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Abstract

A valve system comprising a valve body which influences the flow through a fluid connection of the valve system, wherein the valve body carries out a valve movement, whereby the distance between a valve surface and the valve body changes, while the valve body is in active connection with a damping device counteracting its valve movement.

AUSTRALIA

Patents Act 1990

Stabilus GmbH

ORIGINAL

COMPLETE SPECIFICATION
STANDARD PATENT

Invention Title:

Valve system

The following statement is a full description of this invention including the best method of performing it known to us:-

Title

A Valve

Technical Field

The invention concerns a valve for controlling flow through a fluid connection.

Background of the Invention

A basic problem of a valve system is the occurrence of flow or switching noises. In the case of damping valves smoothing channels have been introduced to prevent a sudden pressure difference between the flow-in and flow-out sides.

Furthermore, in the case of switchable damping valves it is known that the switching process will be carried out only when the pressure in the oscillation damper falls short of a maximum pressure, otherwise switching noises would be perceivable. In this conjunction experiments have been carried out to limit the operating travel of the valve body so that no switching noises could occur.

An advantage of at least one embodiment of the invention is to produce a valve system which possibly does not generate switching noises, has small internal friction and carries out the intended operating movement under all operating circumstances.

Summary of the Invention

The present invention provides a valve for controlling flow through a fluid connection, comprising a housing having a portion forming a damping chamber that is filled with a fluid and has a throttling outlet opening, a valve member received in the housing and movable relative to a valve seat in the housing, a displacer movably received in the damping chamber, operatively associated with the valve member and biased relative to the housing such as to tend to close the valve, and a resilient releasable catch arrangement acting between the housing and the displacer and having catch elements and countercatch elements which when engaged define at least two positions of the displacer relative to the housing.

The damping device prevents a too fast an opening movement of the valve, which may cause flow noises by the flow medium or even switching noises due to the weight of the valve body.

In a further advantageous refinement the damping device has a velocity-dependent effectiveness. Thus there is a direct relationship between the built-up pressure as the operating force and the counter-measures of the damping system, which also operates in a velocity-dependent manner. A complicated electronic system which may offer the same effect may be dispensed with.

For this purpose the velocity-dependent damping device has a displacer which can be moved in a damping chamber filled with the pressure medium. The same medium which flows through the valve system is used as pressure medium.

Furthermore, the damping device has at least one throttling cross-section. Several throttling cross-sections may be used, so that a stroke-dependent damping effect is possible. In a further advantageous refinement the damping chamber is formed from a cup-shaped housing into which the displacer protrudes.

The throttling cross-section of the damping device is relatively small. So that the gap between the displacer and the housing would not influence the damping effect, the displacer is sealed against the damping chamber.

According to a sub-claim the valve body is axially displaceable and during its operating movement is supported by a slide, while the slide forms a sub-assembly with the displacer.

With regard to a simple manufacturing the damping chamber has a stepped internal shape, while a longitudinal section of the damping chamber represents a movement path for the displacer. For this reason the entire internal wall of the damping chamber must be produced with great accuracy.

A further measure to influence the operating movement of the valve body is that the damping chamber accommodates a locking device which comprises the locking means and counter-locking means, while the locking means are actively connected to the valve body and define at least one operating position of the valve body.

Thus the locking means consist of radially displaceable support means for the valve body, whereby the locking means engage the counter-locking means.

As far as a simple assembly is concerned, the locking means form one sub-assembly with the slide. In practice the locking means are a sleeve-shaped extension of the slide, while the extension has axial slots enabling the radial mobility of the support means.

As far as a defined closing position of the valve body is concerned the slide is biased by a closing spring provided within the damping chamber.

To improve the guiding of the valve body, the valve body comprises a valve slide, which together with a housing of the valve system forms inside the fluid connection a chamber for the built-up pressure, while the chamber for the built-up pressure is sealed by a seal having a construction with limited radial movement. To minimise the problem of friction in conjunction with the operating movement of the valve body, the seal is placed in a groove wherein the seal can reach the base of the groove.. There is only a slight bias of the seal in the built-up pressure chamber. The actual sealing effect is produced by the bias of the built-up pressure.

To prevent a building up of the pressure when the valve slide retracts, the chamber for the built-up pressure has a pressure-equalising connection. For the purpose of saving construction space, the pressure-equalising connection is provided in the valve slide.

In addition, the pressure-equalising chamber has a further pressure-equalising connection which is controlled by a sealing unit which seals the housing of the valve system, while the housing separates two working chambers from each other and moves axially during the operation of the valve system, while the sealing unit has a first external sealing ring having a slight friction and an internal sealing ring pre-tensions radially the external one and depending from its position within an annular groove in the housing controls the further pressure-equalising connection.

The chamber for the built-up pressure has an additional pressure-equalising connection in the valve slide which connects the build-up chamber with a working chamber independently from other pressure-equalising connections. Accordingly, a maximum of three pressure-equalising connections is available, so that even in the case of a malfunction of one of the pressure-equalising connections a reliable valve slide movement is feasible.

Brief Description of the Drawings

The invention is explained in detail based on the following description of the figures. They show in:

Fig. 1 - an exemplary installed situation

Figs. 2a-2d - a constructive version of the invention, sectioned,

Fig. 3 - a damping piston in conjunction with the invention.

Detailed Description of the Drawings

Fig. 1 shows the outline of a motor vehicle 1 with a hatchback lid 3, which is provided in a movable manner about a pivot axis 5 which is transverse to the longitudinal axis of the vehicle. For the purpose of supporting the opening movement between the chassis and the hatchback lid, a piston/cylinder unit 7 is hinged movably via connecting members 9; 11. The piston/cylinder unit comprises a cylinder 13 and a piston rod 15 which can move in it in the axial direction, while one of these components is joined to the chassis of the vehicle and one to the hatchback lid, so that a movement of the hatchback lid is synchronous with the retraction and extension movements of the piston rod. The use of the piston/cylinder unit is not restricted to hatchback lids, but can also be used for other purposes, e.g. vehicle doors.

A first embodiment is illustrated in Figs.2a-2d, while the illustrations of Figs.2b-2d are limited to that part of the piston/cylinder unit 7 which has a piston 17 on the piston rod 15.

In the further description reference is made to Figs.2a and 2b. The piston 17 forms the housing for a valve system 19, which makes the continuously adjustable hydraulic blocking of the piston/cylinder unit 7 feasible, whereby a fluid connection 21 between the working chambers 23; 25 separated by the piston with the piston ring 17b can be arbitrarily controlled. For this purpose the valve system has a first closing valve 27, which has a valve body 29, to be called valve closing body 29 in the following, which is formed by a valve ring and is axially movably mounted in a valve sleeve 31 forming a portion of the piston. At the same time the valve closing body is situated between two mounting surfaces 33; 33′ of the valve slides 35; 35′. The valve slides are axially movably arranged on the piston ring 15 and are axially biased by a closing spring 39; 39′ by interposing a slide 37; 37′. The closing spring, in turn, rests on a bottom 41a; 41a′, which is part of the housing 41, which forms a damping chamber 55. The housing 41 has a stepped internal shape, forming a track 41b; 41b′ for a displacer 37a; 37a′. The displacer 37a; 37a′ is part of the slide 37; 37′. At least one throttling cross-section 41c; 41c′ is machined in the housing 41; 41′. Several throttling cross-sections may be provided which act depending on the stroke position of the displacer 37a; 37a′. As an alternative, in the case of an appropriate construction space, the throttling cross-section may be executed in the displacer. The displacer 37a; 37a′ is sealed relative to the damping chamber 55; 55′ by an annular seal 37b; 37b′, so that the damping effect could not be influenced by leaks.

Inside the damping chamber a locking device 129; 129′ is provided, comprising locking means 131; 131′ and counter-locking means 133; 133′. The locking means 129; 129′ are effectively connected with the first valve closing body 29 via the slides 39; 39′ and consists of radially movable support means which engage a locking shoulder of the counter-locking means. In this embodiment the locking means, the slide and the displacer are constructed as one sub-assembly. In addition to the inherent tensioning of the support means, an annular spring 135; 135′ ensures a radial biasing force between the supporting means and the respective locking shoulder, of which there are at least two. Alternatively, the locking spring 39; 39′ may be provided between the housing 41; 41′ and the supporting means provided construction space permits this.

The valve system 19 is constructed as a mirror image of the first valve system 29, so that a blocking function is feasible in both flow-through directions of the fluid connection 21. In order not to burden the figures with reference numerals, some reference numerals are shown only on one half of the mirror-imaged valve system.

Fig.2b shows the valve system in the blocked position. Both working chambers 23; 25 are under the same operating pressure. The closing valve body 29 of the first closing valve 27 is situated in the central portion of the valve sleeve 31, a sealing surface region 43. Starting from the sealing surface region a throttling device 45; 45′ joins the discharge side of the first closing valve 27. The throttling device comprises a number of grooves 45, 45′ having different lengths to expose a larger flow-through cross-section with the increasing adjusting travel of the valve closing body 29.

The valve system 19 comprises two closing valves 27; 47 and 27′; 47′ which are arranged in series, while the second closing valve 47; 47′ opens only when the first is already switched to open. The second closing valve 47; 47′ also has a valve closing body 49; 49′ which is constructed as a radial elastic seal. The elastic seal is located in a groove 51; 51' and can deform radially up to a groove base 53; 53' of the valve slide 35; 35' of the second single valve 47; 47'. Thus the second valve closing body 49; 49' has only a very weak bias, so that only a very weak frictional force emanates from the valve closing body 49; 49'.

A throttling device is connected downstream also to the second closing valve 47; 47' on the discharge side, which throttling device comprises slots 57; 57' in the tubular body 17a of the piston. This should prevent that the total built-up pressure, acting on the second closing valve, can flow unhindered through the second closing valve 47; 47' and will not generate noises.

For reasons of operating comfort a gradual valve release force should be used for the valve system 19. For this purpose areas of different sizes are provided to the pressurised in both closing valves 27; 47; 47'. In the case of the first closing valve 27 the pressurised area is to be equated that of the first valve closing body 29, to be called A_{ring} in the following. The second closing valve 47; 47' has a considerably larger pressurised area, resulting from the annulus between the valve slide 35; 35' and the valve slide body 49; 49', to be called $A_{valve\ slide}$ in the following. The $A_{valve\ slide}$ area is obviously greater, so that the forces effectively keeping open the second closing valve are many times greater than the actuating forces on the first closing valve, while the actuating forces have to be applied only momentarily followed by a comfortable operation.

The opening forces are determined not only by the closing spring 39; 39' but also in conjunction with the locking device 129; 129'.

A basic problem of valve slides is always present when they are pushed into and moved out from a hollow space of the fluid connection 21. A pressure build up or a vacuum should be prevented, which would obstruct the movement of the valve slide. For this reason the valve system 19 has a non-return valve 67; 67′ which switches on or off a pressure-equalising connection 69; 69′. The non-return valve is constructed from the piston ring 17b of the piston, which divides hydraulically the two working chambers 23; 25. The piston ring is guided inside a piston ring groove 71, while the piston ring groove has a switching path 73; 73′ into which the pressure-equalising connections 69; 69′ open. The piston ring 17b is to be regarded as a sealing unit, comprising an internal sealing and pre-tensioning ring 17ba and an external sealing and sliding ring 17bb. The external sliding ring 17bb is designed principally with a low frictional coefficient with the cylinder 13 and can be made from teflon, for example. In contrast, the internal pre-tensioning ring is made from an elastomer.

As a further measure of the "prevention of vacuum in the fluid connection 21,21′" function is carried out by a further pressure-equalising connection 87; 87′ which is executed in the valve slide 35; 35′ and has a non-return valve in the form of a rocker disc 89; 89′ which is open towards the fluid connection 21; 21′, which rocker disc is switched on by pressure between two limit stops on the valve slide.

In the idle position the first closing valve 27 and both second closing valves 47; 47′ as well as the non-return valves with the rocker disc 89; 89′ between the fluid connection 21; 21′ and the working chambers 23; 25 are closed. The first pressure-equalising connection 69; 69′, which is switched on by the piston ring, is open in the idle position.

When the piston rod moves in the direction of the arrow, a large portion of the medium flows from the working chamber 25 to the valve slide 35. The inflowing medium is deflected in the direction of the inside diameter of the slide 35 and is further conveyed by means of longitudinal channels 79 in the direction of the first closing valve 27. The face 33 of the slide 35 facing the closing body 29 of the first closing valve 27 is inclined, so that the closing body assumes a defined operating position, but on the other side an annular chamber 81 is present in which a pressure can build up which can displace axially the valve closing body 29 of the first closing valve 27.

To commence the opening movement of the first closing valve 27 and of the second closing valve 47 a situation arises which can be recognised from Fig.2c. A small portion of the medium flows into the gap 83′ between the tubular body 17a and the cylinder 13 up to the piston ring 17b. The switching path 71′ between the sealing unit and the sealing groove is closed. At that instant, when the piston ring 17b closes the pressure relieving connections 69′, the valve closing body 29 of the first closing valve 27 has not yet reached the grooves 45′ past the valve sealing region 43, a vacuum would build up in the fluid connection 21. This vacuum would be disadvantageous for the opening conditions of the second closing valve 47′. Accordingly, at this instant the non-return valve 87; 89′ opens the further pressure-equalising connection 89′ and lets the medium to flow in from the working chamber 23 into the fluid connection 21′. As soon as the first closing valve 27 permits a passing through of the medium, by virtue of the different sizes of the pressurised areas the built-up pressure in the fluid connection 21′closes again the non-return valve 87; 89′. At the same time it is insignificant for the operation of the left second closing valve 47′ whether the non-return valve on the right valve slide 35 is open or closed.

When the first closing valve 27 is being opened, the medium flowing into the fluid connection 21 through the grooves 45′ in the tubular body 17a can build up a second smaller pressure on the second closing valve 47′. No significant leakage loss will occur. On this occasion the second valve closing body 49′, which has a free access to the groove base 53′, is hydraulically biased in the axial direction against the left lateral wall of the groove and against the inside wall of the tubular body 17a. The built-up pressure, acting on the pressurised area $A_{Ventilschieber}$ moves the valve slide 35′ against the force of the closing spring 39′ and the locking device 129′ together with the valve closing body 49′ in the region of the slots 57′. Thus the second closing valve 47′ also opens, while a further reduction of pressure takes place through the slots 57′ constructed as a throttling device to prevent too great a pressure jumps and, consequently, noises.

The displacement travel of the slide 37′ and of the valve slide 35′ against the holding force of the locking device 129′ results in a movement of the displacer 37a′ of the slide 37′ in the damping chamber 55′. The medium situated in the damping chamber is displaced through the throttling cross-sections 41c′, resulting in a dynamic pressure force acting against the operating movement of the slide 37′ and consequently the first and second valve closing body 29; 49′. Depending on the magnitude of the built-up pressure a damping is produced which prevents a too fast an opening movement of the valve slide 35′ and consequently the generation of noise (Fig.2d).

When the piston rod 15 is no longer moved, the built-up pressure on the second closing valve 47′ also drops until the force of the closing spring 39′ is greater than the opening force of the built-up pressure and the holding force of the locking device 129′, while the supporting means of the locking means 131′ carry out a radial expansion movement and are moved to the first locking position. At the same time the non-return valve 87; 89′ of the second pressure relief connection 89′ is closed, on the other hand the first pressure relief connection 69′ with the non-return valve formed by the piston ring 17b is open, so that the valve slide 35′ can retract into the fluid connection 21 in a controlled manner until the idle position is assumed again. In addition, a third pressure-equalising connection 35a′ is acting, which is constructed radially in the valve slide, possibly in the vicinity of a stop 35b′ of the valve slide 35′. The third pressure-equalising connection connects, via the longitudinal channels 79, the fluid connection 21 with the damping chamber 55′ which, in turn, is connected with the working chamber 25 via at last one throttling cross-section 41c'. By virtue of this no unintentional pressure can build up in the fluid connection 21.

As both valve slides 35; 35' have identical constructions, after a movement of the piston rod a blocking in the opposite direction with the right second closing valve 47 can be exactly understood, as already described. One can, however, provide closing springs 39; 39' with different spring forces to match the forces of the built-up pressure to suit the requirements.

There are applications, where there is a great probability that the blocking function will not be used within a certain range. For this purpose the cylinder has at least one bypass groove 91 or an increased diameter, which allow a connection between the two working chambers 23; 25 independently from the operating position of the valve system 19.

The further description refers to Fig.2a. The entire piston 17 can be pre-assembled with its internal components as one sub-assembly, independently from the piston rod 13. The fastening of the piston is carried out by annular holding elements 93; 95, which are brought inside the piston rod 15 in the region of the beads 97; 99. The piston is aligned between the two beads in the piston rod. When the desired position of the piston is assumed, these two holding elements are pressed into the respective bead resulting in a form-locking joint. This determines the axial position of the piston.

It has to be taken into consideration that, for example, in the case of an accident the blocking function of the valve system cannot be disengaged for inexplicable reasons. For this reason the holding element 95 which assumes, for example, the support when the lid has to remain open, will tear off if a tearing off force is introduced which is greater than the reasonable actuating force. In this actual embodiment it is the holding element 95 between the piston 17 and a separating piston 101.

The separating piston 101 is biased by a helical spring 103. It would be, however, absolutely sensible to bias the space 105 between the separating piston and a bottom 107 of the piston/cylinder unit using pressurised gas, so that an operating pressure would act on one face of the piston rod, which displaces the piston rod in the direction of extension.

In addition to the valve system 19 the piston/cylinder unit 7 has a mechanical-hydraulic pressure stop 111, which has a stop sleeve 113 supported on a piston rod guide unit 117 via a pressure stop spring 115.

The stop sleeve has a continuous flange 119, into which a sealing ring 121 is inserted, which seals a gap between the stop sleeve 113 and the cylinder 13. The internal wall of the stop sleeve has a stepped design, while starting from the point of application of the pressure stop 111 a face 123 contacts a flange 125 of the holding element 93.

The diameter of the step from the inlet side of the stop sleeve relative to the diameter of the holding element 93 is such that no significant throttling takes place. The actual throttling is assumed by a damping opening 127 in the stop sleeve. This damping opening connects the rear side of the stop sleeve with the flow-in side in the piston 17.

With regard to the overall length of the piston/cylinder unit the illustration in Fig.2a is compressed. The pressure stop, of course, is not immediately next to the bypass groove 91. The distance between the bypass groove and the point of application of the pressure stop has to be adapted to suit the relevant application.

When the piston rod moves in such a manner that the working chamber 23 is reduced, the holding element 93 moves from a defined stroke position in the stepped internal wall with an almost constant speed. The medium, situated in the working chamber 23, as a rule a fluid, as it has already been described, can flow through the open valves 27; 47 of the piston. As soon as the face 125 of the holding element 93 abuts against the face 123, as a result of the damping opening 127 a damping force builds up, reducing the speed of the piston. The flange 125 and the shoulder surface form then a seal which is effective at least in the dynamic sense. Thus the damping effect is determined deliberately exclusively by the cross-section of the damping opening.

Fluid can further flow into the piston, since at the point of application of the pressure stop 111 a distance will still exist between the bottom 41a' and the stop sleeve 113.

With the reduction of speed of the movement of the piston the speed of flow of the fluid through the piston will be also reduced and, consequently, also the built-up pressure on the valves 27; 47; 47'. If a threshold value for the built-up pressures is not reached, the valves 27; 47; 47' move to the blocking position (cf. Fig.2b). The piston, the piston rod and consequently the vehicle's door remain forcibly standing, while the entire process will progress not abruptly, but continuously by the damping effect of the damping opening in such a manner that the introduction of the force into the chassis of the vehicle assumes a controllable level. The valves 27; 47; 47' situated in the blocking position prevent also a fast return of the vehicle's door. In this conjunction it should be mentioned that the spring force of the pressure stop spring 115 is so small that the pressure stop can move back to its initial position against the frictional forces between the seal 121 and the cylinder 13, but cannot exert any considerable force on the piston rod.

Fig.3 shows a piston 17 of an oscillation damper, like one which is used, for example, in a motor vehicle and is known from DE 34 29 473 A1. Parts having the same functions are designated by the same reference numerals as used in Figs.2a-2d.

When the piston rod moves in the direction of the working chamber 23, the damping medium moves into the fluid connection 21 and strikes a valve body in the execution of at least one valve disc. A small pre-opening cross-section 29a is effective when the flow velocities are small. When the velocities are greater, the valve body lifts off a valve seat 17c of the piston. Depending on the location of the displacer relative the housing 41 the displacer 37a protrudes into the damping chamber 55, consequently braking the working movement of the valve body 29. Although the valve body 29 can lift off the valve seat by the same distance, the lifting velocity will be limited. The spring force of the closing spring 39 may be reduced and the damping effect of the damping chamber considered when designing the piston valve.

THE CLAIMS DEFINING THE INVENTION ARE AS FOLLOWS:-

1. A valve for controlling flow through a fluid connection, comprising a housing having a portion forming a damping chamber that is filled with a fluid and has a throttling outlet opening, a valve member received in the housing and movable relative to a valve seat in the housing, a displacer movably received in the damping chamber, operatively associated with the valve member and biased relative to the housing such as to tend to close the valve, and a resilient releasable catch arrangement acting between the housing and the displacer and having catch elements and countercatch elements which when engaged define at least two positions of the displacer relative to the housing.

2. The valve according to claim 1 wherein the damping chamber is defined by a cup-shaped end portion of the housing.

3. The valve according to claim 1, wherein the displacer is sealed off from the damping chamber.

4. The valve according to claim 1, wherein the valve member is movable axially of the housing and is supported by a slide, the slide forming a structural unit with the displacer.

5. The valve according to claim 1, wherein the portion of the housing forming the damping chamber has a stepped inner contour, a surface of which forms a guide surface for the displacer.

6. The valve according to claim 1, wherein the catch elements include radially movable members that engage in the countercatch elements.

7. The valve according to claim 6, wherein the catch elements are radially deflectable fingers separated by slots and unitary with the displacer.

8. The valve according to claim 1, wherein the displacer is biased by a locking spring received within the damping chamber.

9. The valve according to claim 1, wherein the valve member includes a slide which forms with the housing a back-pressure region of the fluid connection, the back-

pressure region being sealed off by a valve seal that is supported by the slide and is radially movable relative to the valve seat in the housing.

10. The valve according to claim 9, wherein valve seal in the closed position is held in a groove in the valve seat, the seal being in spaced apart relation to a bottom surface of the groove.

11. The valve according to claim 9, wherein the back-pressure region has a pressure-equalizing connection which, upon movement of the slide into the back-pressure region, relieves the pressure in the fluid.

12. The valve according to claim 11 wherein the pressure-equalizing connection is arranged in the slide.

13. The valve according to claim 11, wherein the back-pressure region has an additional pressure-equalizing connection which is controlled by a sealing unit that seals off the housing of the valve, wherein the housing separates two working spaces from one another and in operation of the valve is moved axially, and wherein the sealing unit has a first outer sealing ring that is designed for low friction and an inner sealing ring that preloads the outer ring radially and, as a function of its position within an annular groove in the housing, controls the additional pressure-equalizing connection.

14. The valve according to claim 11, wherein the back-pressure region has an additional pressure-equalizing connection in the slide which, independently of other pressure-equalizing connections, connects the back-pressure region with a working space.

15. A valve substantially as described herein with reference to Figures 2a to 3.

DATED this fifteenth day of May 2003

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F.B. RICE & CO.

Fig.1

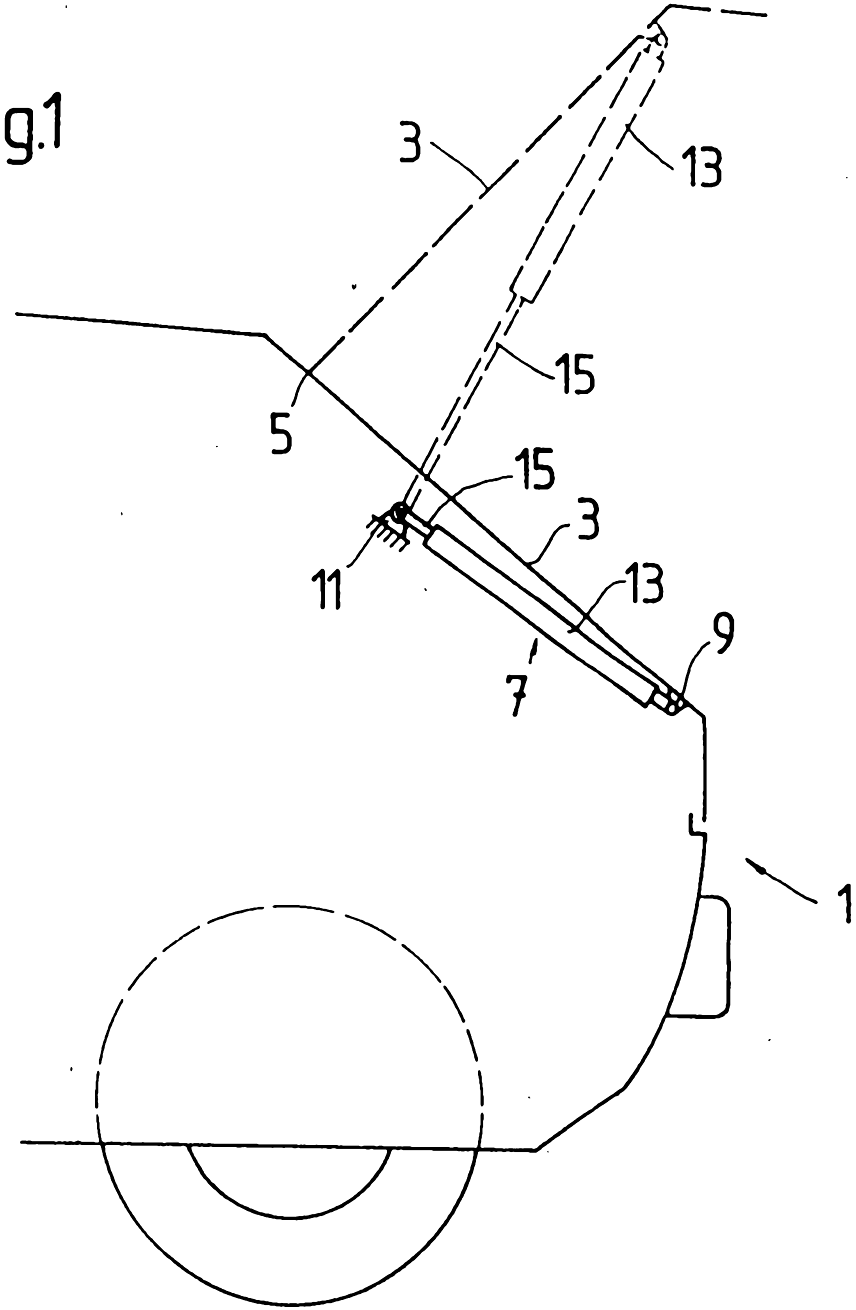

Fig. 2a

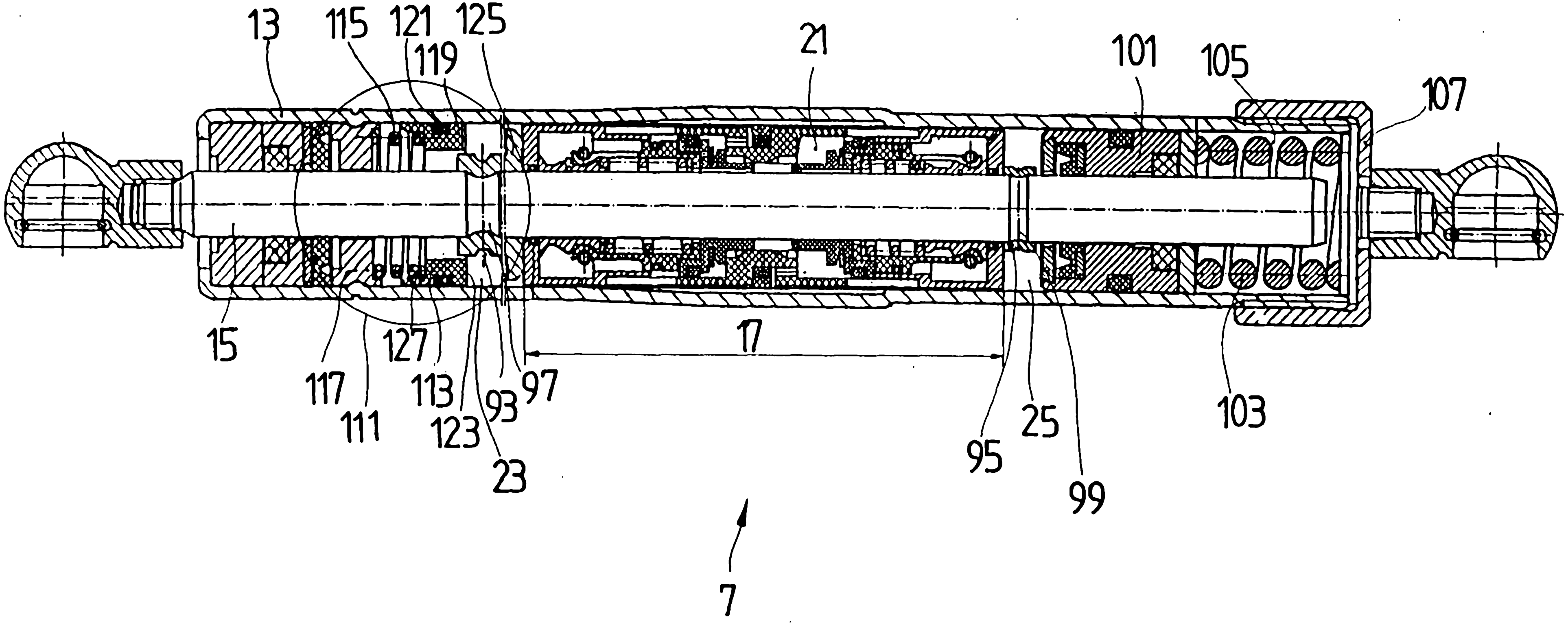
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Fig. 2b

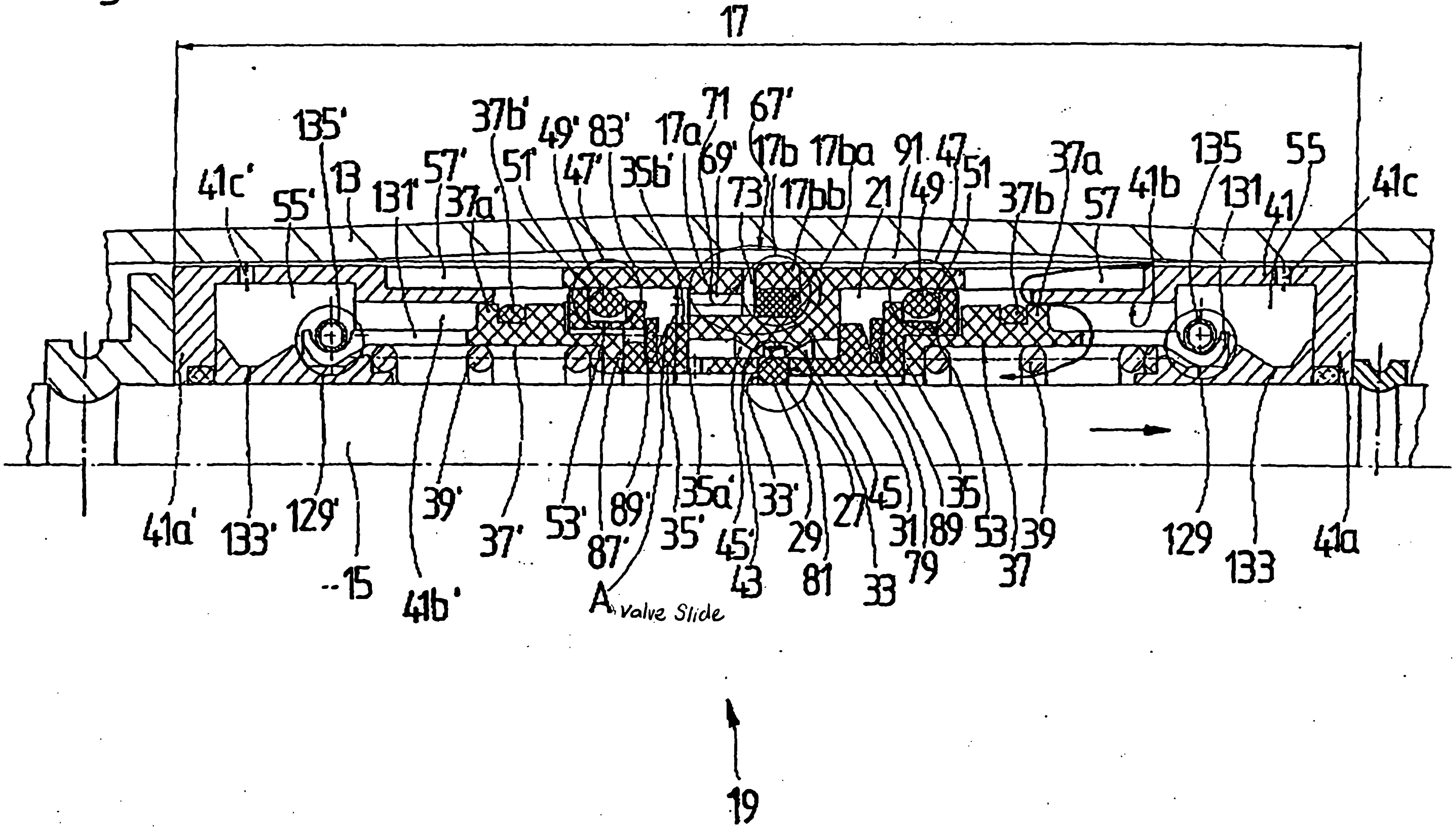

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Fig. 2c

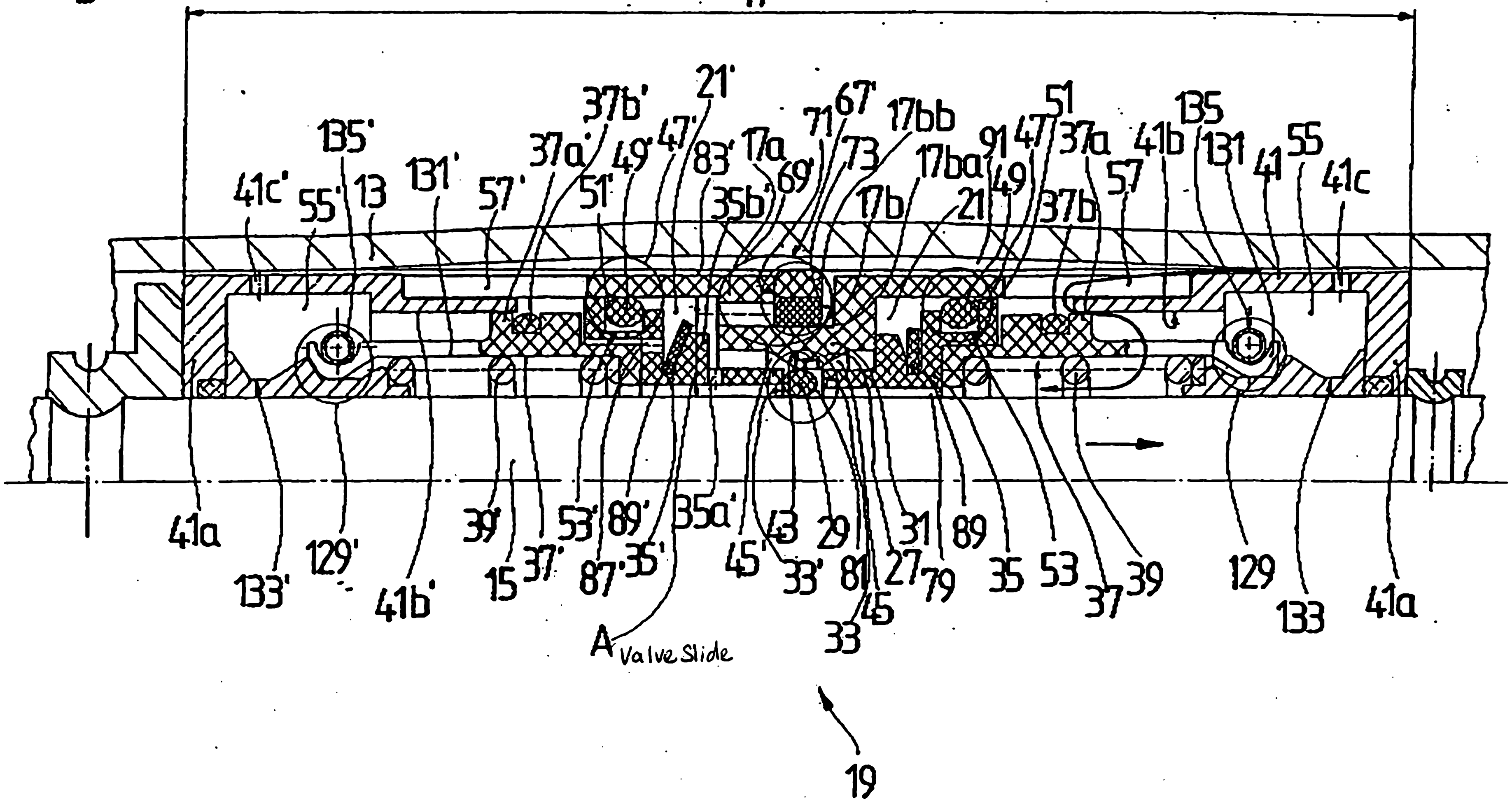

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Fig. 2d

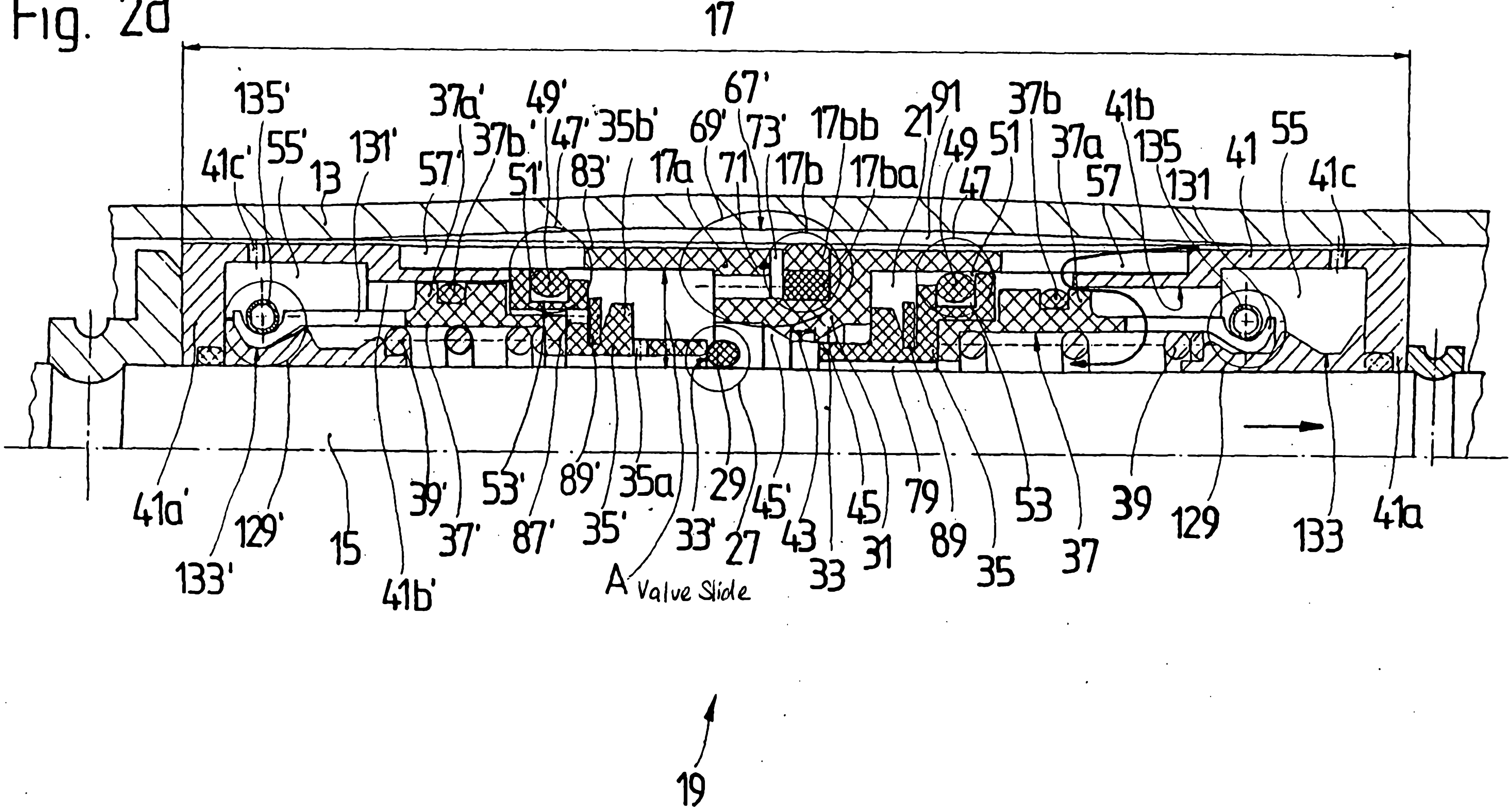

Fig. 3