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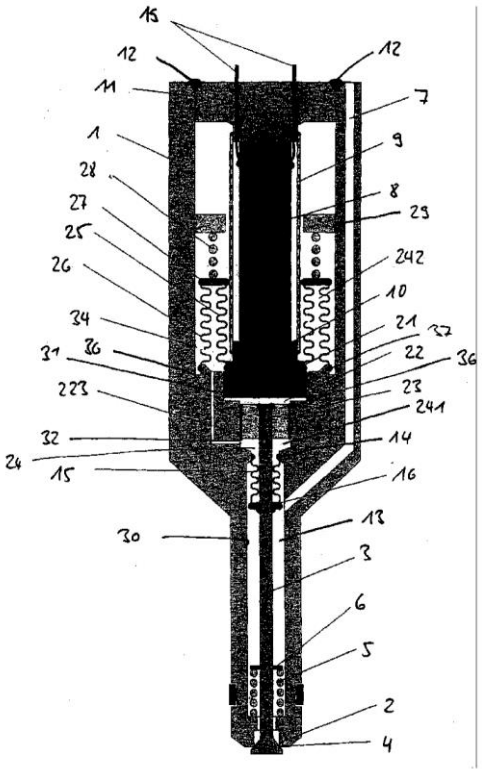
(54) Hydraulic device for transmitting an actuator movement

(57) A device for transmitting a movement of an actuator to a final control element has a first piston element, which is fixedly connected to the actuator, and a second piston element, which is fixedly connected to the final control element, wherein a hydraulic chamber is formed between the first piston element and the second piston element, and wherein a storage chamber, which is connected to a hydraulic chamber via a throttle gap, comprises a pressure-loaded storage chamber region, the region boundaries of which are designed to be elastic.



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Description

The invention relates to a hydraulic device for transmitting a movement of an actuator to a final control element according to the preamble of claim 1, in particular for use in a fluid dispenser. Such a device, also referred to below as a transmission element, is known from DE 197 08 304 A1.

Storage injection systems using very high injection pressures are increasingly being used in motor vehicle technology. In such injection systems, known, for example, under the designation "common rail systems", fuel is applied under high pressure to injection valves arranged in the cylinders of the internal combustion engine. The injection process in the cylinder is triggered by opening and closing the injection valves, wherein the injection valves are actuated via actuators which operate according to the electromagnetic principle and, in order to achieve high switching speeds, also according to the piezoelectric, electrostrictive or magnetostrictive principle. The actuators in the injection valves actuate a valve needle in the injection valve, optionally with the interposition of a servo valve.

The following requirements in particular are placed on a fuel injection valve suitable for series production:

The valve needle is to be arranged in the injection valve either unloaded or loaded with a pressure-dependent force. If the fuel pressure on the valve needle increases, it must be ensured that the valve needle is pressed ever more firmly onto the valve seat to ensure sufficient tightness as the fuel pressure increases.

Furthermore, the injection valve should be insensitive to thermal or pressure-induced longitudinal expansions. The functionality of the injection valve should also not be impaired by settling effects, which can be triggered, for example, by aging processes of the actuator. In order to prevent changes in length in the injection valve caused by thermal, pressure or settling effects, the valve needle and the other components in the injection valve are usually made from special steels which, however, are very expensive. Furthermore, it is also necessary when using such expensive special steels to provide a sufficient distance between the individual components in order to be able to absorb any longitudinal expansion between the components. However, this necessary safety distance of 3 μm to 5 μm is lost as a usable stroke of the actuator, which can lead to problems when opening the valve needle, especially when using a piezo actuator which produces only a small stroke.

To avoid having to provide a gap between the individual components in the injection valve, DE 197 08 304 A1 proposes a hydraulic transmission element which transmits the deflection of the actuator in the injection valve to a drive plunger of the servo valve or a guide shaft of the valve needle. In this case, the hydraulic transmission element is essentially cylindrical and has a hydraulic chamber delimited by a flexible membrane. The drive plunger of the servo valve or the guide shaft of the valve needle rests against the flexible membrane. A connecting bore with a throttling effect leads from the hydraulic chamber to a storage chamber which is provided inside the transmission element and is closed off by a prestressed spring plate. A rigid cover plate is arranged over the spring plate in the hydraulic chamber and rests against the actuator of the injection valve. The hydraulic chamber and the storage chamber are filled with a hydraulic medium.

When at rest, the pressure of the hydraulic medium prevailing in the storage chamber is transmitted to the hydraulic chamber via the connecting bore, so that the flexible membrane always rests against the drive plunger of the servo valve or against the guide shaft of the valve needle, even if the arrangement of the individual components in the fuel injection valve is displaced due to thermal effects or aging processes. When the actuator is actuated, the deflection of this actuator is transmitted via the transmission element essentially unchanged to the drive plunger of the servo valve or the guide shaft of the valve needle. In this case, the connecting bore between the hydraulic chamber and the storage chamber is designed in such a way that essentially no hydraulic medium can flow out of the hydraulic chamber into the storage chamber due to the actuation times being in the range of milliseconds.

However, the transmission element known from DE 197 08 304 A1 is characterized by a complicated structure. Furthermore, this known transmission element makes it difficult to ensure temperature compensation over the entire operating range of the fuel injection valve from approximately −40°C to +150°C. In this large temperature range, the volume of the hydraulic medium used in the transmission element can change by up to 20%. However, such a large volume fluctuation can only be handled with great difficulty by the structure chosen in DE 197 08 304 A1.

The object of the present invention is to provide a play-free hydraulic device for transferring a movement of an actuator to a final control element, which is characterized by great reliability under high continuous loads and strong temperature fluctuations.

This object is achieved by a device according to claim 1. Further advantageous embodiments of the invention are specified in the dependent claims.

The device according to the invention is characterized by a transmission element, which has a first piston element that is fixedly connected to an actuator, and a second piston element that is fixedly connected to a final control element, wherein a hydraulic chamber is formed between the first piston element and the second piston element, and wherein a storage chamber, which is connected to the hydraulic chamber via a throttle gap, comprises a pressure-loaded region whose region boundaries are designed to be elastic. This structure reliably ensures automatic compensation of large status changes between the actuator and the final control element, which can be caused by thermal pressure or settling effects. In addition, the elastic design of a storage chamber region ensures that the transmission element remains functional over a wide temperature range, in particular over the entire operating range of a fuel injection valve from approximately −40°C to +150°C. In addition, the transmission element according to the invention can be used both in an inwardly and outwardly opening fuel injection valve. Furthermore, the transmission element is characterized by a very compact design, a very high hydromechanical transmission efficiency and excellent dynamic transmission properties, since only a very small hydraulic chamber is required between the first and the second piston element.

According to a preferred embodiment, the elastic storage chamber region is delimited by a spring bellows arrangement, preferably consisting of metal bellows. Such metal bellows are designed to be very stiff radially but very soft in the axial direction and can thus reliably absorb volume changes in the hydraulic fluid contained in the transmission element.

According to a further preferred embodiment, the actuator is prestressed by a spring element which is fixedly connected to the first piston element of the transmission element. By means of this embodiment, a return spring, which returns the final control element to its initial position after the end of the actuation of the actuator, can be dimensioned to be small, since the function of the return spring is supported by the retraction movement of the actuator, which acts on the final control element via the transmission element.

According to a further preferred embodiment, by suitably selecting the size ratios of the pressure-effective surfaces of the first piston element and of the second piston element, a stroke translation of the actuator movement to the actuator can take place. This ensures that even when using a piezo element as an actuator, a sufficient stroke to actuate the final

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control element is generated.

The invention is explained in more detail below with reference to the drawings.

FIG. 1 shows an outwardly opening fuel injection valve with a hydraulic transmission element according to the invention in a first embodiment;

FIG. 2 shows an inwardly opening fuel injection valve with a hydraulic transmission element according to the invention in the first embodiment; and

FIG. 3 shows an outwardly opening fuel injection valve with a hydraulic transmission element according to the invention in a second embodiment.

The fuel injection valve shown in **FIG.** 1, which opens outward into a combustion space of an internal combustion engine, is operated with fuel under high pressure. In this injection valve, a drive unit is installed in the upper part of a housing **1**, which drive unit has a piezoelectric multilayer actuator **8** in low-voltage technology as an essential component. This piezoelectric multilayer actuator **8** is surrounded by a tube spring **9** which is welded between a head plate **10** and a foot plate **11**, the tube spring **9** being prestressed in such a way that the piezoelectric multilayer actuator **8** is under a mechanical pressure prestress. The housing **1** is furthermore connected to a base plate **11** of the drive unit as rigidly as possible, preferably via a weld **12**.

The piezoelectric multilayer actuator **8** acts on the rear end of a valve needle **3** via a hydraulic transmission element when it is driven electrically via its supply lines **19**. The valve needle **3** is arranged in the front part of the housing **1** of the injection valve in a continuous inner bore **30** and, in the rest state, closes a valve seat **2** on the housing **1** with a valve disk **4** arranged at the front end of the valve needle **3**. The closed initial state in the injection valve is ensured in this case by a prestressed nozzle spring **5** which is connected to the valve needle **3** via a snap ring **6** and presses the valve disk **4** onto the valve seat **2**. When the piezoelectric multilayer actuator **8** is actuated, the deflection transmitted by the transmission element to the rear end of the valve needle **3** lifts the valve disk **4** from the valve seat **2**, so that fuel, which is fed into a fuel chamber **13** in the housing **1** via a fuel feed line **7**, can be injected past the valve needle **3** into the combustion space of the internal combustion engine.

Since the fuel is under very high pressure in the fuel chamber **13** in the housing **1**, this region must be reliably sealed off from the other regions in the housing **1** of the injection valve, in particular from the drive region. A metal bellows **15** is used for the hermetically sealed and axially very soft passage of the valve needle **3** from the fuel chamber **13** into the region in which the transmission element and the drive unit are installed. Adjacent to the fuel chamber **13** is an annular step **14** which projects into the inner bore **30**. Furthermore, an annular connecting piece **16** is attached to the valve needle **3**. Between the connecting piece **16** on the valve needle **3** and the annular encircling step **14** in the housing **1**, the metal bellows **15** is welded in, extending parallel to the valve needle **3** and serving to hermetically seal the fuel chamber **13** from the other housing regions in which the drive unit and the transmission element are located. Furthermore, the valve needle **3** is secured against rotation by the metal bellows **15**. This can be advantageous in particular when a stroke-limiting stop for the valve needle **3** is built into the fuel injection valve.

The use of the metal bellows **15** for needle feed-through enables the high-pressure region in the injection valve to be sealed off perfectly, permanently and reliably from the other regions. As calculations and tests have shown, the metal bellows **15** withstands very high pressures despite low wall thicknesses of, for example, **50** um to **500** um due to its high radial rigidity, without being irreversibly deformed. The metal bellows **15** can furthermore be designed in such a way that a high axial flexibility, i.e. low spring rate in the direction of movement of the valve needle **3**, is achieved by a sufficient number of shafts in order not to impair the deflection of the valve needle **3** and in order to keep the forces introduced into the valve needle **3** as low as possible by temperature-induced changes in the length of the needle feedthrough. Furthermore, the use of the metal bellows **15** in the needle feedthrough can prevent fuel leakage with high reliability.

The needle feedthrough comprising the annular step **14**, the metal bellows **15** and the connecting piece **16** can furthermore be designed in such a way that the pressure-induced forces acting on the valve needle **3** compensate one another, so that the valve needle **3** as a whole is kept free of force. This makes it possible to design the injection valve in such a way that a switching behavior that is virtually independent of the fuel pressure becomes possible, since the opening and closing forces are then determined solely by the piezoelectric multilayer actuator **8** and the force of the prestressed nozzle spring **5**.

On the other hand, the valve needle feedthrough formed by the annular step **14**, the metal bellows **15** and the connecting piece **16** can also be designed in such a way that a force results which increases with increasing fuel pressure in the fuel chamber **13** and with which the valve disk **4** is pressed into the valve seat **2**. By selecting the hydraulic diameter, which is defined by the step **14**, the metal bellows **15** and the connecting piece **16**, it is thus possible to hold the valve needle **3** of the injector in the desired manner in terms of compressive force, i.e. completely free of compressive force, overcompensated or undercompensated.

Due to its metallic material, the metal bellows **15** furthermore has a wide operating temperature range with constant functionality. Due to the low axial spring constant of the metal bellows **15**, the thermal changes in length of the metal bellows **15** themselves lead only to a negligibly small introduction of force into the valve needle **3** in the axial direction. The metal bellows **15** can also partially or completely replace the nozzle spring **5** due to its mechanical spring action in the axial direction.

In order to transmit the stroke of the piezoelectric multilayer actuator **8** to the valve needle **3**, the transmission element is provided between the drive unit and the valve needle **3**. This transmission element serves primarily as a hydraulic play compensation element in order to exclude any play between the piezoelectric multilayer actuator **8** and the valve needle **3**. Furthermore, a stroke translation can be effected with the transmission element.

The transmission element has a primary piston **21** and a secondary piston **23**, which are arranged in a bore portion adjoining the annular step **14** in the housing **1**. This bore portion is designed in two stages, with a first, wider bore portion **31**, which adjoins the drive unit and in which the primary piston **21** is seated, and a second narrower bore portion **32**, which adjoins the stop **14** in the housing **1** and in which the secondary piston **23** is arranged. The primary piston **21** is of essentially cylindrical design and is attached to the head plate **10** of the drive unit or is preferably fixedly connected thereto by welding. The head plate **10** and the primary piston preferably consist of one part. The secondary piston **23** is designed as a hollow cylinder and is plugged onto the rear end of the valve needle **3**, the end face of the secondary piston **23** facing the primary piston **21** being arranged essentially flat with respect to the end face of the valve needle **3**. The valve needle **3** and the secondary piston **23** are likewise preferably fixedly connected to one another via a weld, but at least without play and mechanically as stiffly as possible.

The primary piston **21** and the secondary piston **23** are furthermore spaced apart from one another in such a way that

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a hydraulic chamber **22** is formed between the opposing end faces in the region of the transition from the first bore portion **31** to the second bore portion **32**. Furthermore, a two-part storage chamber **24** is provided in the transmission element, which has a first storage chamber region **241** in the inner bore **30**, which is delimited by the lower end face of the secondary piston **23** and by the connecting piece **16** of the metal bellows **15** on the valve needle **3**. This first storage chamber region **241** is connected via an unthrottled connecting bore **223** formed in the housing **1** to a second storage chamber region **242** which is arranged around the drive unit in the housing region **34** adjoining the first bore portion **31**. The second storage chamber region **242** is delimited by two spring bellows **25**, **26** arranged concentrically with respect to one another and a pressure ring **27**, which in turn is held by a pressure spring **28**, which is supported on a perforated plate **29** in the housing region **34**. The inner spring bellows **25** is welded in between the inner side of the pressure ring **27** and the rear end face of the primary piston **21**, which protrudes from the first bore portion **31**. The outer spring bellows **26** is welded to the outer side of the pressure ring **27** and to a housing step **30** adjoining the first bore portion **31**. The connecting bore **223** opens into the housing step **30** between the two spring bellows **25**, **26**.

The hydraulic chamber **22** and the storage chamber **24** are connected to one another via a throttle gap **36**which is formed between the circumferential wall of the secondary piston **23** and the inner wall of the second bore portion **32**, and via a throttle gap **37**which is formed between the circumferential wall of the primary piston **21** and the inner wall of the first bore portion **31**. Furthermore, the entire interior of the transmission element is filled with a hydraulic fluid which is under a slight overpressure generated by the compression spring **28** which acts on the second storage chamber region **242** via the compression ring **27**. Before the interior of the transmission element is filled with hydraulic fluid, this hydraulic fluid is degassed in order to dissolve any gas bubbles present in the fluid.

The injection valve with the transmission element operates as follows:

To initiate the injection process, the piezoelectric multilayer actuator **8** is charged via the electrical supply lines **19**. This causes the piezoelectric multilayer actuator **8** to deflect axially and push the primary piston **21** downward into the first bore portion **31** via the head plate **10**.

In the case of the transmission element, the throttle gap **36** on the secondary piston **23** and the throttle gap **37** on the primary piston **21**, which produce a connection between the hydraulic chamber **22** and the storage chamber **24**, are dimensioned in such a way that only a negligible exchange of hydraulic fluid between the hydraulic chamber **22** and the storage chamber **24** can take place during the typical actuation times of the piezoelectric multilayer actuator **8** of 1 to 5 ms. This means that the volume of the hydraulic fluid during the injection time is determined only by the compressibility of the hydraulic fluid, and the hydraulic chamber **22** can thus be regarded as a rigid piston. The spring constant c of the hydraulic chamber **22** can be estimated as follows.

$$\kappa = -1/V \cdot \delta \text{ and } V = A \cdot H \Longrightarrow dP = -1/(\kappa V) \cdot A\, dh$$

κ: Compressibility of the hydraulic fluid
V: Volume of the hydraulic chamber
P: Pressure
A: Cross-sectional area of the hydraulic chamber
h: Height of the hydraulic chamber

$$\text{with } dF = dP/A \Rightarrow dF = -1/(\kappa V)\, A^2 dh$$

F: Force
giving us

$$c = -dF/dh \Longrightarrow c = A/(\kappa h)$$

The above equation shows that the greater the spring constant c of the hydraulic chamber **22**, the smaller its height and the greater its effective cross-sectional area. Simulation calculations have also shown that thermal and pressure-induced expansions of a maximum of **50** um are to be expected. In order to be able to regard the hydraulic chamber **22** as a rigid piston, the spring constant c of the hydraulic chamber **22** should be in the range of 10^8 N/m or higher. This means that, assuming that the compressibility of the hydraulic fluid κ is approximately $10 \cdot 10^{-10}$ m^2/N, which corresponds to a typical value for a hydraulic fluid, the desired value for the spring constant c, for example with a cross-sectional area of **1** cm^2 and a height of 0.1 cm, can be achieved. However, the exact design of the height and cross-sectional area of the hydraulic chamber **22** can in each case be adapted to the conditions in the injection valve in order to achieve a compact design.

As a result of the design of the hydraulic chamber **22** as a rigid piston, the movement of the primary piston **21**, which is triggered by the piezoelectric multilayer actuator **8**, is transmitted directly to the secondary piston **23** with little loss. The movement of the secondary piston **23** is only slightly damped by the hydraulic fluid located in the first storage chamber region **241**, since the excess hydraulic fluid is forced away into the second storage chamber region **242** via the unthrottled connecting bore **223** due to the rapid pressure increase in the first storage chamber region **241**. The two spring bellows **25**, **26**, which are arranged concentrically with respect to one another and delimit the second storage chamber region **242**, are designed to be very stiff radially but very soft in the axial direction. Preferably, metal bellows are used as spring bellows, which essentially correspond to the metal bellows **15** of the needle feedthrough. At this point, however, it is also possible to use elastomeric materials for the spring bellows **25**, **26**. When hydraulic fluid is pressed from the first storage chamber region **241** via the connecting bore **223** into the second storage chamber region **242**, the second storage chamber region **242** expands axially in the direction of the base plate **11** of the drive unit against the holding force of the compression spring **28** loading on the pressure ring **27**.

The movement of the secondary piston **23** triggered by the primary piston **21** displaces the valve needle **3** connected to the secondary piston **23** downward against the restoring force of the nozzle spring **5**, so that the valve disk **4** lifts off from the valve seat **2** and the injection valve opens. The elongation of the piezoelectric multilayer actuator **8** is thereby translated into a displacement of the secondary piston **23** and thus of the valve needle **3**, which corresponds to the ratio of the pressure-effective surfaces of the primary piston **21** and of the secondary piston **23** in the hydraulic chamber **22**. By suitably matching the primary piston surface to the secondary piston surface, it is thus possible, for example, to set an increase in the stroke of the piezoelectric multilayer actuator **8** with respect to the stroke of the valve needle **3**. This reliably ensures that the extremely short stroke of the piezoelectric multilayer actuator **8** is sufficient to open the valve needle **3** under all operating conditions of the injection valve.

The injection process is terminated by discharging the piezoelectric multilayer actuator **8** again via the electrical supply lines **19**. As a result, the piezoelectric multilayer actuator **8** is shortened to its initial length, with the tubular spring **9** preventing the piezo-ceramic from coming under tensile stress due to mass inertia effects when contracting.

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Since the primary piston **21** is fixedly connected to the drive unit via the head plate **10**, the primary piston **21** is also retracted from the first bore portion **31** by the contraction of the piezoelectric multilayer actuator **8**. As a result, a pressure drop is briefly produced in the hydraulic chamber **22**, which pressure drop cannot be immediately compensated by subsequent flow of hydraulic fluid from the storage chamber **24** via the throttle gaps **36**, **37** due to the extremely short switching times of the piezoelectric multilayer actuator **8** and the small throttle gap **36**on the secondary piston **23**. This pressure drop in the hydraulic chamber **22** with respect to the pressure prevailing in the storage chamber **24** leads to a pressure difference which drops across the side of the secondary piston **23** facing the hydraulic chamber and the side of the secondary piston **23** facing the first accumulator chamber region **241**. As a result, the resetting of the valve needle **3** is supported by the nozzle spring **5**, so that a rapid closing of the injection valve is achieved, which has a favorable effect on the course of combustion.

The design of the transmission element according to the invention also makes it possible to automatically compensate for all thermal length changes in the injection valve caused by settling effects of the drive unit or pressure-related changes. If, for example, the valve needle **3** is extended due to thermal or pressure-induced expansion in relation to the housing **1** of the injection valve, the secondary piston **23** is drawn upward into the second bore portion **32**. The throttle gap **36** on the secondary piston **23** is furthermore designed such that hydraulic fluid can be exchanged between the storage chamber **24**and the hydraulic chamber **22** via the throttle gaps **36**, **37** during the thermal processes which are in the range of a few seconds to minutes in time. Thus, when the secondary piston **23** advances into the second bore portion **32** due to the thermal processes, hydraulic fluid flows out of the hydraulic chamber **22** via the throttle gaps **36**, **37** into the accumulator chamber **24** until a pressure equilibrium is again established in the hydraulic chamber **22** and in the storage chamber **24**. The length compensation is limited exclusively by the height of the hydraulic chamber **22**.

For the transmission element to function reliably, it is particularly necessary for the hydraulic fluid to be hermetically sealed with respect to the fuel chamber or the drive unit. This results in high demands on the spring bellows **25**, **26**, which are therefore preferably designed as metal bellows. These metal bellows are corrugated, since this allows a very small axial spring constant to be achieved. The axial deformation of the metal bellows due to a pressure load is by no means small, but, like the forces acting on the individual bellows corrugations, is almost canceled out in total over the entire length of the metal bellows. A particularly favorable shape for the bellows shafts has been found to be a geometry which, viewed in longitudinal section, consists of semicircular segments which are joined to one another and have straight intermediate pieces. Compared to a sinusoidal waveform, the walls consisting of semicircular segments have lower mechanical stresses in the axial direction with higher axial flexibility:

As shown, the hydraulic fluid in the hydraulic chamber **22** and the storage chamber **24** is under a slight overpressure generated by the compression spring **28** which acts on the pressure ring **27** of the second storage chamber region **242**. This low overpressure combined with bubble-free filling of the hydraulic fluid into the transmission element ensures that the rapid switching processes of the piezoelectric multilayer actuator **8** do not lead to cavitations in the hydraulic fluid. Alternatively, the compression spring **28** can also be partially or completely replaced by a spring action of the spring bellows **25**, **26**. Furthermore, it is possible to provide the throttle gaps **36**, **37** in each case only on the secondary piston **23** or on the primary piston **21**, so that hydraulic fluid is

exchanged between one of the storage chamber regions **241**, **242** and the hydraulic chamber **22**. However, throttle gaps can also be provided on the primary piston **21** and on the secondary piston **23**.

FIG. 1 shows an outwardly opening injection valve. However, it is also possible to use the transmission element according to the invention with an inwardly opening injection valve. As is shown in **FIG.** 2, the valve needle **3** is then placed under compressive stress by the nozzle spring **5** instead of under tensile stress, so that the valve needle **3**, in the rest state, is seated with a conical needle tip **104** on a conical valve seat **102** in the injection valve, below which a fuel injection hole **103** for injecting fuel into the internal combustion engine is formed.

The injection valve shown in **FIG.** 2 is operated exactly in the opposite direction to the injection valve shown in **FIG.** 1. For safety reasons, the injection valve is designed such that, when the piezoelectric multilayer actuator **8** is not actuated, the injection valve is closed, i.e., the needle tip **102** is pressed against the valve seat **104**.

When the internal combustion engine is started, the piezoelectric multilayer actuator **8** is activated at the same time. The latter extends and thereby pushes the primary piston **21** into the first bore portion **31**. The resulting pressure increase in the hydraulic chamber **22** is compensated via the throttle gap **36** on the secondary piston **23** and the throttle gap **37** on the primary piston **21**, which ensure an exchange of hydraulic fluid with the storage chamber **24**. As a result, a state of equilibrium is then established again within fractions of a second, in which the injection valve remains closed. The injection valve is operated in such a way that the valve needle **3** always opens when the piezoelectric multilayer actuator **8** is discharged and the primary piston **21** and thus also the secondary piston **23**to which the valve needle **3** is fastened thereby withdraw.

FIG. 3 shows a further embodiment of an outwardly opening injection valve in which the base pressure on the hydraulic fluid in the transmission element is generated by a compression spring **128** arranged centrally in the primary piston **121**. For this purpose, the primary piston **121** is cup-shaped, an additional third storage chamber region **243** being formed between the head plate **10** and a bottom surface of the primary piston **121**. The primary piston **121** is arranged in such a way that it extends with its bottom surface into the first bore portion **31**. The side walls of the primary piston **21**, on the other hand, are located essentially in the housing region **34** in which the drive unit is arranged.

The compression spring **128** arranged in the third storage chamber region **243** is welded between the head plate **10** of the drive unit and a pressure plate **127**which is provided in the third storage chamber region **243**. In this case, the compression spring **128** is protected towards the third storage chamber region **243** by a metal bellows **125**.

The second storage chamber region **242** is formed between the head plate **10** of the piezoelectric multilayer actuator **8** and the housing step **35** around the wall of the primary piston **121**. This second storage chamber region **242** is connected via a connecting bore **136** to the third storage chamber region **243** in the primary piston **121**. The mode of operation of the transmission element shown in **FIG. 3** corresponds to the transmission element shown in **FIG. 1**. By providing an internal compression spring **128**, however, a higher base pressure and a possibly more compact design of the transmission element can be achieved.

Instead of the internal compression spring **128**, the gas-tight volume bounded by the metal bellows **125** and the head plate **10** can also be acted upon by a compressed gas, so that instead of the mechanical compression spring **128**, a gas compression spring ensures the maintenance of the basic pressure in the storage chamber regions **241**, **242**, **243**.

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The features of the invention disclosed in the above description, the drawings and the claims can be of significance both individually and in any combination for the realization of the invention in its various embodiments.

Patent claims

1. Device for transmitting a movement of an actuator (**8**) to a final control element (**3**) having a transmission element which establishes an operative connection between the actuator (**8**) and the final control element (**3**) and defines a hydraulic chamber (**22**) and a storage chamber (**24**) which are filled with a hydraulic fluid and are connected to one another via at least one throttle gap (**36**, **37**), **characterized** in that the transmission element has a first and a second piston element (**21**, **23**; **121**), the first piston element (**21**; **121**) being fixedly connected to the actuator (**8**) and the second piston element (**23**) being fixedly connected to the final control element (**3**), the hydraulic chamber (**22**) being formed between the first piston element (**21**; **121**) and the second piston element (**23**), and the storage chamber (**24**) comprising a pressure-loaded storage chamber region (**242**, **243**), the region limits of which are formed elastically.

2. Device according to claim 1, characterized in that the storage chamber region (**242**, **243**) acted upon by pressure is delimited by a spring bellows arrangement (**25**, **26**; **125**).

3. Device according to claim 1 or 2, characterized in that the storage chamber region (**242**, **243**) acted upon by pressure is acted upon by a compression spring (**28**) via a pressure plate (**27**; **127**).

4. Device according to claim 1 or 2, characterized in that the pressure-loaded storage chamber region (**242**, **243**) is acted upon by a gas pressure spring.

5. Device according to one of claims 1 to 4, characterized in that the first piston element (**21**; **121**) and the second piston element (**23**) are arranged in a two-stage inner bore region of a housing (**1**), the first piston element (**21**; **121**) and the second piston element (**23**) being spaced apart from one another in such a way that the hydraulic chamber (**22**) is formed between the mutually opposite end faces in the region of the transition from a first bore portion (**31**) to a second bore portion (**32**), the stroke of the first piston element (**21**; **121**) being transmitted to the second piston element (**23**) in accordance with the ratio of the end faces.

6. Device according to claim 5, characterized in that the storage chamber (**24**) is designed in two parts, having a first storage chamber region (**241**), which is delimited by the lower end face of the second piston element (**23**) and a bushing (**14**, **15**, **16**) on the final control element (**3**), and a second storage chamber region (**242**), which is arranged in a housing region (**34**) accommodating the actuator and is connected to the first storage chamber region (**241**) via a connecting bore (**223**), the second storage chamber region (**242**) being delimited by two spring bellows (**25**, **26**) arranged concentrically with respect to one another and a pressure ring (**27**), on which a compression spring (**28**) rests.

7.Device according to claim 5, characterized in that the first piston element (**121**) is of cup-shaped design and the storage chamber (**24**) is of three-part design, having a first storage chamber region (**241**), which is delimited by a lower end face of the second piston element (**23**) and a passage (**14**, **15**, **16**) on the actuator (**3**), a second storage chamber region (**242**), which is formed in a housing region (**34**) which accommodates the actuator (**8**) around the first piston element (**121**), and a third storage chamber region (**243**), which is formed in the first piston element (**121**), the first storage chamber region (**241**) being connected to the second storage chamber region (**242**) via a first connecting bore (**223**) in the housing (**1**) and the second storage chamber region (**242**) being connected to the third storage chamber region (**243**) via a second connecting bore (**136**) in the first piston element (**121**), and a compression spring (**128**), which is arranged between a head plate (**10**) of the actuator (**8**) and a compression plate (**127**) and is clamped by a spring pressure or a gas spring (**125**) in the third storage chamber (**243**). **8**. Device according to one of claims **1** to **7**, characterized in that the actuator (**8**) is a piezoelectric multilayer actuator which is prestressed by a spring element (**9**), and in that the first piston element (**21**; **121**) is fixedly attached to a head plate (**10**) of the multilayer actuator (**8**).

9. Device according to one of claims 1 to 7, characterized in that the actuator operates according to the electrostrictive or magnetostrictive principle.

10. Device according to one of claims 1 to 9, characterized in that the spring bellows (**25**, **26**; **125**) are metal bellows, which are preferably composed of semicircular segments with straight sections located in each case between them.

11. Device according to one of claims 1 to 10, characterized in that the spring bellows (**25**, **26**; **125**) are made of an elastomeric material. **12**. Device according to one of claims 1 to 11, characterized in that the hydraulic chamber (**22**) is designed in such a way that the spring constant of the hydraulic chamber (**22**) is at least 10^8 N/m.

Related 3 page(s) of drawings

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Fig. 1

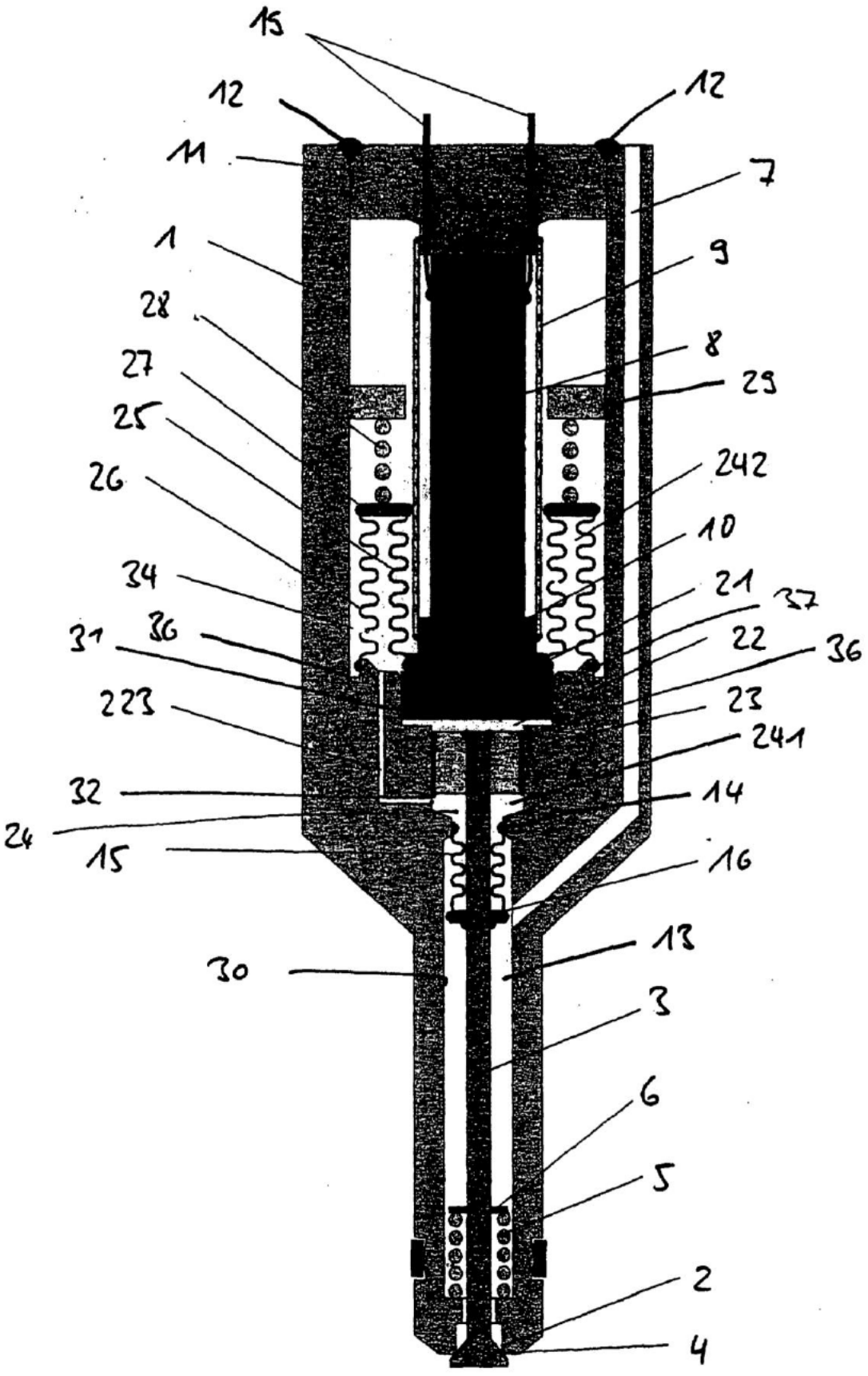

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Fig. 2

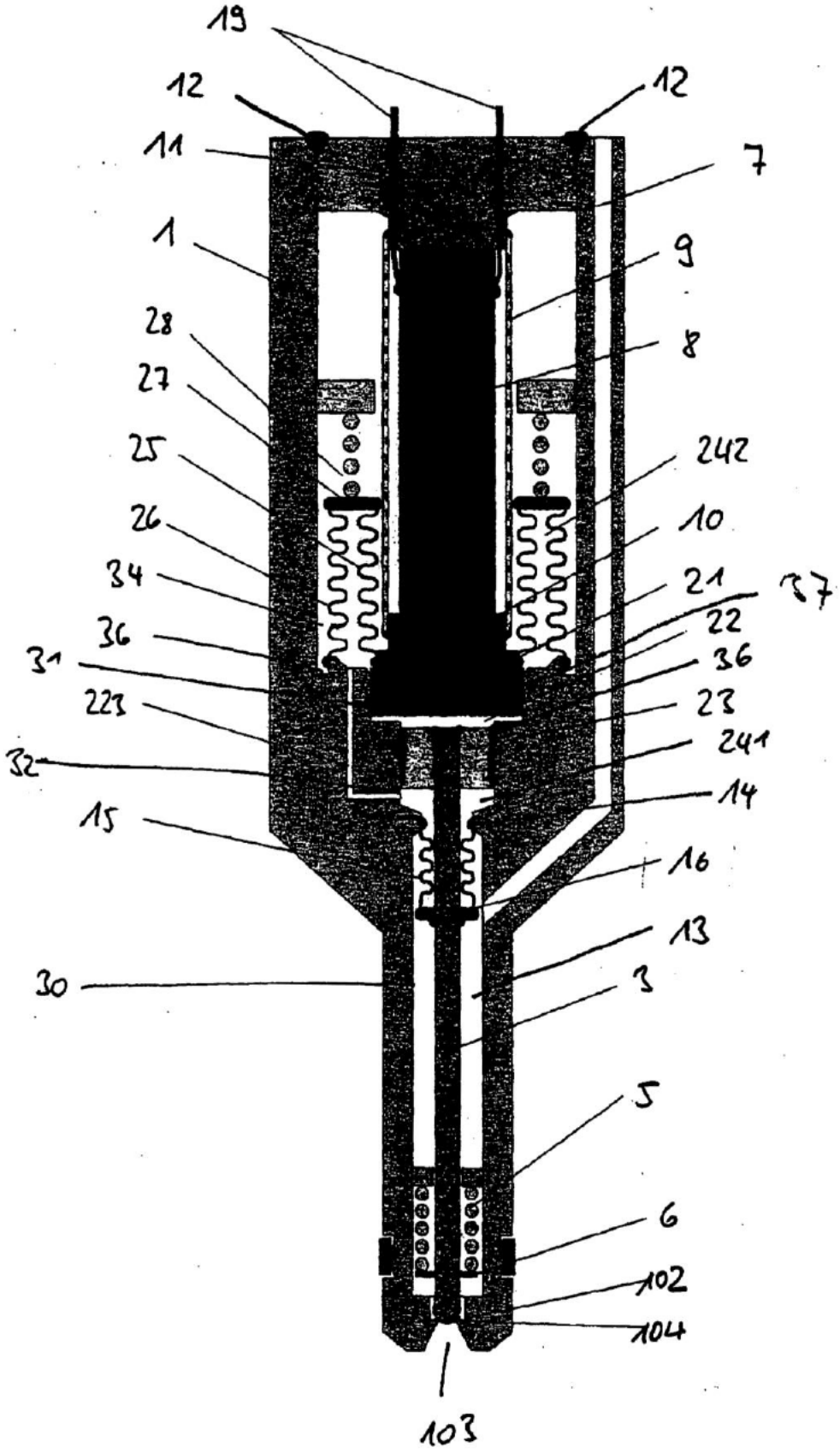

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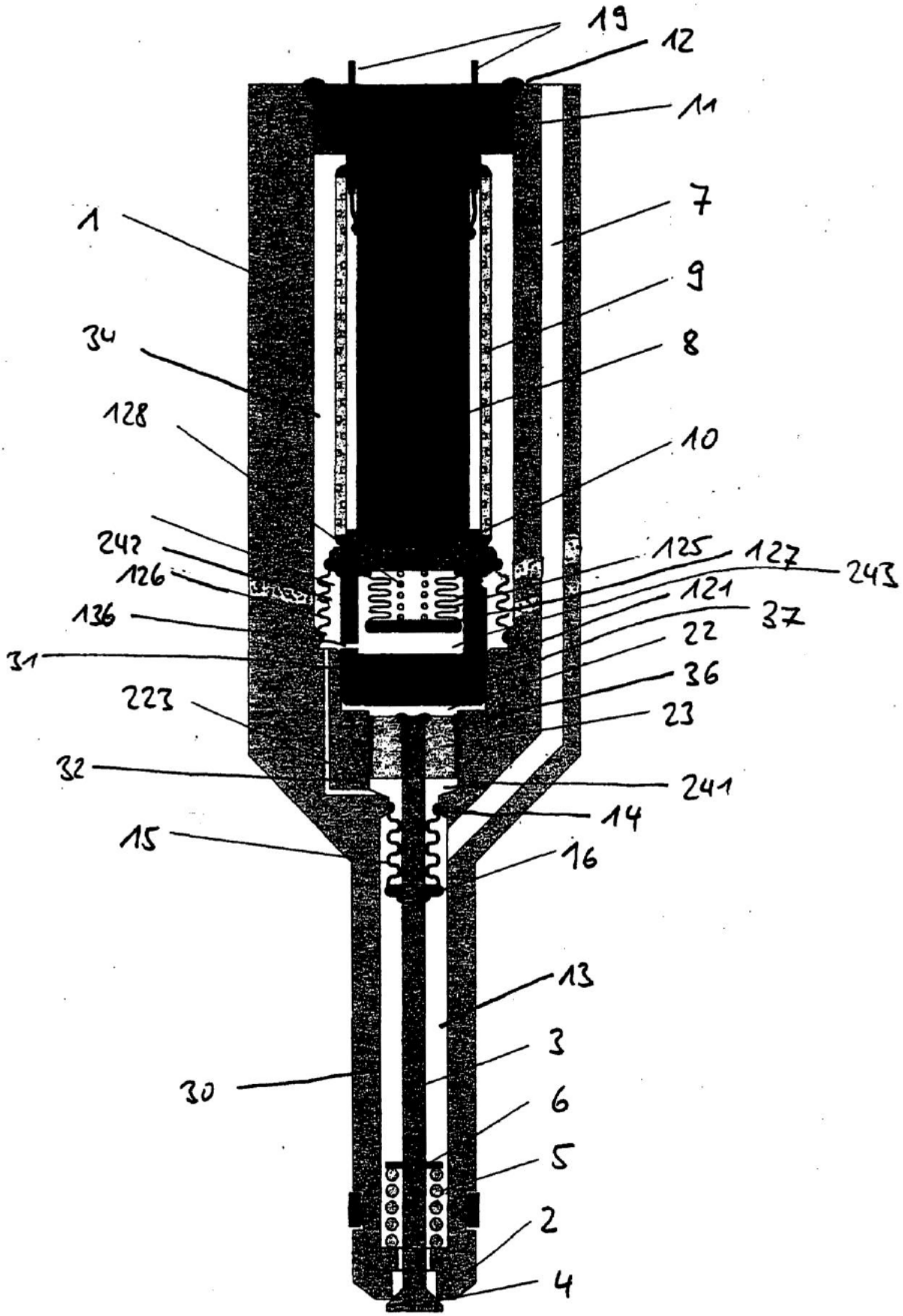
Fig. 3



TRANSLATION CERTIFICATION

Date: March 6, 2023

To whom it may concern:

This is to certify that the attached translation from German into English is an accurate representation of the documents received by this office.

The documents are designated as:

- DE19946732B4 (Schmutzler)
- DE19962177A1 (Gottlieb2)
- DE19856617A1 (Klugl2)
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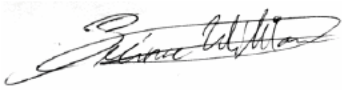
Prince Williams, Project Manager, attests to the following:

"To the best of my knowledge, the aforementioned documents are a true, full and accurate translation of the specified documents. I affirm that the provided translation was produced under the ISO 9001:2015 certified quality management process and the agents responsible for the specified translations are qualified to translate and review documents for the above language pair."



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