic composite material, preferably a high molecular glass-filled nylon material withstanding high surface pressure.

5 31. Pressure intensifier (30) according to any of claims 17-30, **characterized by** a fourth sealing ring (24) provided facing the low pressure side (LPS) behind the last metal ring (5; 5") of the secondary seal (3; 103; 203), said fourth sealing ring being a synthetic lip seal providing the final approximately fluid-tight sealing towards the low pressure side (LPS).

10 32. Pressure intensifier (30) according to any of claims 17-31, **characterized in** that the metal ring (-s) of the primary seal (402) and the metal ring (-s) of the secondary seal (403) are formed as one integrated unit.

15 33. Pressure intensifier (30) according to any of claims 20-32, **characterized by** an arrangement for controlling the gap between the metal rings (4', 5, 5') of the seal assembly (101) and the high pressure piston (43), said arrangement comprising coaxial inner (68) and outer casings (69) forming the high pressure cylinder (31), a pressure fluid space (60) provided between the two casings and a pressure fluid supply (70, 61, 62) for supplying pressurized fluid to said pressure fluid space to counteract the outwardly directed forces applied to the
20 high pressure cylinder (31) and tending to expand the cylinder wall during operation of the pressure intensifier.

25 34. Pressure intensifier (30) according to claim 33, **characterized in** that a control unit (71) is connected to the pressure fluid supply (70, 61, 62) for continuously controlling the pressure level of the fluid supply therefrom to the pressure fluid space (60) in dependence upon the detected working pressure of the pressure intensifier, to maintain the "zero play" condition under the varying pressures appearing during operation of the pressure intensifier.

30 35. Pressure intensifier (30) according to any of claims 20-32, **characterized by** an arrangement for controlling the gap between the metal rings (4', 5, 5') of the seal assembly (101) and the high pressure piston (43), said arrangement comprising a fluid space (60') in the wall of the high pressure cylinder (31'), surrounding the seal assembly (101), at a relatively small

radial distance therefrom and communicating directly with the high pressure chamber (46'), thereby continuously and automatically controlling said gap in dependence upon the actual fluid pressure prevailing in the high pressure chamber 46'.

5 36. A method of sealing the gap between two linearly relatively moveable, generally cylindrical parts (26, 27) to prevent fluid leakage from a high pressure side (HPS) to a low pressure side (LPS), by providing at least one ring seal (2, 3; 102, 103; 202, 203; 302, 303; 402; 403) having a first peripheral surface (8, 9) supported by one (26) of said parts and a
10 second peripheral surface (10, 11) for sealing against said other part (27), **characterized by** providing a primary metal seal (2; 102; 202; 302; 402) facing the high pressure side, by forming several small peripheral sealing lips (6) at the second peripheral surface (10) of the primary metal seal (2; 102; 202; 302; 402), for performing an initial sealing function and for simultaneously providing a pressure drop thereacross, by providing a secondary metal seal (3; 103; 203; 303; 403) behind the primary seal, with regard to the high pressure side, by forming
15 at least one recess (13) in the second peripheral surface (11) of the secondary seal, and by positioning a soft sealing ring (14) in the or each such recess.

37. A method according to claim 36, for applications working with a fluid pressure on the high pressure side (HPS) of above approximately 2000bar, **characterized by** determining the
20 number of metal rings (4; 4'; 4"; 5; 5'; 5") forming the primary and/or secondary (2, 3; 102, 103; 202, 203; 302; 303) based on the magnitude of the fluid pressure on the high pressure side (HPS).

38. A method according to claim 36 or 37, **characterized by** forming the primary and/or
25 secondary seals (2, 3; 102, 103; 202, 203; 302; 303) by separate metal rings (4, 4', 4'', 5, 5', 5") and in that said metal rings are arranged side by side.

39. A method according to claim 36 or 37, **characterized by** forming the primary and/or
secondary seals (402, 403) as an integrated unit.

30 40. A method according to any of claims 36 - 38, **characterized by** providing radial seals (18, 18', 18'', 23, 25) between each adjacent metal rings (4, 4', 4'', 5, 5" 5") for preventing radial

leakage, and by forming several small peripheral sealing lips (19, 20) on the first peripheral surface (8; 9) of at least some of the metal rings (4, 4', 4'', 5' 5'), for sealing against said one part (26).

5 41. A method according to claim 40, **characterized by** providing deformable radial seals (18', 18'') between each metal ring (4, 4', 4'', 5, 5', 5'') of the primary and secondary seals (2, 3; 102, 103; 202, 203; 302, 303).

10 42. A method according to claim 41, **characterized by** clamping the metal rings (4, 4', 4'', 5, 5', 5'') of the primary and secondary seals (2, 3; 102, 103; 202, 203; 302, 303) firmly together to deform the radial seals (18', 18'') into grooves (15) formed in sides (16a, 16b, 17a, 17b) of the metal rings.

FIG. 1

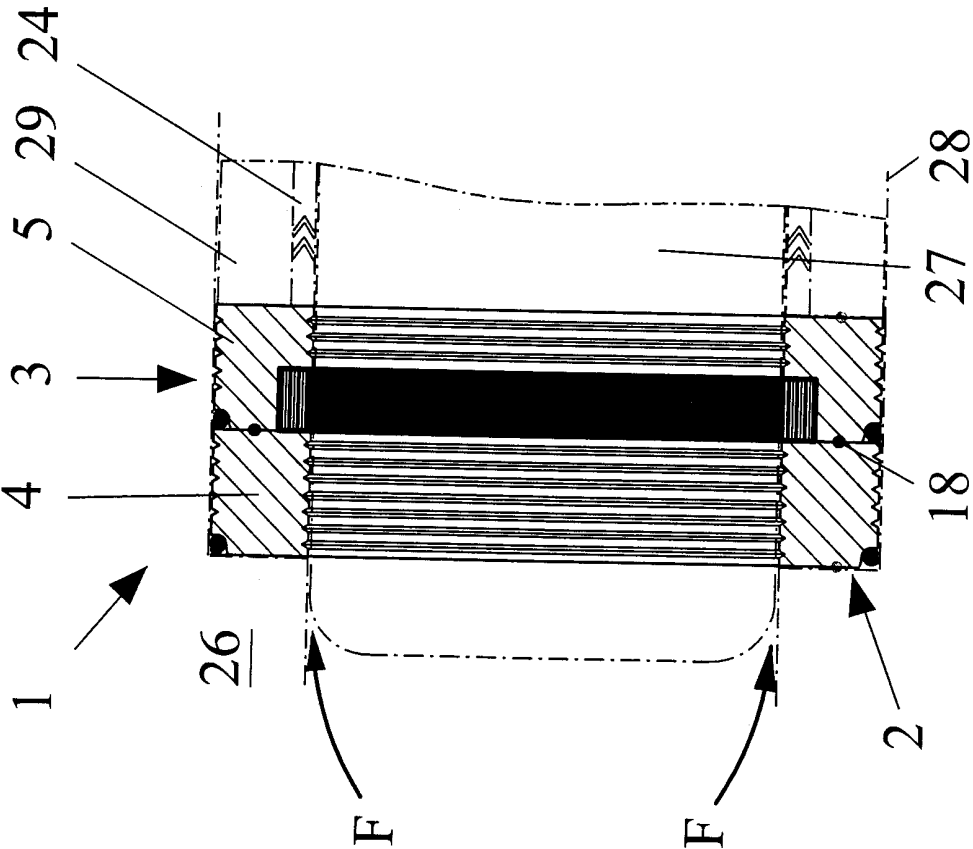
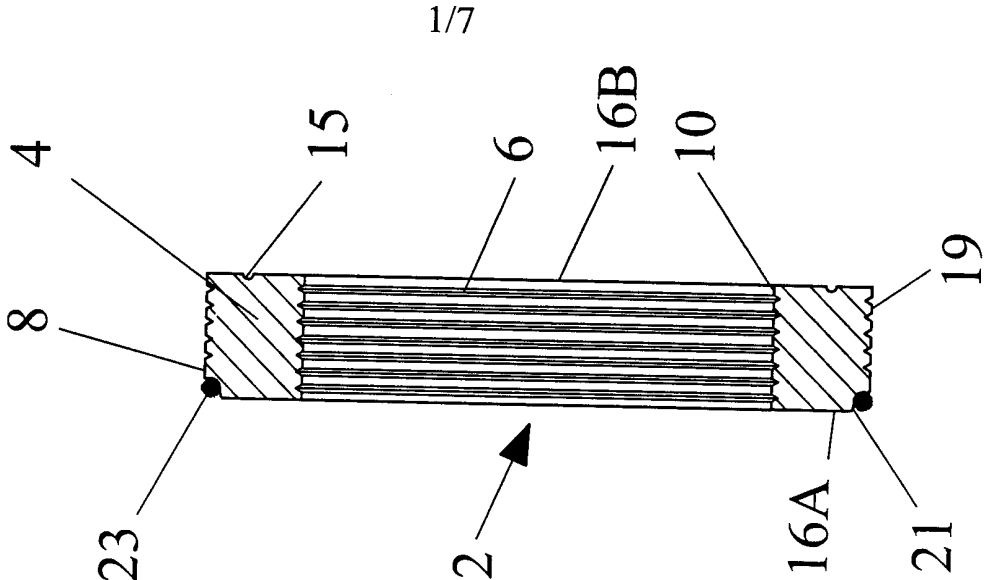


FIG. 2



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FIG. 3B

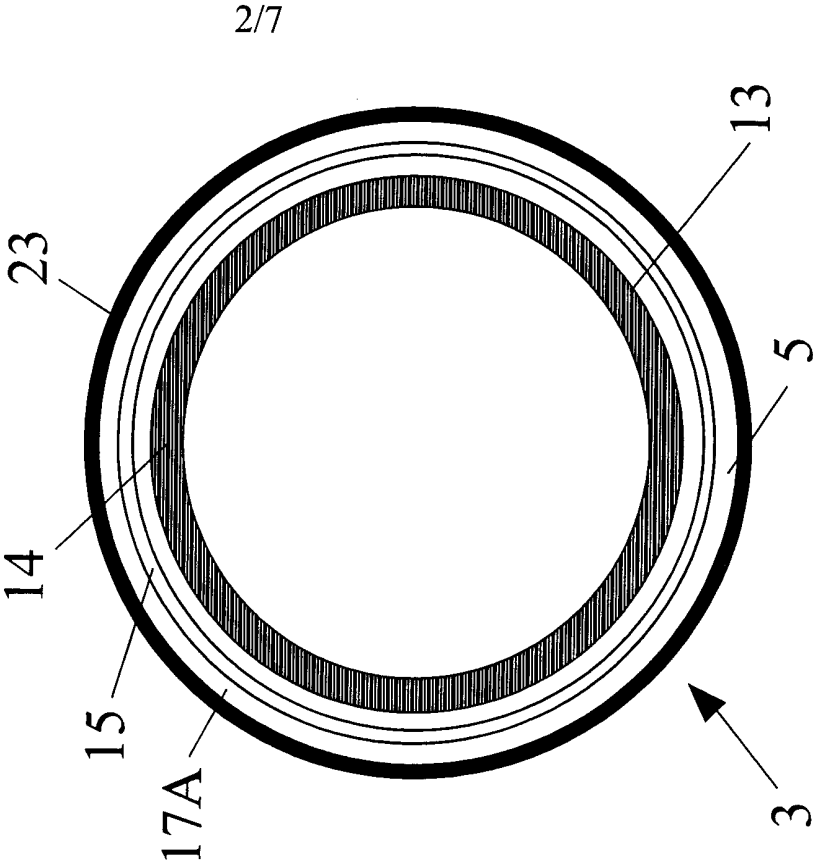


FIG. 3A

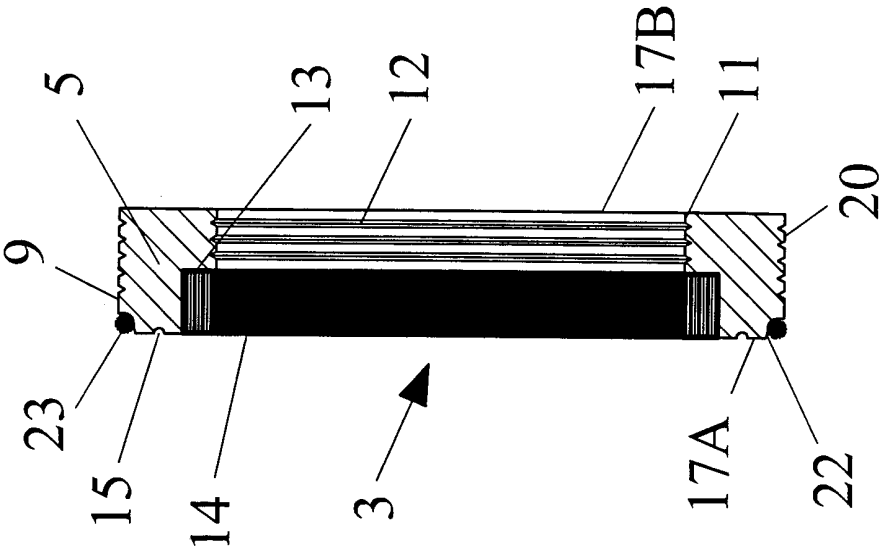


FIG. 5B

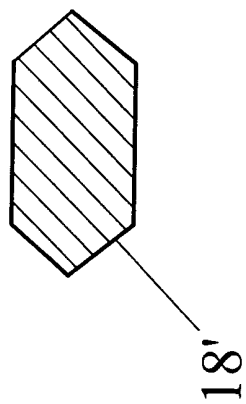


FIG. 5C

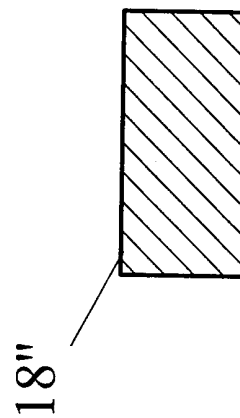


FIG. 4

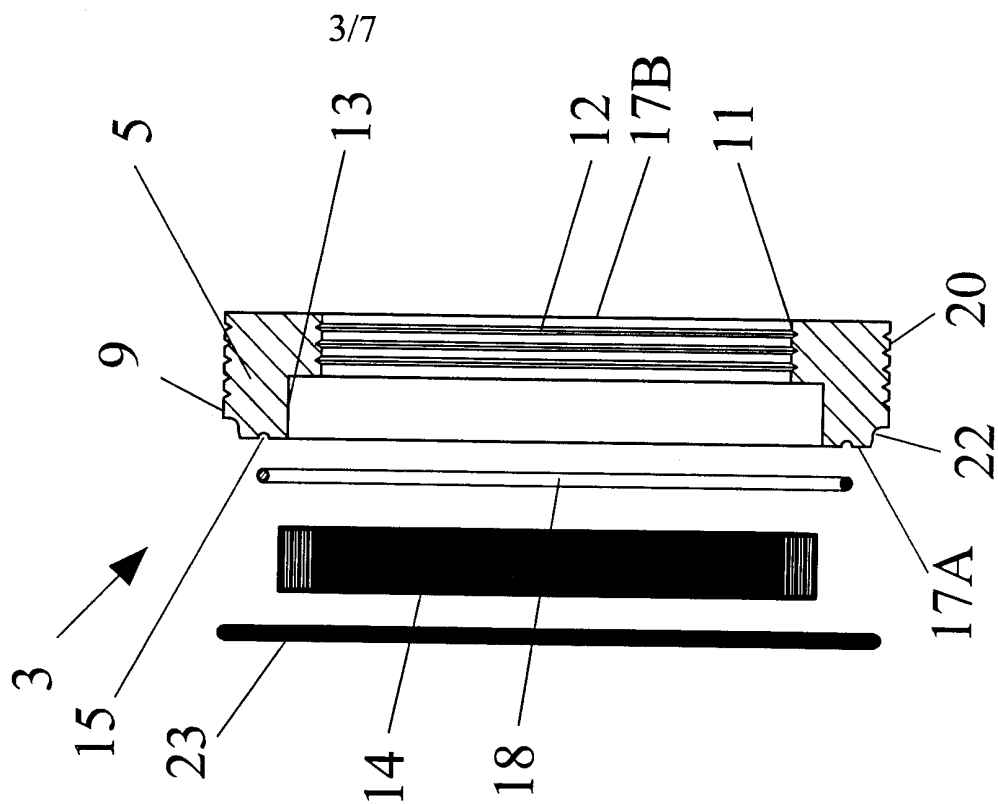


FIG. 5D

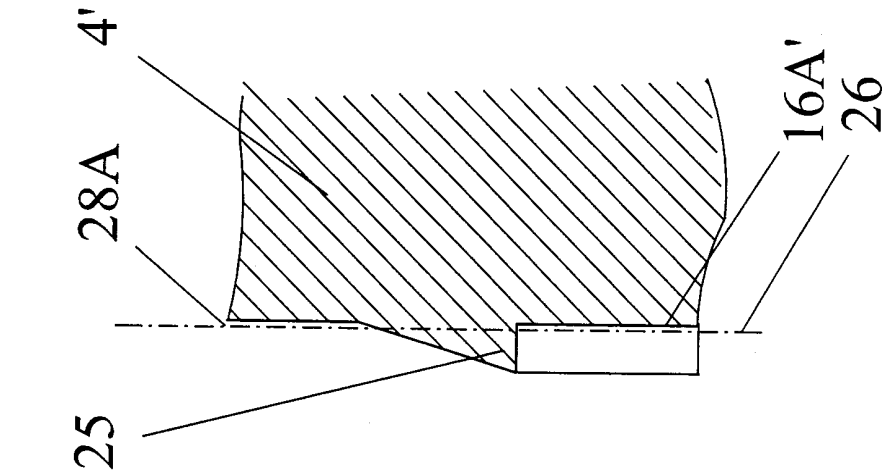
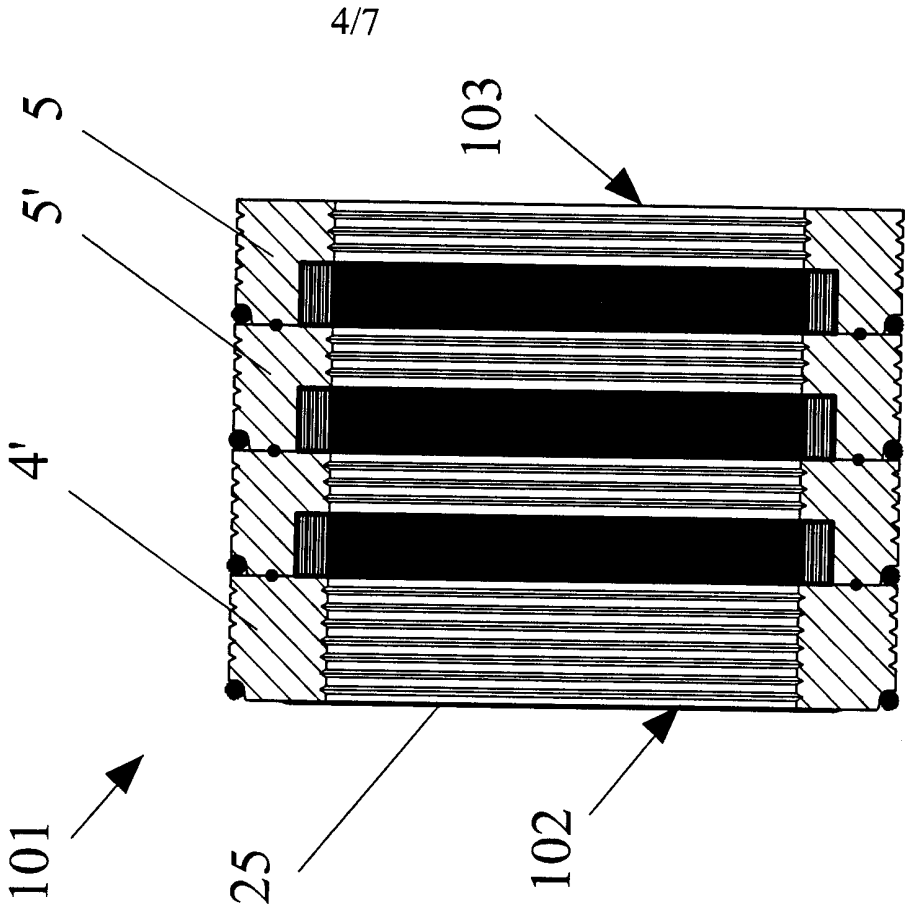


FIG. 5A



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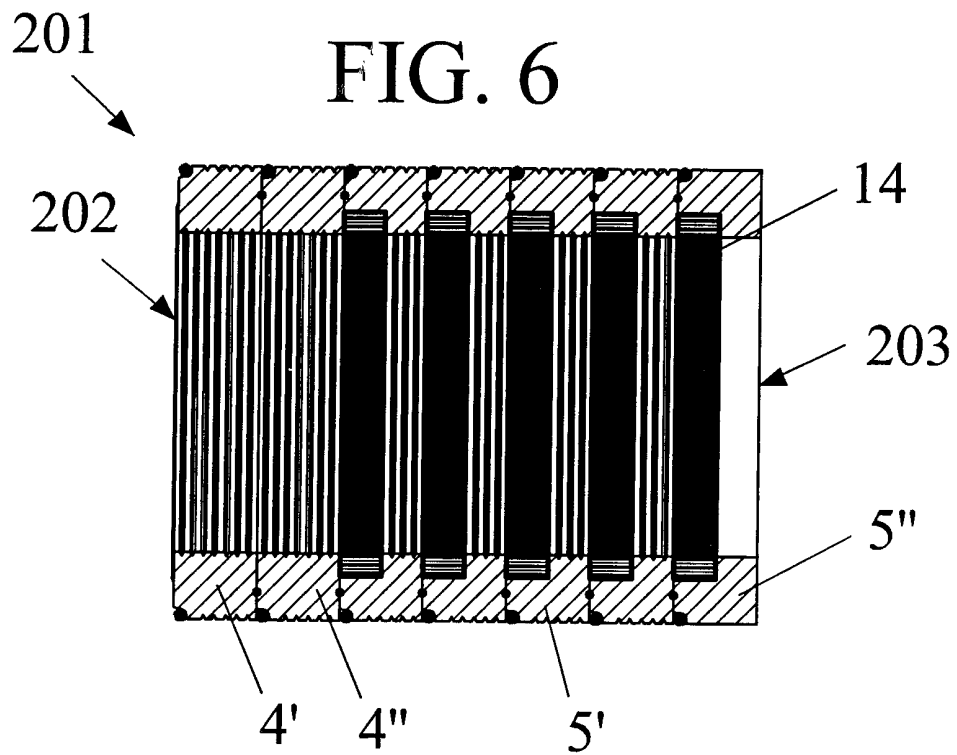


FIG. 7

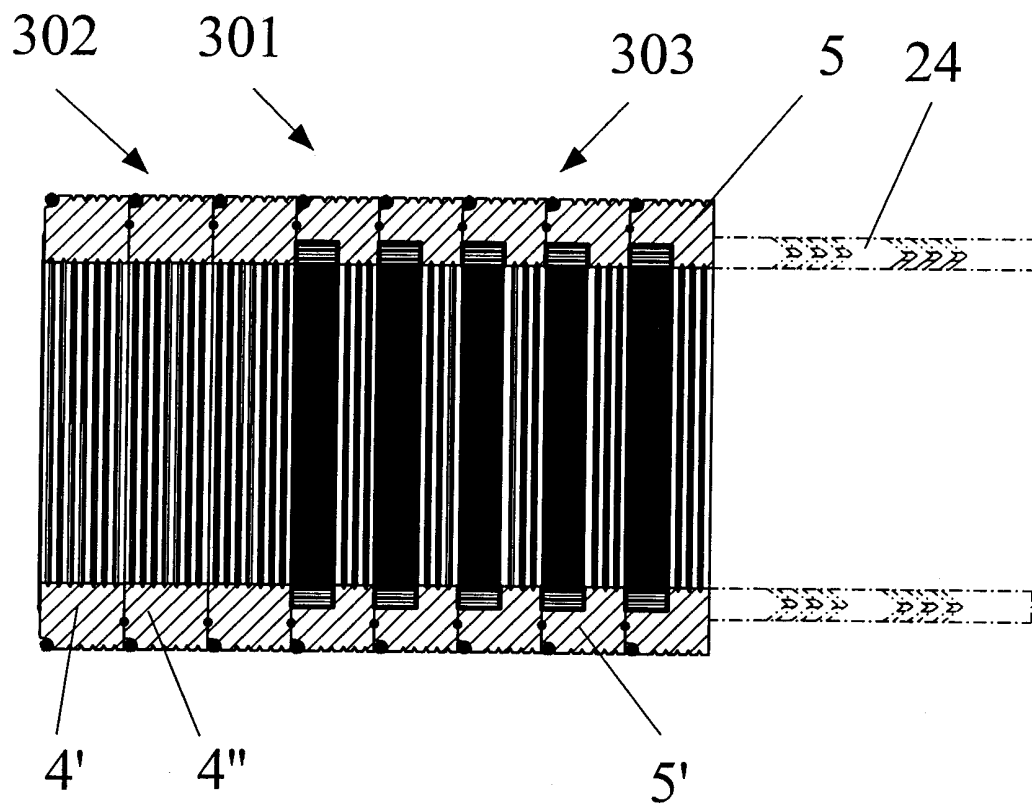


FIG. 9

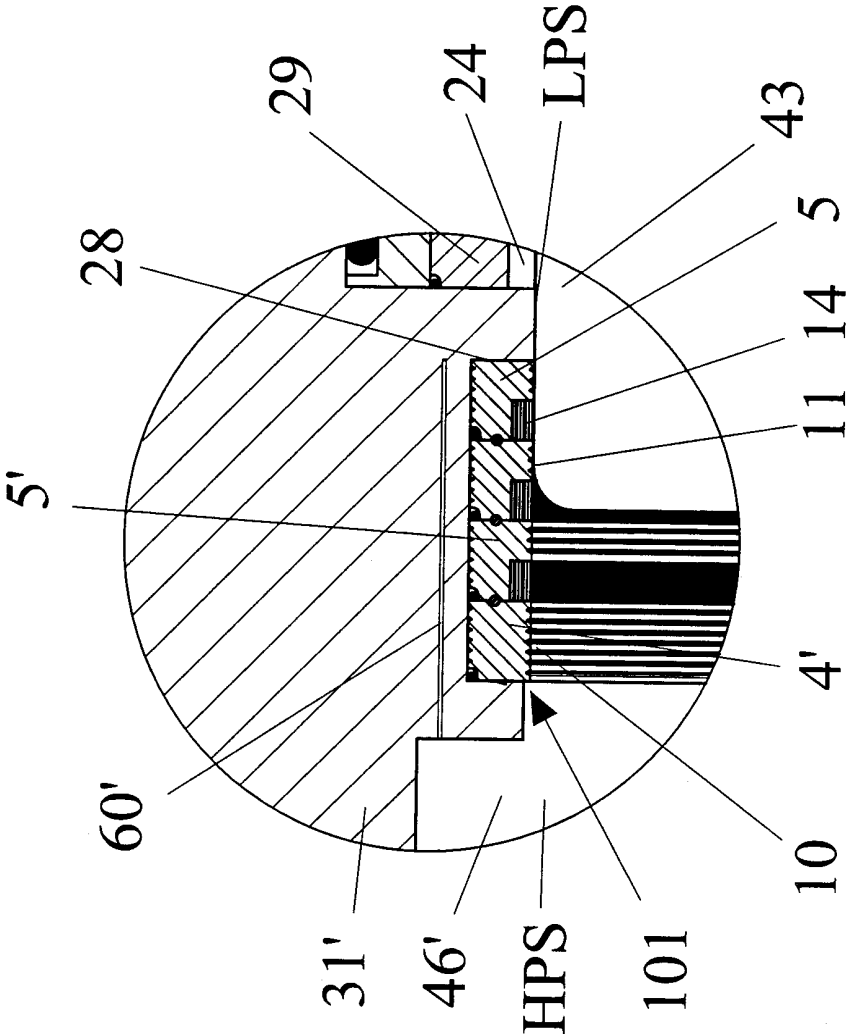
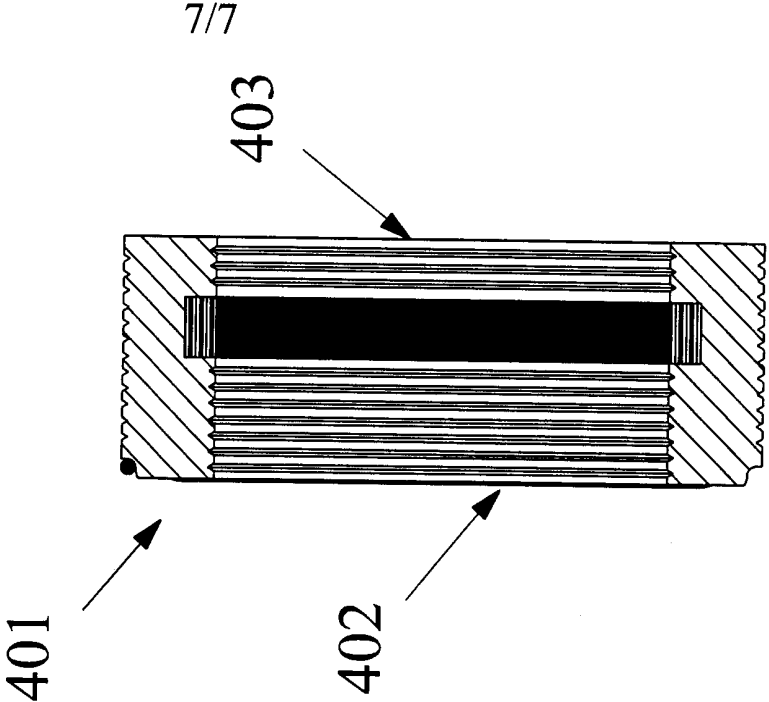


FIG. 10



INTERNATIONAL SEARCH REPORT

International application No.

PCT/SE 00/02367

A. CLASSIFICATION OF SUBJECT MATTER

IPC7: F16J 15/26

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

IPC7: F16J

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

SE,DK,FI,NO classes as above

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	CH 641883 A5 (CARRIER CORP, SYRACUSE/NY /US), 15 March 1984 (15.03.84), page 3, column 1, line 57 - column 2, line 31, figures --	1-16,36-42
X	US 4466620 A (ORLOWSKI), 21 August 1984 (21.08.84), abstract, figures --	1-16,36-42
X	US 4706968 A (ORLOWSKI), 17 November 1987 (17.11.87), column 4, line 3 - line 45, figures --	1-16,36-42
X	US 5074567 A (ORLOWSKI), 24 December 1991 (24.12.91), abstract, figures --	1-16,36-42



Further documents are listed in the continuation of Box C.



See patent family annex.

* Special categories of cited documents:

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"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

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"&" document member of the same patent family

Date of the actual completion of the international search

Date of mailing of the international search report

12 February 2001

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Name and mailing address of the ISA/

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INTERNATIONAL SEARCH REPORT

Information on patent family members

25/02/01

International application No.

PCT/SE 00/02367

Patent document cited in search report			Publication date	Patent family member(s)	Publication date
CH	641883	A5	15/03/84	US 4257617 A	24/03/81
US	4466620	A	21/08/84	NONE	
US	4706968	A	17/11/87	NONE	
US	5074567	A	24/12/91	NONE	