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(54) Benævnelse: **Hydraulisk forspændt elastomert fjederelement og dets anvendelse i lejer til vindkraftanlæg**

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Description

The invention relates to spring elements and to bearings which contain these spring elements, the spring stiffness of the bearings or spring elements being individually adjustable by deformable elastomer bodies or elastomer layers by pre-tensioning. The deformation of the elastomer layers and thus of the spring elements is achieved by an appropriately designed hydraulic device. The leak-tightness of the spring elements and bearing elements according to the invention is achieved through special hydraulically pre-tensioned sealing elements, use of a pressure membrane and special design features of the elastomer elements. The elastomer bearing arrangements according to the invention, including their spring elements, are suitable, in particular, for use in two- or multi-point bearing arrangements in rotor and gearbox constructions of large wind turbines, where they serve to absorb damping and drive-train vibrations, which can lead, in particular, to displacement forces, bending forces and deformation forces and can occur especially in extreme loading situations of the turbine.

Elastic spring elements are known and are used in many areas of technology, primarily for damping vibrations and forces. The spring element here has a defined stiffness which is predetermined by the type, size, shape and number of the elastomer layers present and which can optionally be designed to be variable within certain limits after installation of the spring element.

In the case of large wind turbines with outputs of more than two megawatts, strong forces quite often act, in particular, on rotor blades, rotor shaft and drive-train bearing arrangement, but also on the nacelle itself, and these forces subject the two-, three- or four-point bearings usually used for the gearboxes and generators to large loads, meaning that corresponding, in particular also vertical, displacements, deformations, distortions and pitch movements of the turbine can result, possibly with damage to the material or individual components.

The drive train of wind turbines is generally fixed elastically to the bedplate. This is carried out for reasons of structure-borne noise insulation and also to enable displacements of the system. The known bearings are usually fitted with passive elastic elements which are able to absorb at least some of such constraining forces and generated vibrations owing to their orientation and different pre-set stiffnesses.

In the event of very strong, in particular suddenly and rapidly occurring loads (extreme load case) especially in the vertical direction, i.e. normally in the direction in which the elastomeric spring elements usually arranged have high spring stiffness, the elasticity or stiffness of these elements, which, with respect to their adjustable stiffness, are generally designed for average loads and the forces usually acting, in particular on the drive-train bearing arrangement, is not sufficient, so that the aforementioned deformations, displacements and distortions which occur in the turbine lead to damage to the turbine, especially to the gearbox, as described in more detail below.

In the prior art, various systems are employed for supporting the drive train of wind turbines. One of the systems is the moment bearing arrangement with rigid rotor shaft (Figs. 1 and 2). In this system, the rotor shaft is connected to the bedplate in a cardanically rigid manner, i.e. two rolling bearings (02) (movable bearing and fixed bearing) which transmit all the yawing and pitching moments are seated on the rotor shaft. The advantage of this system is that all the loads arising at the rotor (01) are transmitted directly into the tower on a short path. A further advantage is that, on dismounting the rotor, the rotor shaft (03) and the gearbox (04) can remain in the turbine. At the same time, the gearbox can also be dismounted without removal of the rotor shaft. The support can be provided as described by two rolling bearings or by one large moment bearing which performs the same mechanical function. In accordance with the prior art, the torque is absorbed on both sides of the gearbox. The disadvantage of this system is the indeterminate support resulting from four bearing points. This indeterminate support gives rise to constraining forces which may have the following causes: manufacturing and assembly tolerances, misalignment, inclination at the rotor shaft and at the gearbox flange and not least bending of all load-bearing elements relative to one another in the event of asymmetrically acting force transmission by the rotor. Besides the drive torque, other moments additionally occur at the rotor. These are caused, for example, by uneven wind flow or by the rotor blade moving past the tower.

In accordance with the prior art, elastomer components that are as soft as possible are used for absorbing the gearbox torque, so that the displacement forces are kept as small as possible. However, this also causes a large rotary movement of the gearbox on load introduction, which in turn causes a displacement of the gearbox output shaft relative to the generator shaft, which is disadvantageous, and consequently this softness of the elastomer components can only be used to a limited extent, or correspondingly complex couplings are required between gearbox and generator.

The system described in EP 1 566 543 A1 enables the constraining forces explained above to be removed from the four-point bearing arrangement of the turbine. This application describes an elastomer bearing arrangement for wind turbines with adjustable stiffness, in which the stiffness of the elastomer elements in said vertical direction can be varied by a hydraulic or mechanical device. These spring elements essentially consist of a connecting plate and an end plate between which is located at least one elastomer layer, the connecting plate having an opening with a connecting part, through which pressure can be exerted on the elastomer layer by means of a displacement element in the form of a hydraulic fluid or a movable piston element, with the result that an increase in the pre-tension of the spring element and thus a stiffening in the vertical direction is achieved.

In practice, these spring elements, although achieving the desired technical effects in respect of the damping, have proven problematic since, under the requisite high pressures that have to be generated in order to increase the stiffness in the vertical direction sufficiently, leak-tightness problems repeatedly occur, with loss of hydraulic fluid, even though the elastomer layers compressed by hydraulic fluid are securely connected to the surrounding parts by vulcanisation and/or adhesive bonding.

After further investigations and tests with the spring element described in EP 1 566 543, but also generally in other elastomeric bearing arrangements of the prior art, it has now been found, surprisingly, that large-volume elastomer components in particular cannot readily be sealed if they are in contact with hydraulic fluids of whatever type (e.g. water, oil, alcohols, mixtures thereof) under pressure. In the event of continuous loading and under relatively high pressures, small droplets of the hydraulic fluid are apparently absorbed by the porous structure that is present, in particular in the case of large elastomer volumes, and are continuously transported further through the flow structures in the elastomer material, which are difficult to avoid, until they escape at various, often unexpected, points of the component. Without being tied to a theory, these results may be interpreted in such a way that a droplet of the hydraulic fluid in the corresponding hydraulic space is forced into a small notch or pore in the surface of the adjacent elastomer which opens and closes during the dynamic loading, so that the droplet is locked in and continuously conveyed further until it reaches the end of the elastomer or component and thus allows a leak to occur. The escape of hydraulic fluid from the spring elements or bearings of EP 1 566 543 A1 thus represents a serious problem which has to be solved.

In addition, it has been found that direct contact of the hydraulic fluid with the elastomer material of the spring elements may lead to reduced durability or elasticity of the elastomer under the action of high pressures, meaning that the corresponding spring elements may under certain circumstances have to be replaced earlier.

The object is thus to provide corresponding spring elements or bearings based on the basic concept of EP 1 566 543 A1 which, while having the same or improved adjustability of the stiffness of the spring element, do not have the described disadvantages of the corresponding bearing arrangements of the prior art.

The object has been achieved by the spring elements, bearings and their use specified below and in the claims.

The invention thus relates to a spring element which can be adjusted in its stiffness by hydraulically generated pressure and is built up from layers, essentially comprising two or more elastomer layers (1), non-elastic interlayers (2), one or more hydraulic devices including sealing elements (4, 7, 9, 11, 12) and optionally non-elastic end plates, where two elastomer layers are separated from one another by a non-elastic interlayer or intermediate plate, and the non-elastic layers have a centrally arranged opening or hole, so that the spring element has, at least in the interior, a continuous elastomeric core, to which the elastomer layers are connected, where said hydraulic device is provided - viewed in the vertical direction with respect to the layers - on one or both sides of the spring element and causes compression of the elastomer material (1) in the core region thereof in the vertical direction with respect to the layers, by displacement or introduction of a hydraulic fluid into the spring element by means of hydraulic pressure, so that a displacement space is formed or enlarged. Said spring element is distinguished here by the fact that an elastic pressure membrane (5) is arranged between the elastomer layer in the core region of the spring element and the displacement space and is securely connected at its edge, preferably at its outer edge, to the hydraulic device or parts thereof, so that a membrane space (12) forms which, at least in the presence of a hydraulic pressure, corresponds to the displacement space and contains hydraulic fluid, and the hydraulic device comprises one or more sealing elements which have a pre-tension and are arranged in such a way that they generate an increased pressure, compared with the pressure in the membrane space, on the pressure membrane in the region of its connection to the hydraulic device, whereby the secure connection to the membrane space containing hydraulic fluid becomes absolutely pressure-tight even at very high pressures in the interior of the membrane space.

For the purposes of the invention, "displacement space" is generally taken to mean the space which forms owing to compression of elastomer material by the introduction of a hydraulic fluid into the spring element with generation of a hydraulic pressure, or may already be present beforehand. By contrast, "membrane space" (12) is taken to mean the geometrical space which is formed between the membrane plate (9) and the pressure membrane (5) stretched over the latter. Under pressure, the membrane space generally essentially corresponds to the displacement space.

In particular, the invention relates to a corresponding spring element in which the elastic pressure membrane (5) lies on a membrane plate (9) belonging to the hydraulic device and is connected at its edge to said plate via an elastic bead (10) which fills a T-groove-shaped recess (10a) in the membrane plate, and a non-elastic pressure-increasing element (6), preferably a profiled ring, which has a correspondingly shaped profile which, in the case of pre-tension which has to correspond to a higher pressure than the pressure in the membrane space (12), is forced into the elastic material in the region of the bead of the pressure membrane, and thus leads to specific sealing-off in the region of the T groove of the membrane plate, is arranged between elastomer layer (1) and the pressure membrane in the region of its bead,.

Furthermore, it has been found that the sealing-off is particularly advantageous, stable and complete if a pre-tensioning free space (10c), which can be formed by a corresponding, preferably wedge-shaped, recess in the material of the sealing elements or the corresponding parts of the hydraulic device, is provided between the pressure-increasing element (6) and the elastic bead (10) of the pressure membrane,.

In a particularly simple embodiment, the pressure-increasing element (6), for example the profiled ring, is omitted and functionally replaced by the membrane plate (9), which in this case, however, is compressed in the region of the T groove, so that an increased pressure occurs locally, leading to stronger sealing-off in this region (Fig. 9).

In the embodiments of Figures 6, 7 and 8, the membrane is typically a circular membrane which lies flat on the membrane plate (9). By appropriate introduction of a hydraulic fluid, the membrane is inflated, causing compression of the elastomer layers (1) lying on the other side of the membrane and resulting in the formation of a displacement space or membrane space between membrane plate (9) and the membrane (Fig. 6). In a further advantageous embodiment, the membrane plate (9) has a concavely curved surface, so that, when the pressure membrane (5) is tensioned flat, a membrane space is present, even without pressure being generated, which is formed between the lower side of the flat-tensioned membrane and the concave surface of the membrane plate (Figs. 7, 8).

In a further embodiment, the elastomer layer (1) can also have a concavely curved recess in the region of the area of contact with the pressure membrane (5).

In a particularly advantageous embodiment, both the membrane plate (9) and also the elastomer layer lying over the pressure membrane have a correspondingly concave surface, so that a lenticular membrane space or displacement space is present.

The invention also relates to a corresponding spring element in which the membrane plate (9) has hydraulic supply lines (4) and connections (7) which run through the membrane plate and open out into the membrane space and are able to supply the latter with hydraulic fluid.

In a further series of embodiments of the spring elements according to the invention, they have a support plate (11) for the elastomer layers (1), which preferably have a recess providing space for the membrane plate (9) as a separate part of the support plate.

In a particular embodiment, the membrane plate may in this solution also be tensioned against the support plate itself.

The invention furthermore relates to spring elements of the same principle as described, but which, owing to their space-saving design, have special structural features. Thus, the invention encompasses a spring element in which the elastic pressure membrane (5) has a circumferential terminal elastic T-shaped bead (10), consisting of an outer bead (26) and an inner bead (23), and is connected in a pressure-tight manner to a membrane plate (9) belonging to the hydraulic device in such a way that said beads have been inserted into a correspondingly shaped matching T-shaped hollow profile (10b), which is formed by a correspondingly shaped membrane head (9a) and a pre-tensioning disc (21), where (a) the membrane head is part of the membrane plate (9) and is arranged inside the membrane and the membrane space (12) and has a circumferential projecting lip (24) which accommodates the inner bead (23) of the membrane, and (b) the pre-tensioning plate has a ring-shaped opening or recess which accommodates the membrane head (9a) with the pressure membrane with an accurate fit, and the edge of the opening/recess is formed by a projecting lip which accommodates the outer elastic bead (26) of the membrane.

In a particular embodiment, the pre-tensioning disc (21) lies fully or partly on the membrane plate (9) outside the region of the membrane head (9a) and is arranged between the first elastomer layer (1) or, where appropriate, the elastomer support plate and said membrane plate. In this case, the pre-tensioning disc (21) is preferably securely connected to the elastomer layers (1) outside the region of the pressure membrane.

Furthermore, the membrane disc (9) is preferably tensioned against the pre-tensioning disc (21) by means of fastening means, so that the T-shaped elastic bead (23, 26) of the pressure membrane is compressed.

In accordance with the invention, the pre-tensioning disc or plate (21) may also be support plate for the spring element itself, where in this embodiment it has in the region of the pressure membrane an accurately fitting recess, into which the correspondingly shaped membrane plate (9) is inserted by means of its correspondingly shaped membrane head (9a) and is tensioned against the plate (21) by means of fastening means.

In the embodiments concerned, the membrane plate (9) generally has at least one hydraulic supply line and connection (4, 7), where the hydraulic supply line runs through the membrane head (9a) and opens out into the membrane space (12).

Even the spring elements which occupy a smaller physical space preferably have a pre-tensioning free space (22), which, in the untensioned state, is formed in the region in which outer bead (26), membrane disc (9) and pre-tensioning disc (21) coincide and which closes completely when parts (21) and (9) are tensioned by forcing-in of bead material.

The elastic pressure membrane (5) according to the invention should, in accordance with the invention, have a certain softness and flexibility. Such membranes are preferably suitable for spring elements according to the invention in which a large base area is necessary or desired. In the case of spring elements which must not take up much space, such large-area pressure membranes are undesired and are, in accordance with the invention, replaced by the conical or paraboloid membranes, which are preferably employed in a rigid and less flexible design, but one which is elastic with regard to stretching. The invention thus also relates to spring elements in which the pressure membrane (5) is stiff and has a paraboloid or conical spatial shape, which, with the membrane head (9a) located on the base, forms a correspondingly shaped conical hat-like membrane cavity (12) or a membrane conical hat (12a).

In particular, the invention also relates to corresponding spring elements in which the elastomer layers (1) have, in the core of the spring element, a conical cavity which is shaped in a manner corresponding to the membrane cavity (12) and is filled by the membrane conical hat (12a) formed by the pressure membrane. Such spring elements can be modified in such a way that the membrane cavity (12) of the pressure membrane conical hat (12a) has a stop device (50), which prevents the conical pressure membrane from collapsing irreversibly in the event of overloading of the system. Such a stop device is made from a non-elastic material, preferably in the form of a conical hat which is arranged centrally in the interior of the membrane cavity (12) on the base area of the membrane head (12a) and has a supply conduit and a connection to the hydraulic supply line (4). The supply conduit can also be designed in the form of an annular gap.

In general, the hydraulic devices described, including pressure membranes and sealing elements, can be mounted on both (the upper and lower) sides of the spring element. Such spring elements are particularly suitable if they have to be very large, so that a pressure generated on one side may be unfavourable. In most cases, it is sufficient for the spring element to be fitted with a corresponding device on one side only.

In such cases, a spring element of this type can be equipped in such a way that it has, on the side opposite the hydraulic device, a recess or cavity (40) in the elastomer material in the core region of the spring element, which recess or cavity is shaped and designed in such a way that it is separated from the membrane space (12) by means of a bridge (42) of elastomer material of the layers (1), so that the bridge can be forced into the cavity or the recess when the hydraulic pressure in the membrane space is increased. As a result, a greater range of adjustment of the stiffness is possible, particularly if the recess or cavity (40) additionally has a non-elastic calibration element (3, 8), with the aid of which the volume of the recess or cavity (4) can be changed.

The spring elements described are extremely suitable for being employed as bearings for machines or gearboxes. The invention thus relates to a bearing arrangement, comprising at least two opposite spring elements which are aligned or arranged in the same direction and thus act in the same way with respect to their damping properties and between which the gearbox or machine is mounted, as described above, below and in the claims. In order to achieve an optimum damping action, the spring elements according to the invention are connected to one another by hydraulic lines (34, 35) arranged crosswise, thereby providing the bearings with the further described properties. Furthermore, they can be equipped with at least one pressure relief valve (31) and a throttle valve (33, 36, 37) for specific control when the turbines are under load, as likewise explained below.

Finally, the invention relates to the use of the bearings and spring element described. These are intended for use in machines, but in particular wind turbines, where they serve for the absorption of flexural, deformation and displacement forces which occur, particularly in the extreme load case, in the vertical direction with respect to the mounted spring elements.

The figures are explained briefly below:

Fig. 1	Entire system
Fig. 2	Gearbox with elastomer - hydraulic torque bracket
Fig. 3	Properties in the case of torque
Fig. 4	Properties in the case of displacement
Fig. 5	Stiffness characteristic
Fig. 6	Flat membrane
Fig. 7	Flat membrane exploded drawing

Fig. 8	Flat membrane detail of pressure-increasing ring
Fig. 9	Flat membrane increase in pre-tension by deformation of the membrane plate
Fig. 10	Flat membrane increase in pre-tension by deformation of the membrane plate
Fig. 11	Structure with parabolic membrane
Fig. 12	Structure with parabolic membrane exploded drawing
Fig. 13	Parabolic membrane detail of pre-tensioning free space
Fig. 14	Pre-tensioning via inner disc
Fig. 15	Membrane vulcanised on with closable opening for tool penetration
Fig. 16	Hydraulic line with pressure relief valve and throttle
Fig. 17	Parabolic (conical) membrane with cavity in elastomer core
Fig. 18	Parabolic (conical) membrane with cavity in elastomer core and tensioning screw in hole of the membrane plate
Fig. 19	Parabolic (conical) membrane with cavity in the elastomer core and limit stop in the membrane space
Fig. 20	Membrane plate with holes and filling line and filling cock
Fig. 21	Radial stop
Fig. 22	Arrangement on generator on the case of gearless installations
Fig. 23	Section from Fig. 22

The references used in the text, claims and figures are likewise explained:

For Figs. 1 - 5	
01	Rotor
02	Rotor bearing
03	Rotor shaft
04	Gearbox
05	Elastomer-hydraulic torque bracket
06	Elastomer component pressure side bottom
07	Elastomer component relief side top
08	Elastomer component pressure side top
09	Elastomer component relief side bottom
010	Yoke for torque bracket
011	Gearbox-side torque bracket
012	Vertical displacement direction
013	Torque

For Figs. 6 - 23	
1	Elastomer
2	Intermediate plates
3	Calibration element
4	Fluid supply
5	Flat membrane
6	Pressure-increasing ring
7	Hydraulic connection
8	Calibration hole
9	Membrane plate
9a	Membrane head
10	Elastic sealing bead
10a	T-groove-shaped recess in (9)
10b	T-groove-shaped hollow profile (Figs. 11, 12)
10c	Pre-tensioning free space
11	Lower support plate
12	Membrane space (displacement space)
12a	Parabolic (conical) membrane, conical hat
13	Screws or holes for membrane tensioning
21	Pre-tensioning ring, pre-tensioning plate
22	Pre-tensioning free space
23	Inner bead
24	Inner-bead clamping lip
25	Parabolic membrane
26	Outer bead
28	Spacing from membrane plate
31	Pressure relief valve
32	Solenoid valve
33	Throttle in connecting train
34	Pipeline pressure side
35	Pipeline relief side
36	Throttle in main train relief side
37	Throttle in main train pressure side
40	Cavity for volume increase

41	Cavity vent
42	Separating bridge (elastomer) between membrane space and cavity 40
43	Filling cock
44	Filling line
50	Limit stop
51	Fluid conduit through limit stop 50
52	Stop ring
53	Plate of greater diameter and greater thickness
54	Generator gearless installation
55	Bedplate
56	Elastomer-hydraulic bearing
57	Bedplate bearing
58	Generator-side bearing

The elastomer materials used for the layers (1) according to the invention essentially consist of a natural rubber, a natural rubber derivative or of a suitable elastic polymeric plastic or plastic mixture. The elastomer layer may, in accordance with the invention, have different hardness ("Shore hardness") and different damping properties, according to the desired requirements. Elastomers having a Shore A hardness of 20 to 100, in particular 30 to 80, are preferably used. The preparation of such elastomers of different hardness is known in the prior art and adequately described in the relevant literature.

The non-elastomeric intermediate plates or interlayers (2) are, in accordance with the invention, made from very substantially non-elastic materials of low compressibility. These are preferably metal sheets, but other materials, such as hard plastics, composite materials or carbon-fibre-containing materials, can also be used. The intermediate plates and the elastomer materials (4) are generally connected to one another by vulcanisation.

The pressure membrane (5) is generally likewise made from natural rubber or a natural rubber derivative. The material must be tear- and pressure-resistant over a large range. It preferably has a smooth, dense structure which does not have any surface pores, or only has pores of very small diameter which are not capable of absorbing microdroplets of the hydraulic fluid.

Hydraulic fluids that can be used are generally all conventional pressure-stable fluids. For wind turbines, water, to which antifreeze, alcohols or additives have optionally been added, is also suitable, for example. Likewise, hydraulic oils can also be used.

The spring element according to the invention can be employed in two basic design embodiments. The first embodiment is based on a flat membrane (5) which occupies a relatively large area on the membrane plate (9). The displacement space, or membrane space (5) is formed, in particular, by the inflation of the membrane lying flat on the membrane plate, with simultaneous compression of the elastomer layers (1) above the pressure membrane. Recesses, preferably of concave type, in the elastomer layer above the membrane and/or in the membrane plate (9) below the membrane can create a displacement space even in the unpressurised state. Whereas the flat membrane (Fig. 6) has a relatively flat, preferably lenticular shape, the second basic embodiment is represented by the parabolic or conical membrane (Figs. 11 - 19), which represents said conical shape or a paraboloid even in the unpressurised state. The membrane essentially represents a sealing function between the fluid and the elastomer body.

Figs. 6 and 11 - 19 illustrate an elastomer component having a plurality of layers (e.g. 4 layers). The special feature compared with conventional layered springs is that the intermediate plates (2), which are preferably circular, have a hole, preferably in the centre, so that the non-elastic structure is present only in the outer region. In the inner region, the core region, only elastomer (1) is present, preferably having a recess in the region which accommodates the membrane area. By introducing more or less fluid into the membrane, elastomer volume is displaced, resulting in an increase in stiffness for the same physical height of the element as a whole.

The spring elements according to the invention comprise a membrane plate (9), which contains an opening for connection of the hydraulic lines. Furthermore, the membrane plate contains an ring-shaped T groove (10), into which the flat membrane (5) is vulcanised. The special feature of the vulcanisation is that bonding occurs only in the region of the T groove. The remaining region remains without bonding, resulting in the formation of a cavity for the introduction of the hydraulic medium into the membrane space (12). In accordance with the invention, the leak-tightness of the T-shaped connection is very important. In the prior art, this is achieved on the one hand by applying a binder into the T groove prior to the vulcanisation, so that a rubber-metal bond results.

According to the prior art, this rubber-metal bond is said to be absolutely leak-tight. As already described in detail above, however, practical tests on which this invention is based have shown that this is not the case, since the droplet migration discovered by the inventor is responsible for this and leads to the binder between rubber and metal being splintered off fractionally on dynamic loading. It becomes clear from this that a higher pressure must be generated at these locations than in the other regions in order to achieve leak-tightness. This higher pressure is achieved in accordance with the invention by the pressure-increasing ring (6). This lies on the membrane and, owing to the high pressure and the relatively large upper surface, receives a pre-pressure, by means of which it can be forced into the T groove of the membrane and thus generates a pressure increase in the T-shaped region (10), thereby preventing the binder from being splintered off by ingressing hydraulic fluid. The pressure-increasing ring can be designed in a wide range of embodiments here. The crucial factor is ultimately that the pressure in the region is increased by a pre-tension.

A simple variant dispenses with a dedicated pressure-increasing ring. In these variants, described in Figs. 9 and 10, the T-shaped slot is introduced into the membrane plate (9) and vulcanised. The outer region of the membrane plate is subsequently bent inward, causing a pressure increase to take place in the region of the T groove (10) and ensuring leak-tightness.

In the embodiments of the flat-membrane spring elements (Fig. 6), pressures of 100 bar to 700 bar are intended. A further possibility for sealing off the elements is offered, as mentioned, by the cross-sectionally parabolic membrane (Figs. 11 - 15, 17 - 20), which adopts a conical or paraboloidal spatial shape. The basic structure is similar to that in the case of the flat membrane, but this design requires a smaller area.

This enables a smaller installation space and reduces the costs of the parts. At the same time, however, higher pressures arise owing to the smaller area for the same load. For leak-tight attachment of the parabolic membrane to the membrane plate (9), this is clamped by means of two beads. For this purpose, the parabolic membrane (25) is pulled over the inner-bead clamping ring lip (24) of the membrane head (9a) like a tyre, so that the inner bead (23) is introduced, in a virtually air-free manner, behind the lip (24) with a slight pre-tension. The membrane plate (9) is subsequently screwed to the pre-tensioning ring (21). However, the outer bead (26) has a significantly larger dimension than the cavity between (21) and (9), so that a pre-tension results here. This pre-tension amounts to approximately 10% to 40% of the thickness of the outer bead (26), so that part of the volume of the outer bead is displaced towards the inner bead (23), so that a pre-tension also arises here and leak-tightness of the bead is thus achieved. In this type of design, the problem may occur that the outer bead (26) is displaced towards both sides on pre-tensioning and thus slides between the pre-tensioning ring (21) and the membrane plate (9) before they come together and seal the gap. The outer bead (26) would thus be jammed between pre-tensioning ring (21) and membrane (9). In order to prevent this, a preferably wedge-shaped pre-tensioning free space (22) is provided. This pre-tensioning free space is approximately 2 - 3 mm deep, i.e. deeper than the material flows under the given pre-tension, so that undefined jamming of the elastomer from the outer bead (26) is prevented. The clamping can take place without additional medium. For more secure attachment, it is also possible also to introduce adhesive into the joint areas, so that an even greater security of the leak-tightness is ensured here. In the case of spring elements based on a parabolic membrane, pressures of 200 to 1500 bar are intended owing to the smaller physical size. The membrane can withstand this pressure only if it is introduced into the conical cavity of the elastomer component cleanly and at all points and does not come into contact with the hard intermediate plates. During the operation of the membrane, it will expand to such an extent that the elastomer space within the central holes of the intermediate plates (2) is displaced to a very large extent, i.e. to about 80%, so that the membrane comes relatively close to the intermediate plates. A minimum spacing (28) from the intermediate plates (2) is therefore required. This should be at least 5 mm. The pre-tension between membrane plate (9) and pre-tensioning plate (21) can also be reversed in a such a way that the inner-bead clamping region of (9) is separated and can be pulled closer by means of screws (Fig. 14). Here too, a pre-tensioning free space (22) should be provided.

The spring element installed in a wind turbine, and bearing arrangements containing such elements, work as follows with respect to the damping of the installation: the gearbox has a gearbox-side torque bracket (11) on both sides (Figs. 1, 2). This is embraced by a yoke (10). Elastomer elements (06, 07, 08 and 09) are arranged between the gearbox-side torque bracket (11) and the yoke (10). The elastomer elements contain a cavity which is filled with fluid. These cavities are interconnected, as illustrated in Figs. 3 and 4, by hydraulic lines (34, 35), which may be pipelines or hoses. In the case of the normal torque occurring in the wind turbine, the spring elements (06, 08) are loaded. They are interconnected by the hydraulic line (34) in such a way that, due to the pressure which arises at the same time, no fluid movement takes place. Likewise, the elastomer elements (07, 09) are simultaneously relieved, so that hydraulic fluid is not moved in the hydraulic line (35) either. In the torque direction, the springs thus exhibit a stiff spring characteristic, i.e. they are not influenced by the hydraulic system.

As already mentioned, the described spring elements according to the invention are particularly suitable for use as bearing points in two-, four- or multipoint bearing arrangements, preferably in wind turbines. A bearing point preferably comprises two elastomer spring systems, which hold the machine parliament to be supported in one point (e.g. top and bottom) (Fig. 1). An elastomer spring system of this type preferably consists of an elastomer spring according to the invention, but may also consist of an entire arrangement of correspondingly arranged springs. In the case of the hydraulically regulated two-point bearing arrangement of a wind turbine depicted in Figure 2, two bearing points, each with two elastomer springs, are used. The two bearing points are connected to one another by hydraulic lines (Fig. 3 and 4), in such a way that one elastomer spring of one bearing point is connected to the opposite elastomer spring of the other opposite bearing point (crosswise connection). If the elastomer layer in one bearing point is now compressed by forces acting from the machine region, the hydraulic fluid is inevitably forced through the corresponding line into the opposite bearing point’s spring element which is connected to it and compresses its elastomer layer here. This can result in optimal stiffness regulation and thus damping.

The elastomer-hydraulic bearing arrangement is likewise of importance for gearless installations. In these installations, the generator is equally arranged directly – without elastic coupling – on the rotor shaft. The system thus has the same problems as geared installations. Here too, radial displacement forces enter the rotor. The air gap between the rotor sitting on the rotor shaft and the stator of the generator, which is attached to the chassis, must be kept as small as possible for good efficiency and is, however, changed by the radial constraining forces occurring in accordance with the prior art.

The constraining forces which occur in addition to the undesired displacement behave like in the same way as in a gearbox.

Fig. 4 shows the vertical displacement due to machine deformation. In the case of displacement of the gearbox upwards from the bedplate, the following occurs: the spring elements (07) and (08) experience an increase in pressure, while the pressure drops in the elements (06) and (09). Due to the crosswise laying of the connecting lines of the spring elements (08) to (06), the fluid flows from the upper part, which is loaded to a greater extent, to the unloaded spring element (06), which causes a reduction in the resistance force of the elastomer element (08), while at the same time the fluid reduction occurring in the elastomer element (06) is replenished by the fluid from the spring element (08). The same principle is used between the spring elements (07) and (09), so that this vertical movement takes place with significantly smaller spring stiffness.

Fig. 5 shows a diagram with the high stiffness in the torsion direction and the lower stiffness on vertical displacement of the gearbox with respect to the bedplate. Thus, in the system according to the invention, high torsional stiffness, which is required for the connection between gearbox and generator, can be achieved, while at the same time low displacement stiffness between gearbox and bedplate is achieved, which leads to the reduction in constraining forces and hence to the relief of gearbox bearings and rotor bearings. The torsional stiffness here is greater than the vertical displacement stiffness by a factor of 4 to 100, preferably 4 to 15, in particular 6 to 10.

In order to achieve the large stiffness ratios, as many layers as possible (ideally 5 to 10 layers) are necessary. Since the radial stiffness also drops with a large number of layers, the components tend to buckle radially outwards in the case of the particularly high extreme loads which occur in rare cases. In order to prevent this, an intermediate plate is made with a greater diameter and greater thickness in the central region of the elements item 53 (FIG 21). In order to prevent the layer from buckling outwards, an additional fixed stop ring item 52 is placed around the component with a separation of 3mm to 6mm, which prevents this from buckling outwards. Both parts, 52 and 53, are made from rust-free material in order to prevent corrosive metal dust – abrasion contact of item 53 with item 52 in the extreme load case.

In the extreme load case, as already explained, significantly higher torques occur than in normal operation. The extreme load case describes a critical situation of the wind turbine. The system according to the invention provides that a pressure relief valve (31) is fitted between the pressure line (34) subjected to torque on the relief side (35), which is under weak pressure in this state. If, in this rarely occurring extreme load case, a high pressure occurs in the pressure line (34), the pressure relief valve (31) is opened. The fluid thus flows from the pressurised line (34) into the line (35) with less pressure. The flow rate in this line is reduced by the throttle (33), so that a strong damping function arises. Thus, the loads on the wind turbine that occur under extreme load are reduced with the aid of the valve function (31) of the throttle (33). The elastomer components are designed so that they can also continue to be operated without fluid. On the next stoppage, which occurs at intervals or is initiated after an extreme load situation, the valve (32) is automatically opened, so that the same pressure again arises in the connecting lines (34 and 35), which is the case after a short time. The system is subsequently in the original state again, and the pressure switch (32) can thus be reclosed.

Besides the reduction in vibration, the above-described valve function also has the advantage that the hydraulic lines do not have to be designed for the pressure prevailing under extreme load. Furthermore, the extreme loads can be reduced by the damping occurring in the connecting line with the additional throttle (33). Whereas the pressure relief valve (31) with the throttle (33) are effective only in the extreme load case, further throttles (36 and 37) can be arranged on the pressure side and relief side. These introduce damping into the entire system while the installation is operating. In other words, any displacement of the machine frame is generally to be associated with vibrations of the entire installation. These vibrations are damped by the throttles (36 and 37) present, which brings about a load reduction in the wind turbine. The throttles (36 and 37) may be installed in the form of adjustable throttles in accordance with the prior art. However, it is also possible to dimension all hydraulic lines in such a way that the throttling function is already brought about by the fluid friction in the lines.

All functions described hitherto are passive, i.e. without any outside action. However, it is also possible to operate the system actively. To this end, fluid is alternately pumped into and released from the various chambers, so that a forced movement counteracting the undesired vibrations and movements takes place with this system in the force flow path of the wind turbine.

The elastomer-hydraulic bearing arrangement can likewise be employed in gearless installations. In these installations, the generator is equally arranged directly – without elastic coupling – on the rotor shaft. The system thus has the same problems as geared installations.

On flexing of the rotor shaft and bedplate, unavoidable deformations also occur here.

The radial displacement forces act on the rotor and stator here and thus cause constraints and displacements. The air gap between the rotor sitting on the rotor shaft and the stator of the generator, which is attached to the chassis, must be kept as small as possible for good efficiency.

Due to the radial constraining forces occurring in accordance with the prior art, this is changed and must be correspondingly large in order to prevent contact between rotor and stator.

The constraining forces occurring in addition to the undesired displacement behave in the same way as in a gearbox. They cause additional loads on the bearing arrangement.

Support of the generator 54 on the bedplate 55 – as in the case of the gearbox bearing arrangement – requires at least one pair, consisting of two bearing element pairs, which are connected to one another crosswise by the hydraulic lines described.

In order to introduce the forces into the generator as evenly as possible, it is possible to arrange a plurality of pairs.

As an example, 2 pairs are arranged in Fig. 22.

The element can be arranged axially on the generator – as described in Fig. 22. Figure 22 shows the arrangement of the bearing arrangement behind the generator on the side facing away from the rotor. An arrangement in front of the generator, in the direction of the rotor, is likewise possible.

It is also possible to place the bearing arrangement on the generator periphery in the radial extension. The combination of the two systems or of the three systems mentioned is likewise possible and certainly appropriate in the case of very large generators, greater than 5MW. In summary, the disclosure relates to the following components, devices and uses of said components and devices:

- Spring element whose stiffness can be adjusted by hydraulically generated pressure, comprising

  (i) two or more elastomer layers (1),
  (ii) one or more non-elastic interlayers (2) and
  (iii) at least one hydraulic device (4, 7, 9, 11, 12);

where the elastomer layers are separated from one another by the non-elastic interlayers and the non-elastic interlayers have a centrally arranged opening or hole, so that the spring element has a continuous elastomer core region, at least in the interior, to which the individual elastomer layers are connected, and the hydraulic device is mounted at the upper and/or lower end of the layer-like spring element and, by introduction of a hydraulic fluid into the spring element by means of hydraulic pressure, causes compression of the elastomer material (1) in the vertical direction with respect to the layers of the spring element and thus leads to formation or enlargement of a displacement space containing hydraulic fluid in the elastomeric core region of the spring element, where an elastic, correspondingly shaped pressure membrane (5), (25) is arranged between the elastomer layers (1) of the spring element and the displacement space and is securely connected at its edge to the hydraulic device or parts thereof, resulting in the formation of a membrane space (12) containing hydraulic fluid, which corresponds to the displacement space in the case of a hydraulic pressure, and the hydraulic device is connected to the membrane space via a hydraulic line (4) and comprises one or more sealing elements, (10, 10a, 6, 21, 22, 24), which have a pre-tension and are arranged in such a way that they generate an increased pressure, relative to the pressure in the membrane space, on the pressure membrane in the region of its secure connection to the hydraulic device, so that the membrane space is sealed off in a pressure-tight manner from the layers of the spring element on the one hand and from the hydraulic device or parts thereof on the other hand.

- A corresponding spring element in which the elastic pressure membrane (5) lies on a membrane plate (9) belonging to the hydraulic device, and is connected thereto at its edge via an elastic bead (10), which fills a T-groove-shaped recess (10a) in the membrane plate, and a non-elastic pressure-increasing element (6) is arranged between elastomer layer (1) and the pressure membrane in the region of its bead, which has a correspondingly shaped profile which is forced into the elastic material in the region of the bead of the pressure membrane in the case of pre-tensioning and thus leads to specific sealing off in the region of the T groove of the membrane plate.

- A corresponding spring element in which the non-elastic pressure-increasing element (6) is a profiled ring.
- A corresponding spring element in which a pre-tensioning free space (10c) is present between pressure-increasing element (6) and elastic bead (10) of the pressure membrane.
- A corresponding spring element in which the pressure-increasing element has been replaced by the membrane plate (9) itself, which is compressed in the region of the T groove, so that sealing occurs in this region.
- A corresponding spring element in which the membrane plate (9) represents the base area of a membrane space (12) having a convex surface.
- A corresponding spring element in which the membrane plate is curved in a concave manner.
- A corresponding spring element in which the elastomer layer (1) is curved in a concave manner in the region of the area of contact with the pressure membrane (5).
- A corresponding spring element in which a lenticular membrane space or displacement space is present.
- A corresponding spring element in which the membrane plate (9) has hydraulic supply lines (4) and connections (7) which run through the membrane plate and open out into the membrane space.
- A corresponding spring element in which the spring element has a support plate (11) for the elastomer layers (1).
- A corresponding spring element in which the membrane plate (9) is a separate part of the support plate (11).
- A corresponding spring element in which the membrane plate is tensioned against the support plate.
- A corresponding spring element in which the elastic pressure membrane (25) has a circumferential terminal elastic T-shaped bead (10), consisting of an outer bead (26) and an inner bead (23), and is connected in a pressure-tight manner to a membrane plate (9) belonging to the hydraulic device in such a way that said beads have been inserted into a correspondingly shaped matching T-shaped hollow profile which is

formed by a correspondingly shaped membrane head (9a) and a pre-tensioning disc (21), where (a) the membrane head is arranged inside the membrane and membrane space (12) and has a circumferential projecting lip which accommodates the inner bead (23) of the membrane, and (b) the pre-tensioning plate has a ring-like opening which accommodates the membrane head (9a) with the pressure membrane with an accurate fit, and the edge of the opening is formed by a projecting lip which accommodates the outer elastic bead (26) of the membrane.

- A spring element corresponding to that described in the previous paragraph, in which the pre-tensioning disc (21) lies fully or partly on the membrane plate (9) and is arranged between this and the elastomer layer (1).
- A corresponding spring element in which the pre-tensioning disc (21) is support plate for the spring element and has in the region of the pressure membrane an accurately fitting recess, into which the correspondingly shaped membrane plate (9) is inserted.
- A corresponding spring element in which the pre-tensioning disc (21) is securely connected to the elastomer layers (1) outside the region of the pressure membrane.
- A corresponding spring element in which the membrane disc (9) is tensioned against the pre-tensioning disc (21) by means of fastening means, so that the T-shaped elastic bead (23, 26) of the pressure membrane is compressed.
- A corresponding spring element in which the membrane plate (9) has hydraulic supply line and connection (4, 7), where the hydraulic supply line runs through the membrane head (9a) and opens out into the membrane space (12).
- A corresponding spring element in which a pre-tensioning free space (22) is provided, which is formed in the untensioned state in the region in which outer bead (26), membrane disc (9) and pre-tensioning disc (21) coincide, and which is completely sealed by bead material being forced in on tensioning of the parts (21) and (9).
- A corresponding spring element in which the pressure membrane (25) is rigid and has a paraboloidal or conical surface which, with the membrane head (9a) located on the base, forms a correspondingly shaped conical hat-like membrane cavity (12) or a membrane conical hat (12a).
- A corresponding spring element in which the elastomer layers (1) have, in the core of the spring element, a conical cavity which is shaped in a manner corresponding to the membrane cavity (12) and is filled by the membrane conical hat (12a) formed by the pressure membrane.

- A corresponding spring element in which the membrane cavity (12) of the pressure membrane conical hat (12a) as a stop device (50) which prevents the conical pressure membrane from collapsing irreversibly in the event of overloading of the system.
- A corresponding spring element in which the stop device (50) is made from a non-elastic material in the form of a conical hat which is arranged centrally in the interior of the membrane cavity (12) on the base area of the membrane head (12a) and has a supply conduit and connection to the hydraulic supply line (4).
- A corresponding spring element which has a corresponding hydraulic device on one side.
- A corresponding spring element which has, on the side opposite the hydraulic device in the core region of the spring element, a recess or cavity (40) in the elastomer material which is shaped and designed in such a way that it is separated from the membrane space (12) by means of a bridge (42) of elastomer material of the layers (1), so that the bridge can be forced into the cavity or recess when the hydraulic pressure in the membrane space increases.
- A corresponding spring element in which the recess or cavity (40) has a non-elastic calibration element (3, 8), with the aid of which the volume of the recess or cavity (4) can be altered.
- Machine or gearbox bearing, comprising at least two spring elements, as described in greater detail above and below, oriented in the same direction.
- A corresponding bearing in which the spring elements are connected to one another by hydraulic lines (34, 35) arranged crosswise.
- A corresponding bearing in which the hydraulic lines are fitted with at least one pressure-relief valve (31) and a throttle valve (33, 36, 37).
- Use of a corresponding bearing for the absorption of flexural, deformation and displacement forces in wind turbines.
- A corresponding use for the absorption of flexural, deformation and displacement forces in the extreme-load case which occur vertically with respect to the installed spring elements.
- A corresponding use for gearless installations.

Patentkrav

1. Fjederelement der er justerbart i sin stivhed ved hydraulisk genereret tryk, hvilket fjederelement er egnet til tryk på fra 100 bar til 1500 bar, omfattende

(i) to eller flere elastomerlag (1) og et eller flere ikke-elastiske mellemlag (2), hvor elastomerlagene er adskilt fra hinanden af de ikke-elastiske mellemlag og de ikke-elastiske mellemlag har en centralt anbragt åbning eller hul, så at fjederelementet har, mindst i det indre, en kontinuerlig elastomer kerneregion til hvilken de individuelle elastomerlag er forbundet, og

(ii) mindst en hydraulisk indretning (4, 7, 9, 11, 12), der er monteret ved den øvre og/eller nedre ende af det lag-lignende fjederelement og forårsager kompression af elastomermaterialet (1) i den vertikale retning til lagene af fjederelementet ved indføring af en hydraulisk fluid i fjederelementet ved hjælp af hydraulisk tryk og således fører til dannelsen eller udvidelse af et fortrængningsrum indeholdende hydraulisk fluid i den elastomere kerneregion af fjederelementet, hvor

(a) en elastisk, tilsvarende dannet trykmembran (5), (25) er anbragt mellem elastomerlagene (1) af fjederelementet og fortrængningsrummet og er fast forbundet ved dets kant til den hydrauliske indretning eller dele deraf, så at et membranrum (12) indeholdende hydraulisk fluid dannes, der svarer til fortrængningsrummet i tilfældet af et hydraulisk tryk, og

(b) den hydrauliske indretning er forbundet til membranrummet via en hydraulisk linje (4),

kendetegnet ved at den hydrauliske indretning omfatter ét eller flere tætningselementer (10, 10a, 6, 21, 22, 24) der har en forspænding og er indrettet på en sådan måde at de genererer et øget tryk, sammenlignet med trykket i membranrummet, på trykmembranen i regionen af deres faste forbindelse til den hydrauliske indretning, så at membranrummet er aflukket på en tryk-tæt måde fra lagene af fjederelementet på den ene side og fra den hydrauliske indretning eller dele deraf på den anden side.

2. Fjederelement ifølge krav 1, **kendetegnet ved at** den elastiske trykmembran (5) ligger på en membranplade (9) der hører til den hydrauliske indretning, og er forbundet ved dens kant til denne plade via en elastisk vulst (10), der udfylder en T-rilleformet udsparing (10a) i membranpladen, og et ikke-elastisk trykøgningselement (6), der har et tilsvarende formet profil der, i tilfældet af forspænding, er presset ind i det elastiske materiale i regionen af vulsten af trykmembranen og således fører til specifik tætning i regionen af T-rillen af membranpladen, er anbragt mellem elastomerlag (1) og trykmembranen i regionen af dets vulst.

3. Fjederelement ifølge et af kravene 1 or 2, **kendetegnet ved at** et forspændingsfrirum (10c) er til stede mellem trykøgningselement (6) og den elastiske vulst (10) af trykmembranen.

4. Fjederelement ifølge krav 2, **kendetegnet ved at** trykøgningselementet er blevet erstattet med membranpladen (9) selv, der er sammenpresset i regionen af T-rillen, så at en tætning sker i denne region.

5. Fjederelement ifølge et af kravene 1 til 4, **kendetegnet ved at** det elastomere lag (1) er buet på en konkav måde i regionen af kontaktområdet med trykmembranen (5), så at et linseformet membran- eller fortrængningsrum er til stede.

6. Fjederelement ifølge et af kravene 1 til 5, **kendetegnet ved at** membranpladen (9) har hydrauliske forsyningslinjer (4) og forbindelser (7) der løber gennem membranpladen og åbner ud i membranrummet.

7. Fjederelement ifølge et af kravene 1 til 6, **kendetegnet ved at** fjederelementet har en bærerplade (11) til elastomerlagene (1).

8. Fjederelement ifølge krav 7, **kendetegnet ved at** membranpladen (9) er en separat del af bærerpladen (11) og er spændt mod sidstnævnte.

9. Fjederelement ifølge krav 1, **kendetegnet ved at** den elastiske trykmembran (25) har en periferisk ende-elastisk T-formet vulst (10), bestående af en ydre vulst (26) og en indre vulst (23), og er forbundet på en tryk-tæt måde til en membranplade (9) der hører til den hydrauliske indretning på en sådan måde at vulsterne er blevet indført i en tilsvarende formet matchende T-formet hul profil, der er dannet af et tilsvarende formet membranhoved (9a) og en forspændingsskive (21), hvor (a) membranhovedet er anbragt inde i membranen og membranrummet (12) og har en periferisk fremskudt læbe, der rummer membranens indre vulst (23), og (b) forspændingspladen har en ring-formet åbning der rummer membranhovedet (9a) med trykmembranen med en akkurat tilpasning, og kanten af åbningen er dannet af en fremskudt læbe, der rummer den ydre elastiske vulst (26) af membranen.

10. Fjederelement ifølge krav 9, **kendetegnet ved at** forspændingsskiven (21) ligger helt eller delvist på membranpladen (9) og er indrettet mellem sidstnævnte og elastomerlaget (1).

11. Fjederelement ifølge krav 9, **kendetegnet ved at** forspændingsskiven (21) er bærerplade for fjederelementet og har i regionen af trykmembranen en nøjagtig tilpasningsudsparing, i hvilken den tilsvarende formede membranplade (9) indsættes.

12. Fjederelement ifølge et af kravene 9 til 11, **kendetegnet ved at** membranpladen (9) har en hydraulisk forsyningslinje og tilslutning (4, 7), hvor den hydrauliske forsyningslinje løber gennem membranhovedet (9a) og åbner ind i membranrummet (12).

13. Fjederelement ifølge et af kravene 9 til 12, **kendetegnet ved at** et forspændingsfrirum (22) er tilvejebragt, der, i den uspændte tilstand, er dannet i regionen i den ydre vulst (26), membranskive (9) og forspændingsskive (21) sammenfalder, og der lukkes fuldstændigt når delene (21) og (9) spændes af trykket i vulstmaterialet.

14. Fjederelement ifølge et af kravene 9 til 13, **kendetegnet ved at** trykmembranen (25) er stiv og har en paraboloid- eller konisk overflade, der med membranhovedet (9a) anbragt på basis danner et tilsvarende formet konisk hat-lignende membranhulrum (12) eller en membrankeglehat (12a).

15. Fjederelement ifølge krav 14, **kendetegnet ved at** elastomerlagene (1) i kernen af fjederelementet har et konisk hulrum, formet tilsvarende til membranhulrummet (12), der fyldes med membrankeglehatten (12a) dannet af trykmembranen.

16. Fjederelement ifølge et af kravene 1 til 15, **kendetegnet ved at** det på en side har en tilsvarende hydraulisk indretning og på den modstående side har en hulrums-formet udsparing (40) i elastomermaterialet, der er formet og udformet på en sådan måde at det er adskilt fra membranrummet (12) ved hjælp af et mellemstykke (42) af elastomermateriale dannet af lagene (1), så at mellemstykket kan presses ind i den hulrums-formede udsparing når det hydrauliske tryk i membranrummet tiltager.

17. Maskin- eller gearkasseleje, der omfatter mindst to fjederelementer ifølge kravene 1 til 16 justeret i den samme retning.

18. Anvendelse af et leje ifølge krav 17 til at affjedre bøjnings-, deformations- og fortrængningskræfter i vindkraftanlæg til anlæg udstyret med en gearkasse og til gearløse anlæg.

Fig. 1

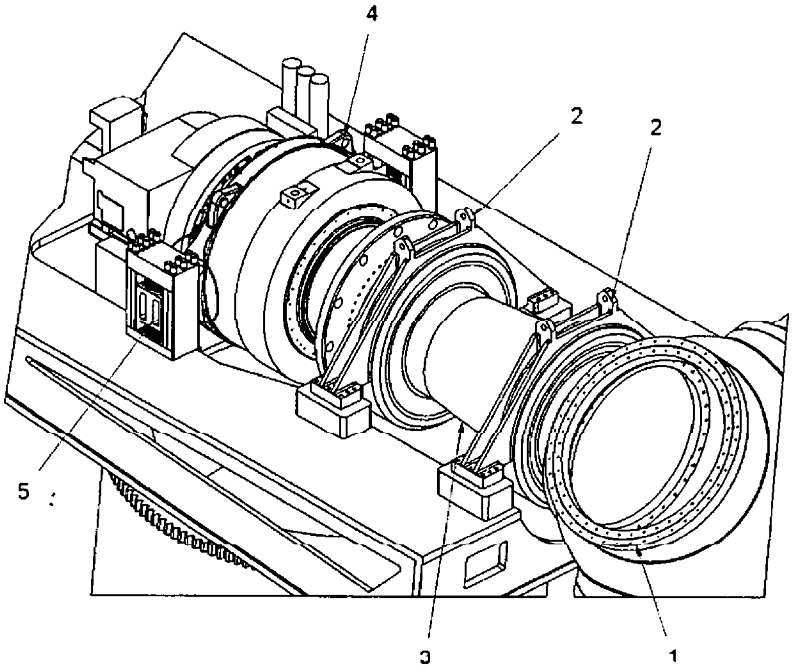

Fig. 2

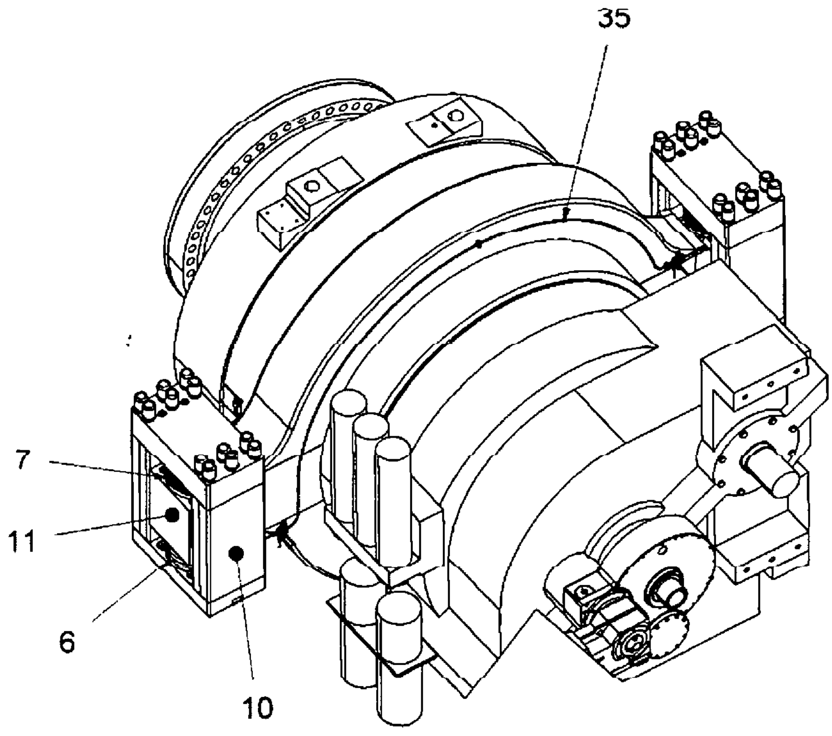

Fig. 3

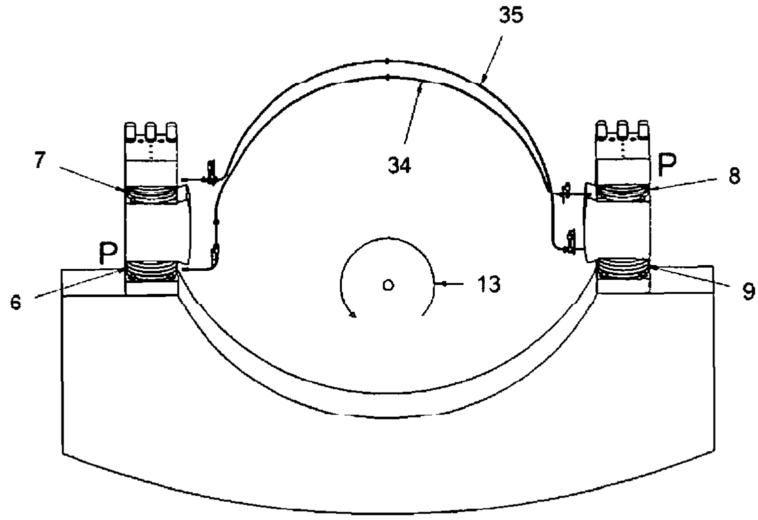
35
34
7
P
8
P
13
6
9

Fig. 4

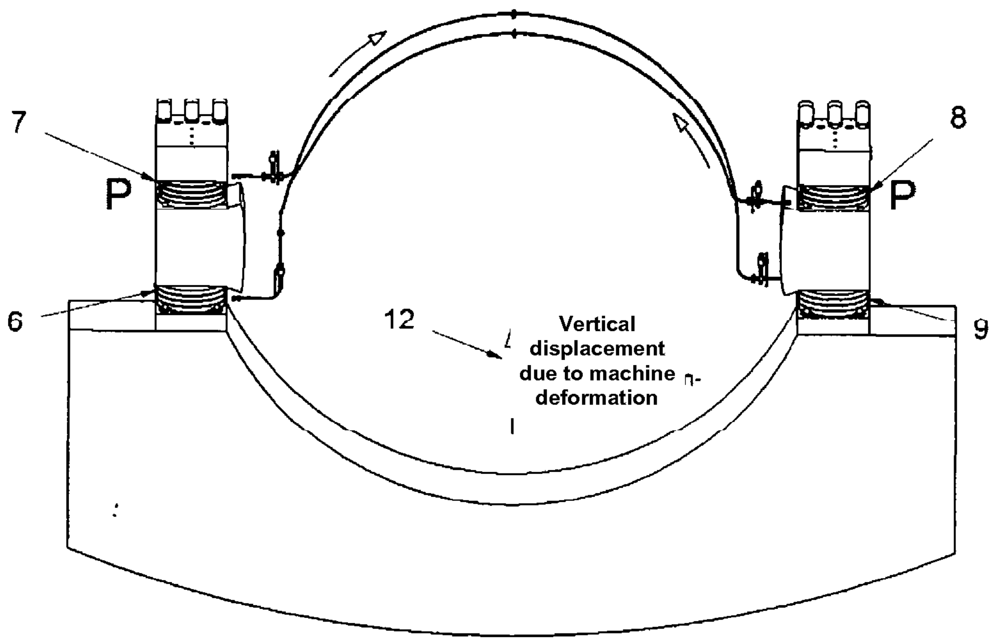
7
8
P
P
6
12
Vertical
displacement
due to machine
deformation
9

Fig. 5

Figure: 5

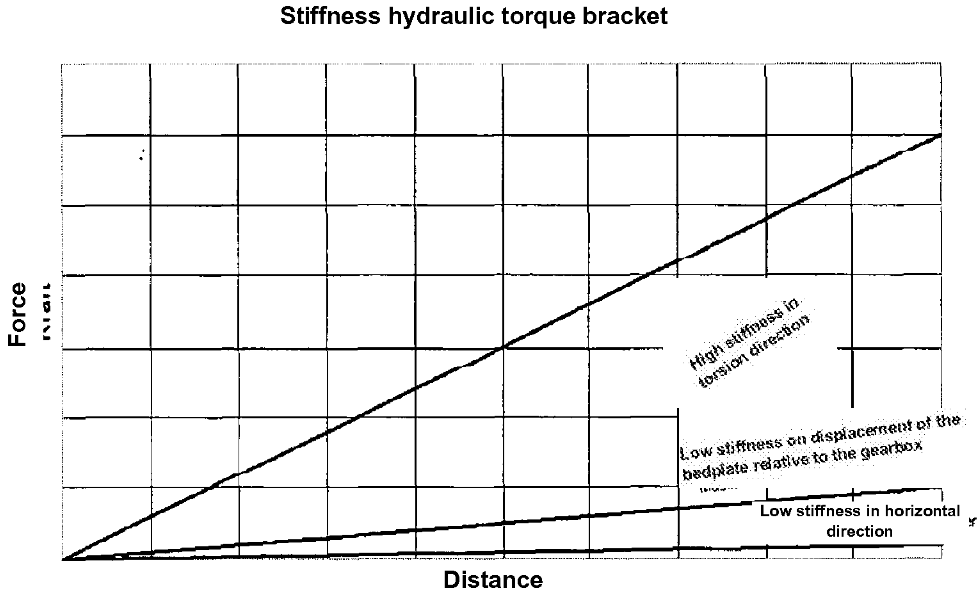

FIG. 6

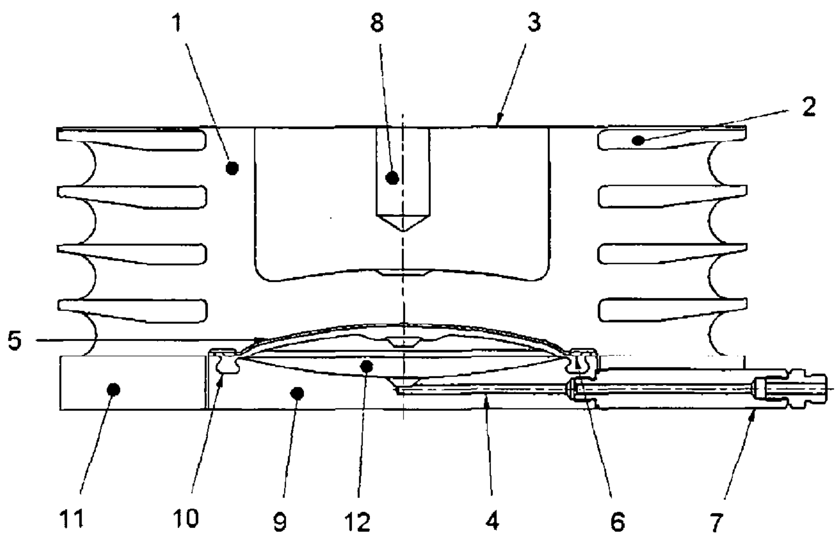

Fig. 7

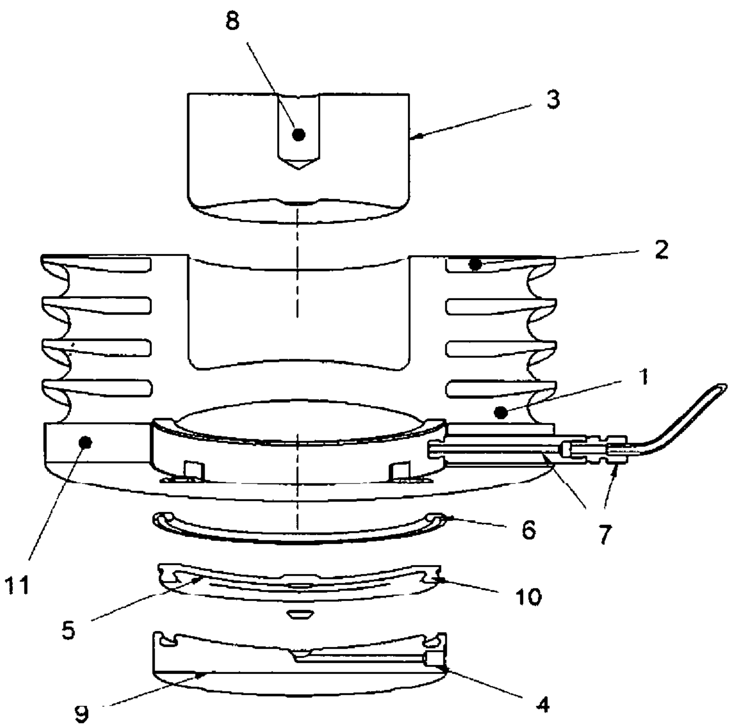

Fig. 8

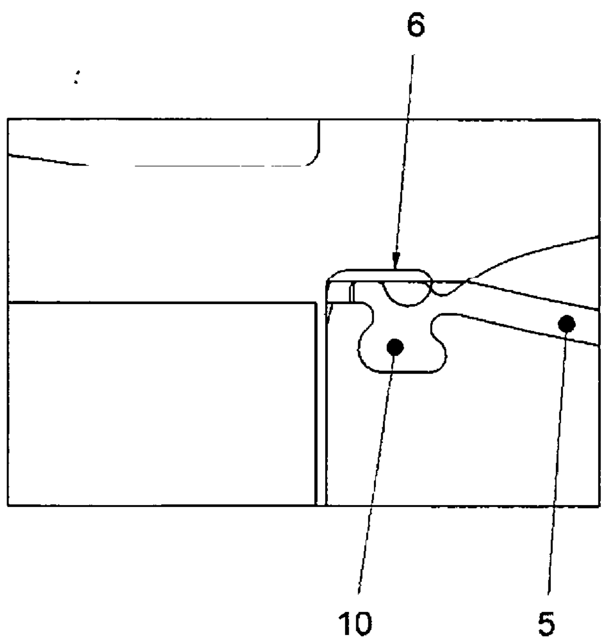

Fig. 9

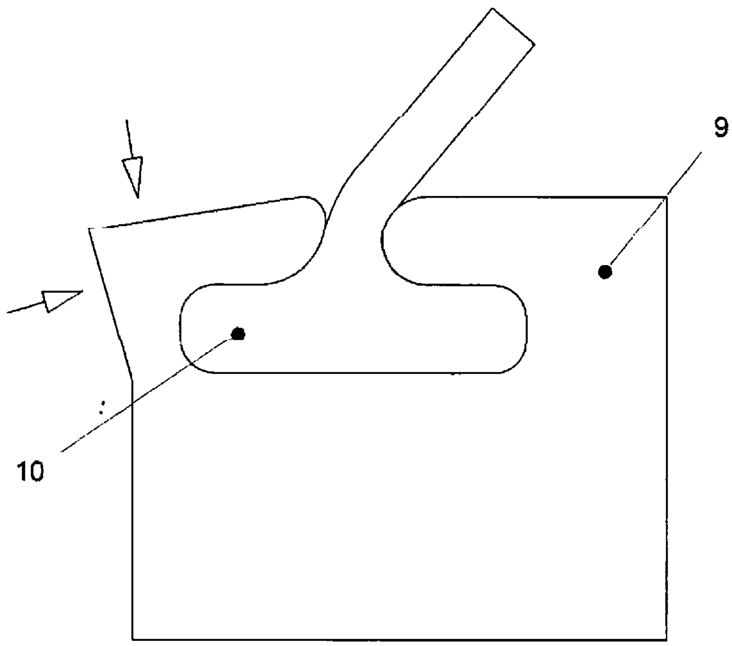

Fig. 10

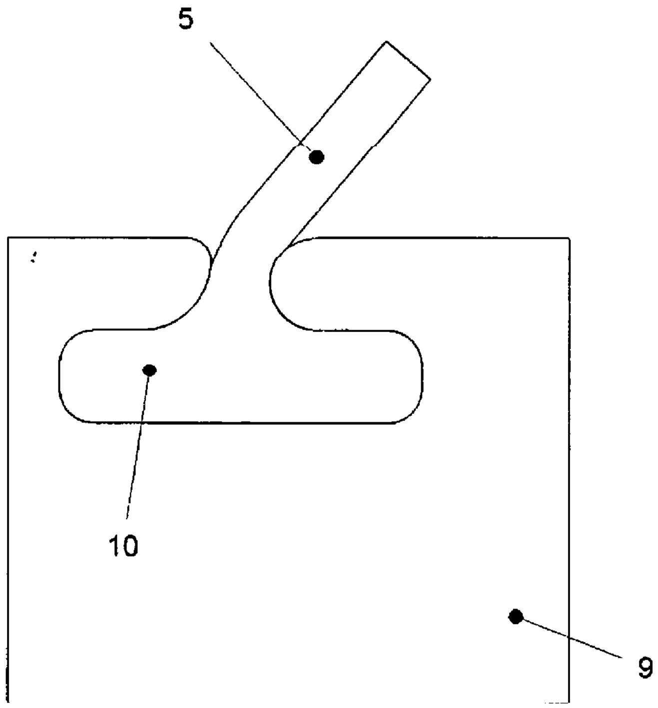

Fig. 11

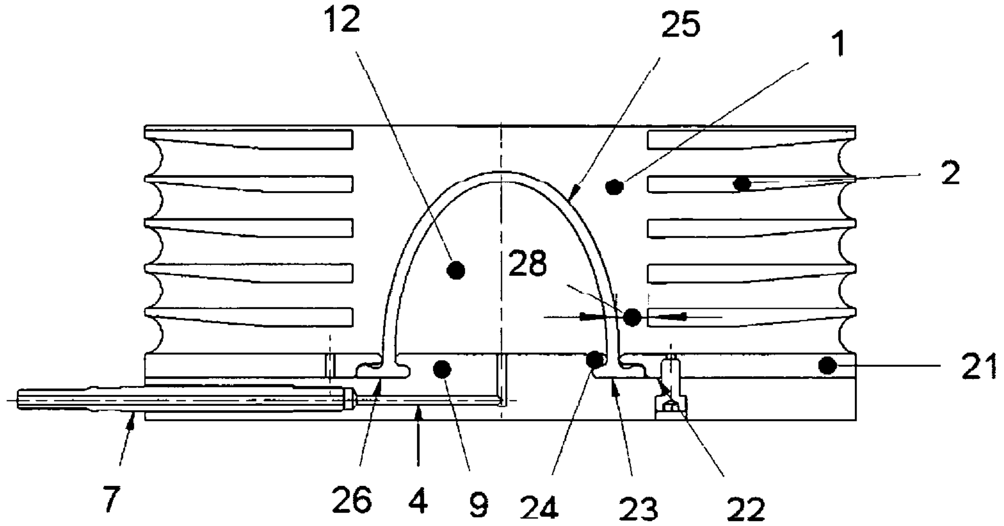

Fig. 12

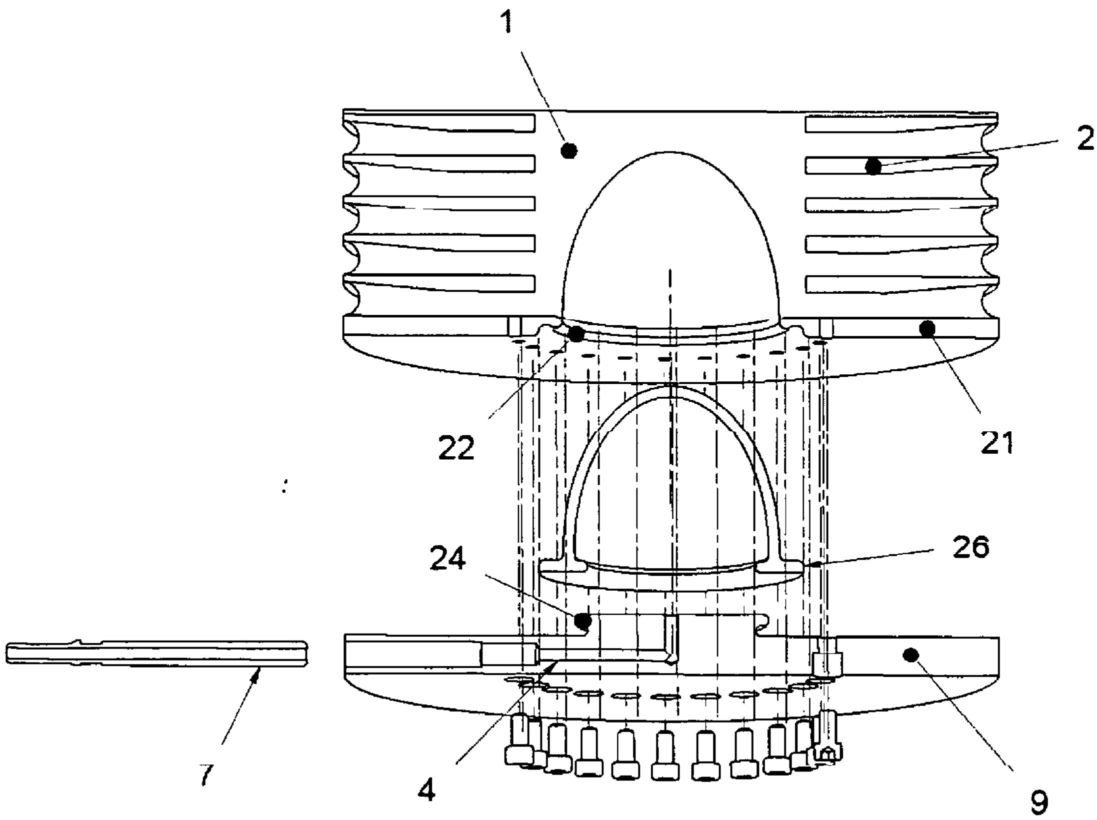

Fig. 13

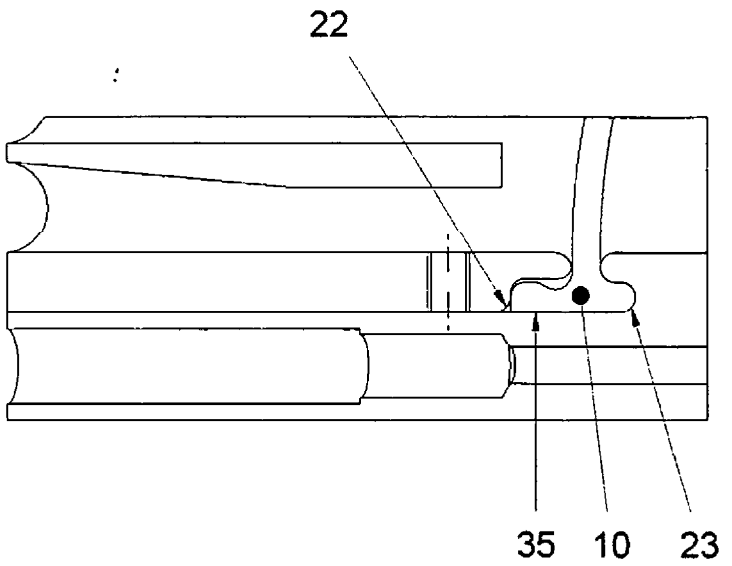

Fig. 14

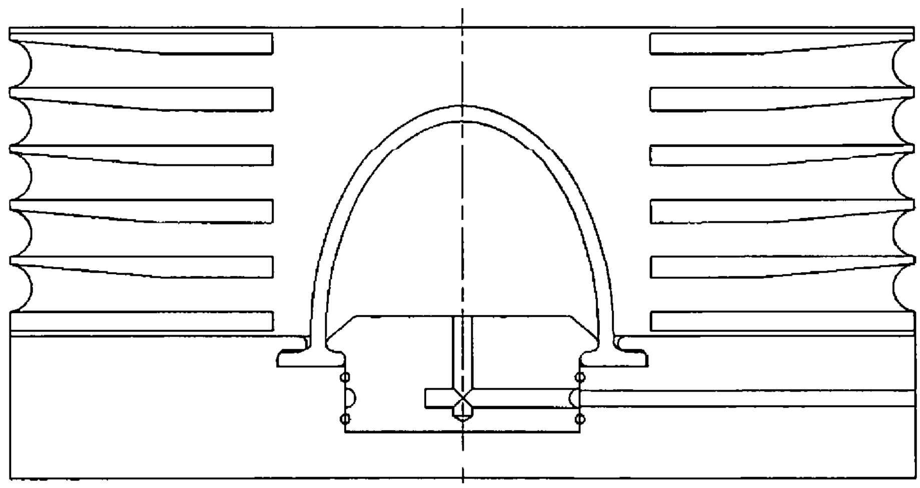

Fig. 15

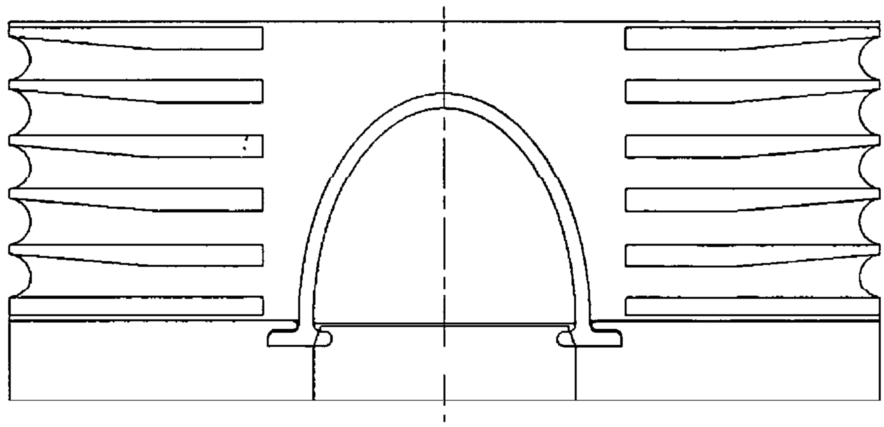

Fig. 16

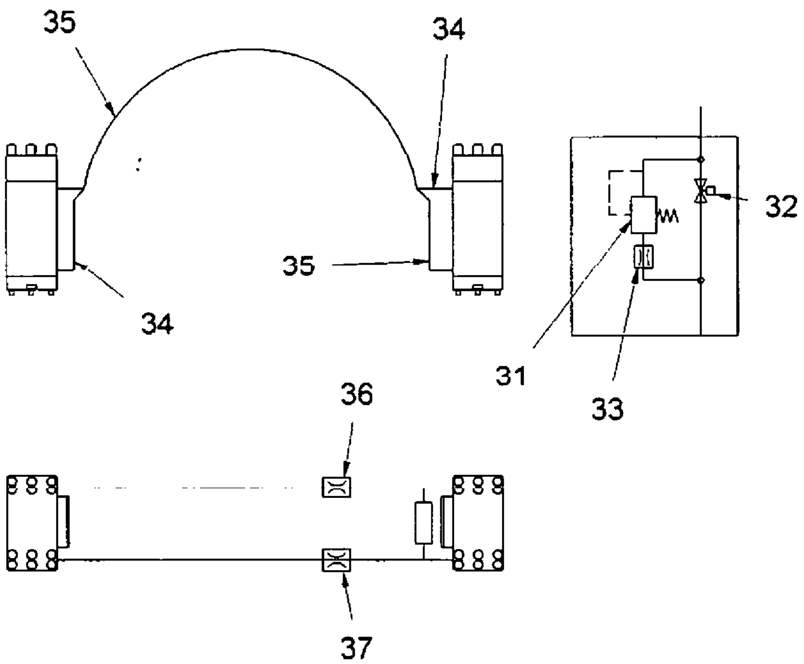

Fig. 17

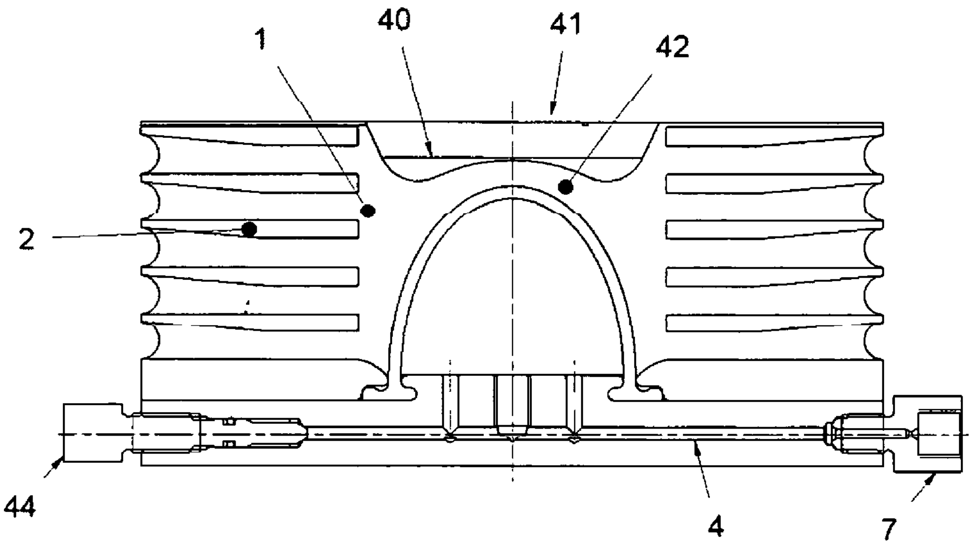

Fig. 18

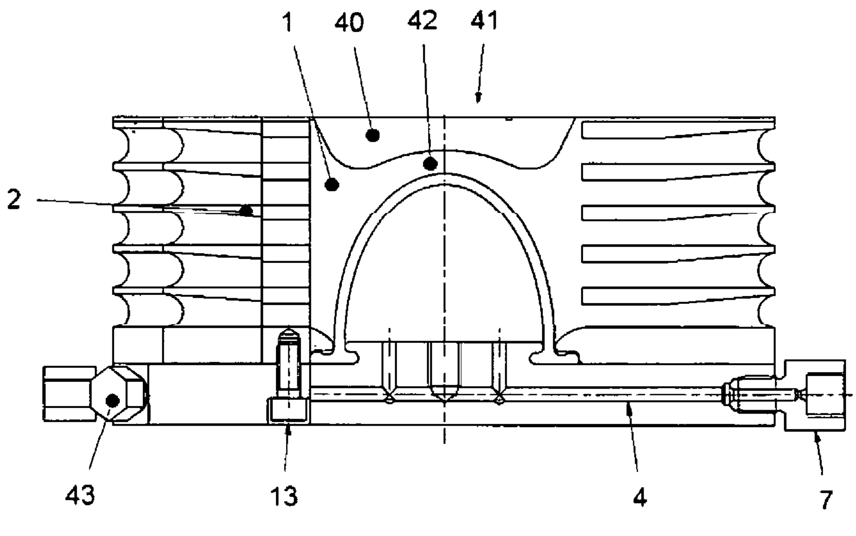

Fig. 19

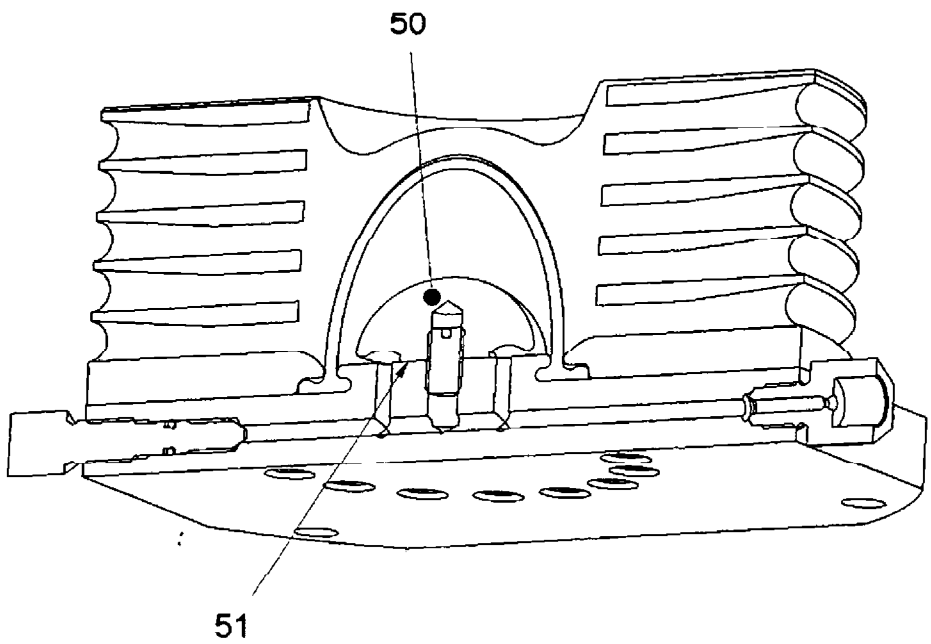

Fig. 20

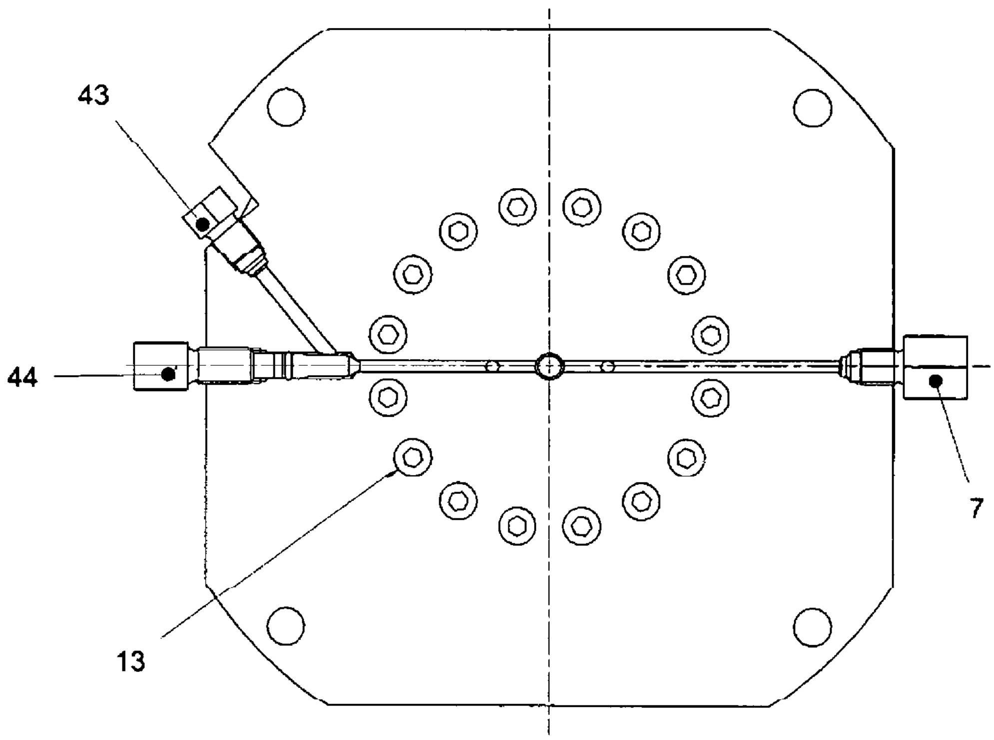

Fig. 21

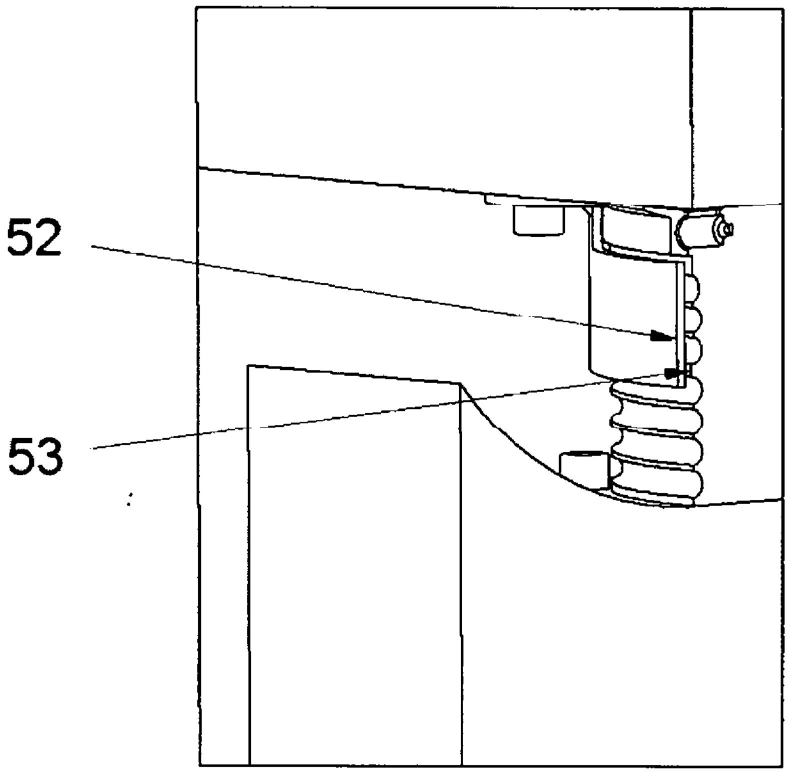

Fig. 22

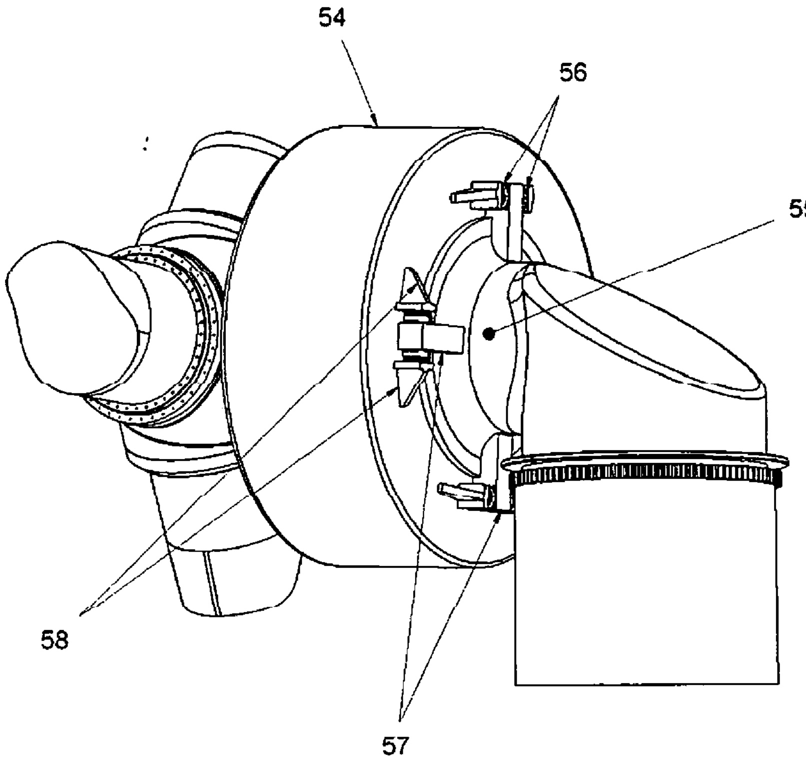

Fig. 23