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(54) **FLUID PRESSURE ACTUATOR**

(57) A fluid pressure actuator (10) comprises a cylinder (11), a plunger (16), and a closed-loop speed control system for adjusting an extension speed of the plunger while moving. The closed-loop speed control system comprises a screw shaft (21) which is rotatably supported by the cylinder and extending axially in a hollow interior (19) of the plunger, an internally threaded member (22) which is disposed in the cylinder and receiving the screw shaft in close fitting relation thereto, and a proportional valve (23) which is rigidly connected to the plunger. The valve adjusts a volume flow rate of the pressurized fluid, when applied to operate the plunger, and the threaded member is constrained to an axial motion along the rotating screw shaft and relative to the cylinder. The threaded member is coupled to the valve such that a cross-sectional area of a fluid passageway through the valve is adjusted in order to compensate for a difference in speed magnitudes of the moving plunger and the oppositely moving threaded member in respect of the cylinder.

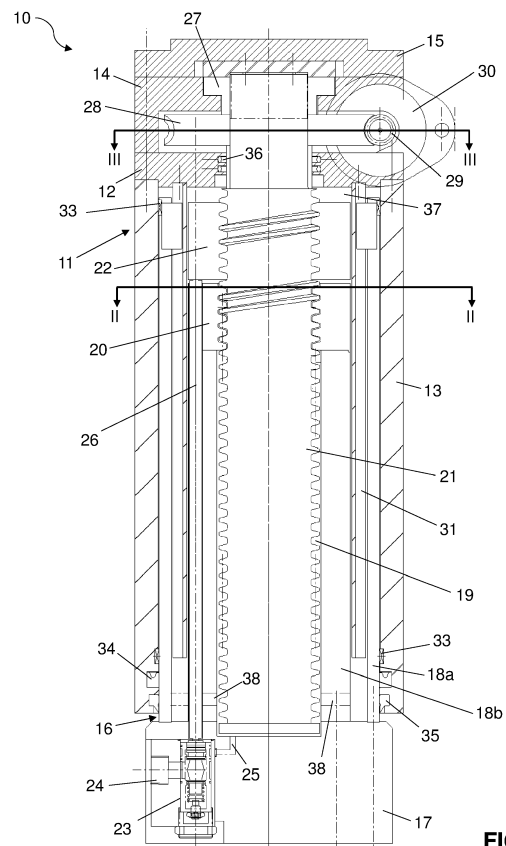


FIG. 1

Description

Field of the invention

[0001] The present invention relates to safe linear fluid pressure actuators. More specifically, it relates to linear hydraulic actuators with cylinder speed control and self-locking means.

Background of the invention

[0002] The lifting of very heavy loads such as oil tanks and bridges is often realized by the concerted action of linear hydraulic actuators. A uniform stroke is desirable for each hydraulic actuator, guaranteeing that the applied load does not tilt or slip during the heaving process. A non-uniform stroke, fluid supply failure or depressurization of one or more actuators can have catastrophic consequences if the load tilts and starts slipping during the heaving process.

[0003] Pairing each hydraulic actuator with a backup actuator is a known safety measure which has the disadvantage that it requires the installation and operation of at least twice as many actuators.

[0004] Document WO2005/017366 A1, Supraventures AG (24-02-2005), discloses a double-acting fluid actuator in which a spindle is received in an internally threaded rod piston to convert an axially directed piston motion into a rotary motion of a spinning disc located outside the fluid pressure chambers of the actuator. The spinning disc can be brought into a ratchet connection with a locking piston such that a counterrotation of the spindle and a downward movement of the piston is prevented in the case of depressurization of the fluid chamber. Alternatively, the spinning disc can be engaged with a gear that couples the rotary motion of disc to the axial motion of a valve shaft such that a valve adject to the actuator cylinder closes as soon as the piston has travelled a predetermined distance. The disclosed fluid actuator has the disadvantage that only control of the final piston position is obtained, but not of the speed at which the piston rod extends. Moreover, the locking mechanism of the fluid actuator requires a separately pumped fluid chamber and locking piston, which increases the cost and complexity of the actuator.

[0005] It is desirable to provide safe and easy to assemble fluid pressure actuators which provide a good control of their extension speed during load lifting.

Summary of the invention

[0006] It is an object of embodiments of the present invention to provide a safe and compact fluid pressure actuator which is capable of reliably controlling the extension speed.

[0007] The above objective is accomplished by a device and system according to the present invention.

[0008] In one aspect the present invention relates to a

fluid pressure actuator which comprises a cylinder and a plunger slidably mounted therein. The plunger extends axially relative to the cylinder upon application of a pressurized fluid. The fluid pressure actuator also includes a closed-loop speed control system for adjusting an extension speed of the plunger while moving. This speed control system comprises a screw shaft, an internally threaded member and a proportional valve. The screw shaft is rotatably supported by the cylinder and extends axially in a hollow interior of the plunger. The internally threaded member is disposed in the cylinder and threadably receiving the screw shaft in close fitting relation thereto. Furthermore, the internally threaded member is restrained from rotation such that a rotary motion of the screw shaft translates into an axial motion of the internally threaded member relative to the cylinder. The proportional valve is rigidly connected to the plunger for adjusting a volume flow rate of the pressurized fluid, when applied to operate the plunger. In addition thereto, the internally threaded member is operatively coupled to the valve such that a cross-sectional area of a fluid passageway through the valve is reduced or increased in order to compensate for a difference in the speed magnitudes, with respect to the rest frame of the cylinder, of the moving plunger and the oppositely moving internally threaded member.

[0009] In embodiments of the invention, the fluid pressure actuator may be a single-acting hydraulic or pneumatic actuator. Embodiments of the invention have the advantage that a compact fluid pressure actuator is provided which does not require expensive control valve systems that are external to the fluid pressure actuator in order to provide feedback on the extension speed.

[0010] In embodiments of the invention, an interior surface of the plunger, delimiting the hollow interior thereof, may comprise a threaded portion for threadably receiving the screw shaft in loose fitting relation thereto. The threaded connection between the threaded portion on the plunger interior surface and the screw shaft is suitable for supporting the cylinder in the absence of the pressurized fluid and for converting a rotary motion of the screw shaft into an axially directed extension of the plunger relative to the cylinder, when the pressurized fluid is applied. An axial backlash associated with the loose fitting relation is greater than a full stroke associated with the proportional valve. Such embodiments have the advantage that the fluid pressure actuator is provided with a safety mechanism in case of a failure of the fluid supply. Fluid pressure actuators according to some embodiments of the invention are capable of lifting axial loads of tens of kN, or more.

[0011] According to some embodiments, the fluid pressure actuator further comprises a worm drive for driving a rotary motion of the screw shaft. The worm drive may include a gear connected to the screw shaft and a worm meshing with the gear. In addition thereto, a motor may be provided which has a motor shaft coupled to the worm of the worm drive. Such a motor may be an electric motor

or a hydraulic motor. In such embodiments the fluid pressure actuator and its driving means are combined into a single device. Moreover, the self-locking connection of the worm and worm gear in the worm drive represents a further safety mechanism of the fluid pressure actuator in case of a failure of the fluid supply.

[0012] According to some embodiments, the proportional valve may be provided in the base member of the plunger. This has the advantage that the valve is easily accessible for the purposes of repair, replacement and/or adjustment.

[0013] According to some embodiments, the proportional valve and the driving means of the fluid pressure actuator are both provided on the same side of the fluid pressure actuator. They may both be located at the base of the plunger. This has the advantage that cabling or piping connections for operating and powering the fluid pressure actuator can be made shorter and do not represent additional loads that are lifted when the plunger is extending relative to the cylinder. Moreover, hanging and/or moving cabling or piping connections can be avoided, which makes an on-site operation of the fluid pressure actuator safer. A further advantage of having the proportional valve and the driving means of the fluid pressure actuator both provided on the same side of the fluid pressure actuator is that the fluid pressure actuator can be assembled with less and/or less complex pieces, making it more economical, because one does not have to account for relative rotations between the cylinder and the plunger.

[0014] According to some embodiments, the fluid pressure actuator may be equipped with rotation hindering means that prevent a rotation of the internally threaded member relative to the plunger. In same or other embodiments, further rotation hindering means may be provided to also prevent a rotation of the cylinder relative to the plunger.

[0015] In another aspect the present invention relates to a lifting system for lifting heavy loads or equipment. The lifting system comprising at least one fluid pressure actuator according to an embodiment of the first aspect, a pump for supplying pressurized fluid to the at least one fluid pressure actuator, and driving means for controlling the rotary motion of the screw shaft of each fluid pressure actuator. In preferred embodiments of the invention, in which the lifting system comprises a plurality of fluid pressure actuators, the driving means of all fluid pressure actuators are synchronized by synchronization means such that the extension speed of the plunger relative to the cylinder is the same for each fluid pressure actuator, regardless of the load that is applied individually to each fluid pressure actuator.

[0016] Particular and preferred aspects of the invention are set out in the accompanying independent and dependent claims. Features from the dependent claims may be combined with features of the independent claims and with features of other dependent claims as appropriate and not merely as explicitly set out in the claims.

[0017] For purposes of summarizing the invention and the advantages achieved over the prior art, certain objects and advantages of the invention have been described herein above. Of course, it is to be understood that not necessarily all such objects or advantages may be achieved in accordance with any particular embodiment of the invention. Thus, for example, those skilled in the art will recognize that the invention may be embodied or carried out in a manner that achieves or optimizes one advantage or group of advantages as taught herein without necessarily achieving other objects or advantages as may be taught or suggested herein.

[0018] The above and other aspects of the invention will be apparent from and elucidated with reference to the embodiment(s) described hereinafter.

Brief description of the drawings

[0019] The invention will now be described further, by way of example, with reference to the accompanying drawings, in which:

FIG. 1 is a cross-sectional view of a fluid pressure actuator according to an embodiment of the present invention.

FIG. 2 and FIG. 3 are top views of cross-sections taken along the lines II-II and III-III of the embodiment in FIG. 1.

FIG. 4 is a magnified view of the proportional valve shown in the embodiment of FIG. 1.

FIG. 5 shows an open and closed configuration of the proportional valve and the corresponding axial positions of the screw shaft.

FIG. 6 is a cross-sectional view of a fluid pressure actuator according to another embodiment of the present invention.

FIG. 7 is a magnified view of another valve which may be used in embodiments of the present invention.

[0020] In the drawings, the size of some of the elements may be exaggerated and not drawn on scale for illustrative purposes. The dimensions and the relative dimensions do not necessarily correspond to actual reductions to practice of the invention.

[0021] Any reference signs in the claims shall not be construed as limiting the scope.

[0022] In the different drawings, the same reference signs refer to the same or analogous elements.

Detailed description of illustrative embodiments

[0023] The present invention will be described with respect to particular embodiments and with reference to certain drawings but the invention is not limited thereto but only by the claims.

[0024] The terms first, second and the like in the description and in the claims, are used for distinguishing

between similar elements and not necessarily for describing a sequence, either temporally, spatially, in ranking or in any other manner. It is to be understood that the terms so used are interchangeable under appropriate circumstances and that the embodiments of the invention described herein are capable of operation in other sequences than described or illustrated herein.

[0025] Moreover, directional terminology such as top, bottom, front, back, leading, trailing, under, over and the like in the description and the claims is used for descriptive purposes with reference to the orientation of the drawings being described, and not necessarily for describing relative positions. Because components of embodiments of the present invention can be positioned in a number of different orientations, the directional terminology is used for purposes of illustration only, and is in no way intended to be limiting, unless otherwise indicated. It is, hence, to be understood that the terms so used are interchangeable under appropriate circumstances and that the embodiments of the invention described herein are capable of operation in other orientations than described or illustrated herein.

[0026] It is to be noticed that the term "comprising", used in the claims, should not be interpreted as being restricted to the means listed thereafter; it does not exclude other elements or steps. It is thus to be interpreted as specifying the presence of the stated features, integers, steps or components as referred to, but does not preclude the presence or addition of one or more other features, integers, steps or components, or groups thereof. Thus, the scope of the expression "a device comprising means A and B" should not be limited to devices consisting only of components A and B. It means that with respect to the present invention, the only relevant components of the device are A and B.

[0027] Reference throughout this specification to "one embodiment" or "an embodiment" means that a particular feature, structure or characteristic described in connection with the embodiment is included in at least one embodiment of the present invention. Thus, appearances of the phrases "in one embodiment" or "in an embodiment" in various places throughout this specification are not necessarily all referring to the same embodiment, but may. Furthermore, the particular features, structures or characteristics may be combined in any suitable manner, as would be apparent to one of ordinary skill in the art from this disclosure, in one or more embodiments.

[0028] Similarly it should be appreciated that in the description of exemplary embodiments of the invention, various features of the invention are sometimes grouped together in a single embodiment, figure, or description thereof for the purpose of streamlining the disclosure and aiding in the understanding of one or more of the various inventive aspects. This method of disclosure, however, is not to be interpreted as reflecting an intention that the claimed invention requires more features than are expressly recited in each claim. Rather, as the following claims reflect, inventive aspects lie in less than all fea-

tures of a single foregoing disclosed embodiment. Thus, the claims following the detailed description are hereby expressly incorporated into this detailed description, with each claim standing on its own as a separate embodiment of this invention.

[0029] Furthermore, while some embodiments described herein include some but not other features included in other embodiments, combinations of features of different embodiments are meant to be within the scope of the invention, and form different embodiments, as would be understood by those in the art.

[0030] In the description provided herein, numerous specific details are set forth. However, it is understood that embodiments of the invention may be practiced without these specific details. In other instances, well-known methods, structures and techniques have not been shown in detail in order not to obscure an understanding of this description.

[0031] FIG. 1 shows a cross-section of a fluid pressure actuator 10 according to a first embodiment in elevation. The fluid pressure actuator, which may be implemented as a hydraulic or pneumatic linear actuator, comprises a cylinder 11, which is an assembly containing a cylinder end cap 12, a cylinder bore tube or barrel 13, a cylinder head portion 14, and a load-bearing member 15. In the cylinder assembly 11, the cylinder barrel 13 provides outer sidewalls that project downwardly from the end cap 12. The fluid pressure actuator also includes a plunger 16, e.g. with a base 17, upwardly projecting outer and inner sidewalls 18a, 18b, and a drive nut member 20 firmly secured to the inner sidewalls 18b. The plunger 16 is slidably mounted in the cylinder 11 so that it can be extended and retracted axially relative to the cylinder when the fluid pressure actuator is in use, e.g. when the pressurized fluid is supplied to the cylinder to push out the plunger. In other words, the cylinder 11 and plunger 16 form pair of vertically telescoping sections. The load-bearing member 15 has a flat upper surface on which a load to be lifted is resting while the fluid pressure actuator is operated, e.g. when the pressurized fluid is supplied to the cylinder to push out the plunger, thereby lifting the load. In alternative embodiments of the invention, a load to be moved may be coupled differently to an end of the fluid pressure actuator, for example via a clevis, hub, or cup. Moreover, an interior surface of the plunger 16 is delimiting a hollow interior space 19 and comprises a threaded portion. In the present embodiment, this threaded portion corresponds to the internal thread of the drive nut member 20 rigidly connected to the plunger inner sidewall 18b. In alternative embodiments, the threaded portion may be provided as an internal thread of the plunger inner sidewall 18b, e.g. a thread which is directly formed on an interior surface of the inner sidewall 18b delimiting the hollow interior space 19. Providing a separate nut member comprising the threaded portion has the advantage that different materials may be selected for this nut member, e.g. comprising aluminum or bronze, and a rotation hindering slotted pipe 31, e.g. comprising

steel, to reduce wear through sliding friction.

[0032] The fluid pressure actuator 10 further comprises a closed-loop speed control system for adjusting an extension speed of the plunger while it is in relative motion with respect to the cylinder. The closed-loop speed control system comprises a screw shaft 21, an internally threaded member 22, e.g. a servo nut member, and a proportional valve 23. The screw shaft 21 is axially (e.g. vertically) extending in the hollow interior 19 of the plunger and rotatably supported by the cylinder 11. In particular, the cylinder head portion 14 houses a flanged bushing 27 that is acting as a plain bearing into which the upper end of the screw shaft 21 is journaled. A worm gear 28, fixedly connected to the screw shaft, is housed in the cylinder head portion 14 too. It is part of a worm drive that is used to drive a rotary motion of the screw shaft, e.g. to establish and maintain an rotational speed (e.g. measured in rpm: revolutions per minute) of the screw shaft that is controlling the cylinder speed, i.e. the linear travel velocity at which the plunger is extending from the cylinder under fluid pressure operation. Another part of the worm drive is the worm 29 itself, which meshes with the worm gear 28. Moreover, the screw shaft 21 threadably engages with the threaded portion provided on the interior surface of the plunger, e.g. with the drive nut member 20. This enables a conversion of the screw shaft rotary motion into an axial, linear motion of the cylinder and screw shaft relative to the plunger, hence an extension of the plunger out of the cylinder. It is noted that the screw shaft is only rotatably supported by the cylinder and the screw shaft is restrained from moving axially with respect to the cylinder. A thread form for the threaded connections of the screw shaft with the internally threaded member and/or drive nut member is preferably trapezoidal, but other non-limiting forms may be used, e.g. V-shaped thread, acme, square, etc. A trapezoidal thread form enables self-centering and allows for sufficiently large surfaces of static friction such that the screw shaft is self-locking even in the presence of heavy loads.

[0033] A motor 30 may be included in the fluid pressure actuator, e.g. secured to the cylinder head portion 14, such that a shaft of the motor is operatively coupled to the worm 29. This is further illustrated in FIG. 3, which is a top view of a cross-section taken along the liner III-III in FIG. 1. The motor 30 or the fluid pressure actuator may be equipped with torque control means which allow an accurate setting and feedback control of the amount of torque the motor is applying to the worm. Hence, the applied torque by the motor can be selected as a function of the targeted screw shaft rotational speed (e.g. rpm) or speed profile over time, which depends on the particular application. Non-limiting examples of a motor include an electrical motor, e.g. a DC stepper motor, or a hydraulic motor.

[0034] The internally threaded member 22 is disposed in the cylinder and threadably receives the screw shaft 21. In the present embodiment, a servo nut member is

provided as the internally threaded member and is arranged between the cylinder end cap 12 and the drive nut member 20. The internally threaded member 22 engages with the screw shaft in close fitting relation, e.g. an axial backlash or play associated with the internally threaded member 22 is insignificant compared to the axial backlash associated with the threaded interior surface portion of the plunger sidewall, e.g. the axial backlash associated with the drive nut member 20.

[0035] Furthermore, the internally threaded member 22 is constrained to linear axial motion along the screw shaft 21. In other words, the internally threaded member 22 is climbing or descending the screw shaft, depending on the direction of rotation of the screw shaft, but is restrained from rotating relative to the cylinder 11. This may be achieved by providing the fluid pressure actuator with rotation hindering means as illustrated with reference to FIG. 2, which is a top view of the cross-section taken along the line II-II in FIG. 1. One or more elongated, stiff members 201 are provided, e.g. rods or pins, which have one end portion firmly connected to the drive nut member 20 and another one end portion that is received in corresponding bore holes of the internally threaded member 22, whereby a relative rotation is avoided and a movement of the internally threaded member 22 is constrained to an axial direction. The drive nut member 20 itself is restrained from rotation relative to the plunger 16 by means of a plurality of bolts 203 which secure the drive nut member to the plunger inner sidewall 18b. In alternative embodiments, the one or more elongated, stiff members may be anchored in the plunger inner sidewall 18b, e.g. if the threaded portion is provided directly on the plunger inner sidewall 18b or if the elongated members traverse the drive nut member 20. Additionally, a grooved pipe 31 is connected to the cylinder end cap 12 at one end and extends downwardly in the gap formed between plunger outer sidewall 18a and the plunger inner sidewall 18b. Guiding blocks 232 operatively engage with the grooves 233 of the pipe 31 such that the pipe 31 is allowed to slide axially between the sidewalls 18a, 18b of the plunger 16 but prevented from rotating relative to the plunger 16. Since the pipe 31 is secured to the cylinder 11 assembly, e.g. at the end cap 12, a rotation between the cylinder 11 and the plunger is effectively suppressed. FIG. 2 also shows the connecting element 26, e.g. rod, which couples the axial movement of the internally threaded member to a displacement of a valve member, e.g. spool, in the proportional valve 23.

[0036] The internally threaded member 22 is operatively connected to the proportional valve 23 such that a cross-sectional area of a fluid passageway through the valve 23 is reduced or increased, thereby increasing or reducing the volume flow rate of a fluid supplied to the valve at a constant pressure, in order to compensate for a difference in speed magnitudes of the moving plunger and the oppositely moving internally threaded member, both moving in the rest frame of the cylinder. The fluid passageway whose cross-sectional area is controlled by

the valve 23 typically corresponds to a small orifice, opening, or gap in the valve body, the dimension (e.g. diameter, annular width, etc.) of which can be adjusted by a valve member. The valve member is moved into the fluid passageway to progressively obstruct a fluid flow across the passageway. In contrast to a discrete (directional) valve, a proportional valve allows for indefinitely many positions of the valve member, and thus for indefinitely many volume flow rate of a pressurized fluid there-through.

[0037] FIG. 4 is an enlarged view of the proportional valve 23 in the cross-section in FIG. 1. The proportional valve 23 comprises a valve casing 40 that is lodged in the plunger base 17, a valve sleeve 41 secured in the valve casing 40, and a valve member 42, e.g. spool, slidably supported in the valve sleeve 41 to allow axial displacements of the valve member 42 relative to the valve sleeve 41. Alternatively, the plunger base 17 itself may act as a casing for the valve 23. The valve sleeve 41 has an internal profile that matches with protrusions of the valve member 42, e.g. landings on the spool. An inlet conduit is provided in the valve casing 40 for receiving a pressurized fluid, e.g. a constant pressure fluid that is supplied by an external fluid pump, and directing it towards an input port 24a of the valve, and an outlet conduit 25 is formed in the plunger base 17 for directing the pressurized fluid that is leaving the valve at an output port 25a thereof towards the interior hollow space 19 of the plunger. In FIG. 4, the valve is shown in a closed position, in which an annular protrusion 44 of the sleeve inner profile is contacting a landing 43 of the valve member 42 in a sealing manner. In some embodiments of the invention the annular protrusion 44 of the sleeve may act as a seat for the landing 43 of the valve member 42, i.e. the protrusion 44 and the landing overlap in a radial direction when viewed from the top along the cylinder axis. In other embodiments of the invention, the landing 43 may slide axially through the protrusion 44 while the protrusion encloses the landing 43 in a sealing manner. In contrast thereto, an open position of the valve 23 is associated with a gap between the landing 43 and the protrusion 44, wherein an axial length of the gap determines the volume flow rate for the pressurized fluid that is admitted into the hollow interior space 19 via the outlet conduit 25. A resilient member 45, e.g. a spring, is receiving an end portion of the valve member 42 and is biased such that the valve member is urged into contact with the connecting member 26, e.g. rod, both for a closed position and an open position of the valve 23. An axial thrust associated with the internally threaded member 22 travelling down the rotating screw shaft 21 is transmitted to the valve member 42 through the connecting member 26 and reacts against the restoring force of the compressed spring 45, which results in the axial displacement of the valve member 42, which forces the valve into an open position. Only if the plunger is extending relative to the cylinder at a speed that is greater than the travel speed of the internally threaded member down the screw shaft

and relative to the cylinder, the restoring force of the spring 45 exceeds the axial thrust of the internally threaded member and the valve is closing. The resilient member 45, e.g. spring, hold in place by a retention plate 46, and an adjustment screw 47 is contacting the retention plate 46 and compresses the resilient member 45 to obtain the above-mentioned bias. A further adjustment screw 48 allows a more accurate axial positioning of the sleeve 41 in the casing 40, which has the advantage that length variations of the connecting member 26 can be compensated. Although the valve 23 is located in the plunger base 17 in the present embodiment, this is only one possible way of accommodating the valve in the fluid pressure actuator. The valve 23 shown in FIG. 4, or an equivalently operating valve, may also be located in the plunger inner sidewall 18b or in the drive nut member 20. Accommodating the valve 23 in the plunger base 17 has the advantage that the valve is easier to access, e.g. for repair or replacement, or for readjustment of the screws 47, 48.

[0038] Referring back to FIG. 1, radially outwards directed fluid channels 38 are provided in the inner plunger sidewall 18b to facilitate the flow of the pressurized fluid from the interior hollow space 19 near the outlet conduit 25 towards the fluid pressure chamber 37, where the pressurized fluid is acting on the cylinder end cap 12 to extend the plunger 16 relative to the cylinder 11. The fluid channels provide a passageway for the pressurized fluid to the pipe-receiving gap between the inner and outer plunger sidewalls 18a, 18b, which guides and discharges the fluid into the fluid pressure chamber 37. The flow of the pressurized fluid is facilitated, because the narrow interstitial fluid pathway through the meshing male and female threads of the screw shaft and the drive nut member 20, and also the internally threaded member 22, can be partially bypassed. Furthermore, the fluid pressure actuator 10 may also comprise a plurality of seals 34, 35 for sealing off a fluid pressure chamber 37, which is in fluid connection with the hollow interior 19 of the plunger. In particular, rod seal 34 provides a pressure barrier and may admit a thin lubricating film forming on the exterior surface of the plunger outer sidewall 18a while the plunger is extending, whereas a wiper seal 35 is accommodated in a stepped housing groove of the cylinder barrel 13 to prevent that external contaminants enter the actuator 10. The cylinder end cap 12 provides a movable upper boundary to the extendable fluid pressure chamber 37 onto which the supplied pressurized fluid is acting to extend the plunger 16 from the cylinder 11. Moreover, a pair of guide ring 33 center the cylinder barrel 13 and grooved pipe 31 relative to the plunger sidewalls 18a, 18b and avoid metal-on-metal sliding wear. A further seal 36 is arranged in the end cap 12 and is in a sealing relation with the screw shaft 21 to avoid that pressurized fluid reaches the cylinder head portion 14.

[0039] FIG. 6 shows a cross-sectional view of a fluid pressure actuator 60, which is a variant of the embodiment relating to FIG. 1. The fluid pressure actuator 60

differs from the embodiment of FIG. 1 in that the worm drive 28, 29 and motor 30 are provided at the same side of the fluid pressure actuator as the valve 23. More specifically, the worm gear 28 is housed in a lower portion 17b of the plunger base, whereas the valve is arranged in an upper portion 17a of the plunger base. Upper and lower portions 17a, 17b of the plunger base are secured one to another, e.g. via bolt connections. The worm 29 and motor 30 are secured to the lower portion 17b of the plunger base too. Having both the worm drive and motor arranged on the plunger base has the advantage that no long and moving electric cables or hydraulic pipes for powering the electric or hydraulic motor are required in comparison to the embodiment of FIG. 1. To rotate the screw shaft 21 in the present embodiment, a hollow lower portion of the screw shaft encloses a vertically extending grooved bar 65 and an annular disc 67 with a plurality of teeth on the inner circumferential edge, meshing with the grooves of the bar 65, is connected to the lower end face of the screw shaft 21. As a result thereof, a rotary motion of the grooved bar is transmitted to a rotation of the annular disc 67 and thus the screw shaft 21. An upper end of the grooved bar 65 is freely rotatable inside the hollow portion of the screw shaft, whereas the lower end of the grooved bar 65 is connected to a cylindrical extension 66 onto which the worm gear 28 is mounted. The grooved bar is rotatably supported in the plunger 16 by means of a rotary bearing 68 provided in the lower base portion 17b, but is restrained from axial movements. Likewise, the upper end of the screw shaft is journaled into a rotary bearing 61 for rotatably mounting the screw shaft in the cylinder 11 and prevented from axial movements relative to the cylinder.

[0040] Similar to the embodiment of FIG. 1, the fluid pressure actuator 60 includes pins 201 as rotation hindering means for preventing a relative rotation between the drive nut member 20 and the internally threaded member 22, wherein the drive nut member is fastened to the plunger inner sidewall 18b through bolt connections 203. The embodiment of FIG. 6 has the further advantage that less and/or less complex parts are used to assemble the actuator 60. For example, the plunger sidewall 18 of the fluid pressure actuator 60 consists of a single solid wall instead of an outer and an inner wall that are separated by a gap. Therefore, less material is required for the plunger 16 and the fluid pressure actuator can be built in a more lightweight and economical manner. Moreover, no rotation hindering pipe 31 is required, because action and reaction for rotation of the screw shaft 21 is applied from the same part. If rotation between cylinder and plunger would occur, this would not cause any height difference. Besides, a fluid channel 64 extends through the drive nut member 20 and the internally threaded member 22 to fluidly connect the hollow interior space 19 of the plunger to the fluid pressure chamber 37. This facilitates an inflow of pressurized fluid into the fluid pressure chamber by avoiding the high flow resistance associated with the engaging threads of nuts 20,

22 and screw shaft 21.

[0041] Fluid pressure actuators according to the above-described embodiments may be operated in the following way in order to lift a load. In a first step, a load is contacted with the flat upper surface of the load-bearing member 15. Under the applied load, the screw shaft 21 experiences axial forces and its thread rests on the upper flanks of the internal thread of the driving nut member 20. This corresponds to the situation which is depicted in the left half of FIG. 5 for an embodiment of the invention, in which the proportional valve 23 is lodged in the driver nut member 20 and the plunger inner sidewall 18b as combined valve sleeve 41 and housing. The threads of the screw shaft 21 and the drive nut 20 are adapted to engage in a self-locking manner even in the absence of the pressurized fluid in the fluid pressure chamber 37 and hollow interior 19, i.e. a frictional coupling between the drive nut member and the screw shaft is sufficiently large to prevent the screw shaft from sliding along the thread of the drive nut member and thus to prevent an uncontrolled downward movement of the cylinder relative to the plunger, which would result in the dangerous sudden acceleration of the applied load. In consequence, the drive nut member 20 also acts as a safety means. Additional safety measures in case of fluid supply failure or depressurization of the fluid pressure chamber are furnished by the self-locking connection between the worm and worm gear, when the worm drive is not in use, and the self-locking threaded connection between the screw shaft and the spring-loaded internally threaded member (servo nut) if the drive nut member fails to continue acting as primary safety means. The valve is in an open configuration, in which the a small annular gap region between a landing 43 and a matching protrusion 44 of the sleeve allow a pressurized fluid to pass, when applied to the valve through the inlet conduit 54. As the screw shaft 21 has sunken to rest on the upper flanks of the drive nut member, a vertical clearance "d" between the internally threaded member 22 on the screw shaft 21 and the drive nut member 20 is minimal. The resilient member 25, e.g. spring, is biased to urge the upper end of the valve member 42 into mechanical contact with the connecting rod 26, yet the restoring force of the resilient member 25 is too low to lift the weight of the screw shaft.

[0042] Next, pressurized fluid, e.g. mineral oil, is supplied to the inlet conduit of the fluid pressure actuator. For instance, an external pump or tank may be fluidly coupled to the inlet conduit to supply the fluid at a constant pressure. Alternatively, the pressurized fluid may be applied to the fluid pressure actuator before and during the step of applying the load. In this case, the engaging threads of screw shaft 21 and drive nut member 20 as well as the engaging threads of screw shaft 21 and internally threaded member 22 are already lubricated by a hydrostatic film of pressurized fluid. This has the effect that the pressurized fluid supports most of the applied load, resulting in small residual axial forces that act on the engaged threads of the screw shaft 21 and drive nut

member 20, which minimizes internal stress and wear. Once the axial forces of the applied load have been compensated by the pressure force of the fluid in the fluid pressure chamber 37, acting on the cylinder end cap 12, a further inflow of the pressurized fluid into the fluid pressure chamber 37 causes an upwards directed displacement of the cylinder. This frees the screw shaft from the upper flanks of the drive nut member, which can now rotate freely and without significant frictional losses (due to the lubricating fluid) at a rotational speed set by the driving means, e.g. motor 30 coupled to the screw shaft via the worm drive 28, 29. To extend the plunger in FIG. 5, the screw shaft is rotated counter-clockwise (from left to right). The drive nut member 20 is translating the rotary motion of the screw shaft into an upward directed motion thereof and the receding surface of the cylinder end cap 12 increases the volume of the fluid pressure chamber. For a particular volume flow rate Q associated with a position of the valve member with respect to a closed position, the extension speed of the cylinder relative to the plunger is given as $v_1 = Q/A$, with A being the projected surface area of the receding end cap 12 onto which the pressurized fluid is acting. At the same time, the rotary motion of the screw shaft causes the internally threaded member to travel downwards relative to the cylinder, e.g. at a linear travel speed v_2 determined by the product of rotational speed of the screw shaft in rpm and lead L of the screw shaft thread, $v_2 = \text{rpm} \cdot L$. Consequently, the travel speed v_1 of the upward motion of the internally threaded member 22 together with the screw shaft 21 and cylinder 11 is partially or completely compensated by the travel speed v_2 of the downward motion of the internally threaded member 22 relative to the screw shaft and cylinder. If the travel speed v_1 relating to the upward motion of the internally threaded member 22 is larger than the travel speed v_2 relating to the downward motion, the vertical clearance "d" is increasing and the resilient member 25 urges the valve member farther towards the closed position, thereby reducing the valve orifice 50 and reducing the volume flow rate. Contrarily, if the travel speed v_1 relating to the upward motion of the internally threaded member 22 is less than the travel speed v_2 relating to the downward motion, the vertical clearance "d" is decreasing and the connecting member 26 further compresses the resilient member 25, displacing the valve member farther away from the closed position, thereby increasing the valve orifice 50 and increasing the volume flow rate. It follows that the combination of screw shaft 21, internally threaded member 22 and valve 23 acts as a feedback control system for the extension speed of the plunger relative to the cylinder and that an axial position of the valve member relative to the closed position is a function of the rotational speed of the screw shaft and the applied fluid pressure. If there was a sudden drop in fluid pressure, the safety mechanism associated with the self-locking engagement of the screw shaft thread with the resting flanks of the drive nut member thread would prevent the load from being accelerated downwards or

even falling. Ultimately, if the plunger has been extended by a targeted length, which may be less or equal to the full stroke of the fluid pressure actuator and which may be determined by the product of number of screw shaft rotations and screw shaft thread lead, the motor stops driving the screw shaft. There is a response delay during which the speed control system adapts to the non-rotating condition of the screw shaft and the valve member is moved to the closed position in which the vertical clearance is minimal, e.g. $d_{\min} = 4,53 \text{ mm}$ compared to $d_{\max} = 5,33 \text{ mm}$ for a screw shaft thread lead of 9,00 mm and an axial backlash of about 1, 00 mm for the drive nut member engaging the screw shaft. This is depicted in the right half of FIG. 5, which depicts the valve in the closed configuration and the resulting axial position of the internally threaded member 22 relative to the drive nut member 20. In the closed configuration of the valve, the screw shaft thread has maximally approached the lower flanks of the drive nut member. Typically, the axial backlash between the threaded connection of screw shaft and drive nut member is selected to be slightly larger than the full stroke of the valve member between the close position and the maximally open position so that the screw shaft thread does not contact the lower flanks of the drive nut member. This guarantees that the proportional valve can indeed reach the closed position in the presence of variations in the thread lead and/or axial backlash, e.g. before an upwards directed axial movement of the screw shaft to close the valve is hindered by the fact that its thread is contacting the lower flanks of the drive nut member.

[0043] Eventually, the pressurized fluid can be vacated from the fluid pressure chamber 37 to retract the plunger into the cylinder. This can be done in a pressurized actuator with free, fluid supported screw shaft and closed valve by lowering the pressure at which fluid is supplied. Next, the screw shaft is briefly driven in the counter-clockwise direction before reversing the rotation direction into the clockwise direction. This takes up some or almost all of the axial backlash and allows the internally threaded member to travel downwards, whereby the valve is opened again. The overpressure in the fluid pressure chamber 37 relative to the lower supply pressure of the fluid reverses the flow direction of the fluid. The fluid pressure actuator responds to the outflowing fluid by moving the cylinder end cap 12 and cylinder 11 towards the plunger 16, decreasing the volume of the fluid pressure chamber and retracting the plunger into the cylinder. Hence, the screw shaft descends together with the cylinder, as does the internally threaded member 22 on the screw shaft. Before the screw shaft contacts the upper flanks of the drive nut member thread and thus avoiding self-locking of the threaded connection, the rotary motion of the screw shaft is reversed to obtain an upward directed travel of the internally threaded member relative to the cylinder. However, in order to continue the retraction of the plunger, the upward directed motion of the internally threaded member due to the screw shaft rotation is

now compensated by the downward direction motion of the internally threaded member due to the descending screw shaft and a steady outflow of the fluid through the valve is maintained.

[0044] A single fluid pressure actuator according to embodiments of the invention may be used in a hydraulic jacking device or a telescopic crane, for example. For the latter application the fluid pressure actuator may also be oriented and extended horizontally. However, a plurality of fluid pressure actuator according to embodiments of the invention may be used synchronously to lift heavy equipment such as oil tanks or bridges, e.g. in a jacking and cribbing process.

[0045] The present invention therefore also relates to a lifting system for heavy loads. The system includes at least one fluid pressure actuator according to an embodiment of the previous aspect of the present invention, a pump for supplying pressurized fluid to the at least one fluid pressure actuator, and driving means for driving and controlling a rotary motion of the screw shaft of each fluid pressure actuator. The driving means of each fluid pressure actuator may include an electric or hydraulic motor. Preferably, the lifting system also includes synchronization means that synchronize the individual motors of the fluid pressure actuators such that the plunger of each actuator is extended by the same length per time interval. This ensures that the heavy equipment, e.g. a tank such as for instance an oil tank, or building constructions such as a bridge, is lifted in unison by the plurality of fluid pressure actuators in a smooth and controlled manner. An advantage of such a lifting system is that the extension speed of each commanded fluid pressure actuator is auto-regulated to the assigned linear travel speed of the screw shaft, e.g. by configuring the motor to apply a torque to the screw shaft such that this linear travel speed is obtained. Therefore, the heavy equipment can be lifted continuously without having to permanently verify the travelled distance of each fluid pressure actuator to avoid a dangerous tilted configuration of the heavy equipment. The lifting system also prevents an uncontrolled overshoot of the extended fluid pressure actuators once the driving of all the fluid pressure actuators has been stopped. Moreover, the inherent safety mechanism associated with the self-locking threaded connection between the drive nut member and the screw shaft guarantees that the heavy equipment cannot inadvertently slip and fall if the fluid pressure supply fails, because the differences in screw shaft thread lead and axial backlash across a plurality of fluid pressure actuators is too low to cause a sliding of the lifted equipment. In particular, fluid pressure actuators do not have to be provided and installed pairwise to obtain a safe lifting process in the event that the fluid pressure operation in the fluid pressure actuator of a pair would fail. An exemplary fluid pressure actuator for lifting up bridges is a hydraulically operated linear fluid pressure actuator which has a stroke of about 25 mm for a trapezoidal thread form of the threaded connections between screw shaft and drive nut member with

lead of about 9 mm and screw shaft minor diameter of about 60 mm. However, embodiments of the invention can be conceived to have larger strokes, e.g. a stroke up to 200 cm.

[0046] In such embodiments where a plurality of fluid pressure actuators according to embodiments of the present invention are implemented, the valve used is preferably of a type where the action for rising and descending is implemented in a similar way. FIG. 7 is an enlarged view of such valve 70 in cross-section. Such valve 70 could be used in any of the embodiments of fluid pressure actuators according to the present invention, although not illustrated in detail as such. The valve 70 comprises a valve casing 40 that is lodged in the plunger base 17, a valve sleeve 41 secured in the valve casing 40, and a valve member 42, e.g. spool, slidably supported in the valve sleeve 41 to allow axial displacements of the valve member 42 relative to the valve sleeve 41. Alternatively, the plunger base 17 itself may act as a casing for the valve 23. The valve sleeve 41 has an internal profile that matches with protrusions of the valve member 42, e.g. landings on the spool. For the rising action, an inlet conduit "in" is provided in the valve casing 40 for receiving a pressurized fluid, e.g. a constant pressure fluid that is supplied by an external fluid pump, and directing it towards an input port 24a of the valve, and an outlet conduit 25 is formed in the plunger base 17 for directing the pressurized fluid that is leaving the valve at an output port 25a thereof towards the interior hollow space 19 of the plunger. For the descending action, fluid coming from the interior hollow space 19 of the plunger may be introduced via conduit 25, and evacuated via outlet conduit "out".

[0047] In FIG. 7, the valve 70 is shown in a closed position for the rising action, in which an annular protrusion 44 of the sleeve inner profile is contacting a landing 43 of the valve member 42 in a sealing manner. In some embodiments of the invention the annular protrusion 44 of the sleeve may act as a seat for the landing 43 of the valve member 42, i.e. the protrusion 44 and the landing overlap in a radial direction when viewed from the top along the cylinder axis. In other embodiments of the invention, the landing 43 may slide axially through the protrusion 44 while the protrusion encloses the landing 43 in a sealing manner. In contrast thereto, an open position of the valve 70 for rising is associated with a gap between the landing 43 and the protrusion 44, wherein an axial length of the gap determines the volume flow rate for the pressurized fluid that is admitted into the hollow interior space 19 via the outlet conduit 25.

[0048] In FIG. 7, the valve 70 is shown in a closed position for the rising action, in which an annular protrusion 44 of the sleeve inner profile is contacting a landing 43 of the valve member 42 in a sealing manner. In some embodiments of the invention the annular protrusion 44 of the sleeve may act as a seat for the landing 43 of the valve member 42, i.e. the protrusion 44 and the landing overlap in a radial direction when viewed from the top

along the cylinder axis. In other embodiments of the invention, the landing 43 may slide axially through the protrusion 44 while the protrusion encloses the landing 43 in a sealing manner. In contrast thereto, an open position of the valve 70 for rising is associated with a gap between the landing 43 and the protrusion 44, wherein an axial length of the gap determines the volume flow rate for the pressurized fluid that is admitted into the hollow interior space 19 via the outlet conduit 25.

[0049] Besides the protrusion 44 and the landing 43 at one side of the valve 70, used for the rising action, another protrusion 74 and another landing 73 may be provided at the other side of the valve 70, to allow the valve 70 to function in a similar way in case of rising and descending actions of the fluid pressure actuator.

[0050] In FIG. 7, the valve 70 is shown in an open position for the descending action, in which there is a gap between the other protrusion 74 of the sleeve inner profile and the other landing 73 of the valve member 42. An axial length of the gap determines the volume flow rate for the pressurized fluid that is admitted to flow from the hollow interior space 19 via the conduit 25 towards the outlet "out". In a closed position, the other protrusion 74 and the other landing 73 would contact in a sealing manner. In some embodiments of the invention the other protrusion 74 of the sleeve may act as a seat for the other landing 73 of the valve member 42, i.e. the other protrusion 74 and the other landing 73 overlap in a radial direction when viewed from the top along the cylinder axis. In other embodiments of the invention, the other landing 73 may slide axially through the other protrusion 74 while the other protrusion 74 encloses the other landing 73 in a sealing manner.

[0051] As in the embodiment illustrated in FIG. 4, also in the valve 70 illustrated in FIG. 7, a resilient member 45, e.g. a spring, is provided for receiving an end portion of the valve member 42 and is biased such that the valve member is urged into contact with the connecting member 26, e.g. rod, both for a closed position and an open position of the valve 70, both in rising and in descending actions. An axial thrust associated with the internally threaded member 22 travelling down or up the rotating screw shaft 21 is transmitted to the valve member 42 through the connecting member 26 and reacts against the restoring force of the compressed spring 45, which results in the axial displacement of the valve member 42, which forces the valve into an open position. Only if the plunger is extending or retracting relative to the cylinder at a speed that is greater than the travel speed of the internally threaded member down or up the screw shaft and relative to the cylinder, the restoring force of the spring 45 exceeds the axial thrust of the internally threaded member and the valve is closing. The resilient member 45, e.g. spring, held in place by a retention plate 46, and an adjustment screw 47 is contacting the retention plate 46 and compresses the resilient member 45 to obtain the above-mentioned bias. A further adjustment screw 48 allows a more accurate axial positioning of the sleeve 41

in the casing 40, which has the advantage that length variations of the connecting member 26 can be compensated.

[0052] Although the valve 70 is located in the plunger base 17 in the present embodiment, this is only one possible way of accommodating the valve in the fluid pressure actuator. The valve 70 with double way action possibility shown in FIG. 7, or an equivalently operating valve, may also be located in the plunger inner sidewall 18b or in the drive nut member 20. Accommodating the valve 70 in the plunger base 17 has the advantage that the valve is easier to access, e.g. for repair or replacement, or for readjustment of the screws 47, 48.

[0053] While the invention has been illustrated and described in detail in the drawings and foregoing description, such illustration and description are to be considered illustrative or exemplary and not restrictive. The foregoing description details certain embodiments of the invention. It will be appreciated, however, that no matter how detailed the foregoing appears in text, the invention may be practiced in many ways. Other variations to the disclosed embodiments can be understood and effected by those skilled in the art in practicing the claimed invention, from a study of the drawings, the disclosure and the appended claims. Any reference signs in the claims should not be construed as limiting the scope.

Claims

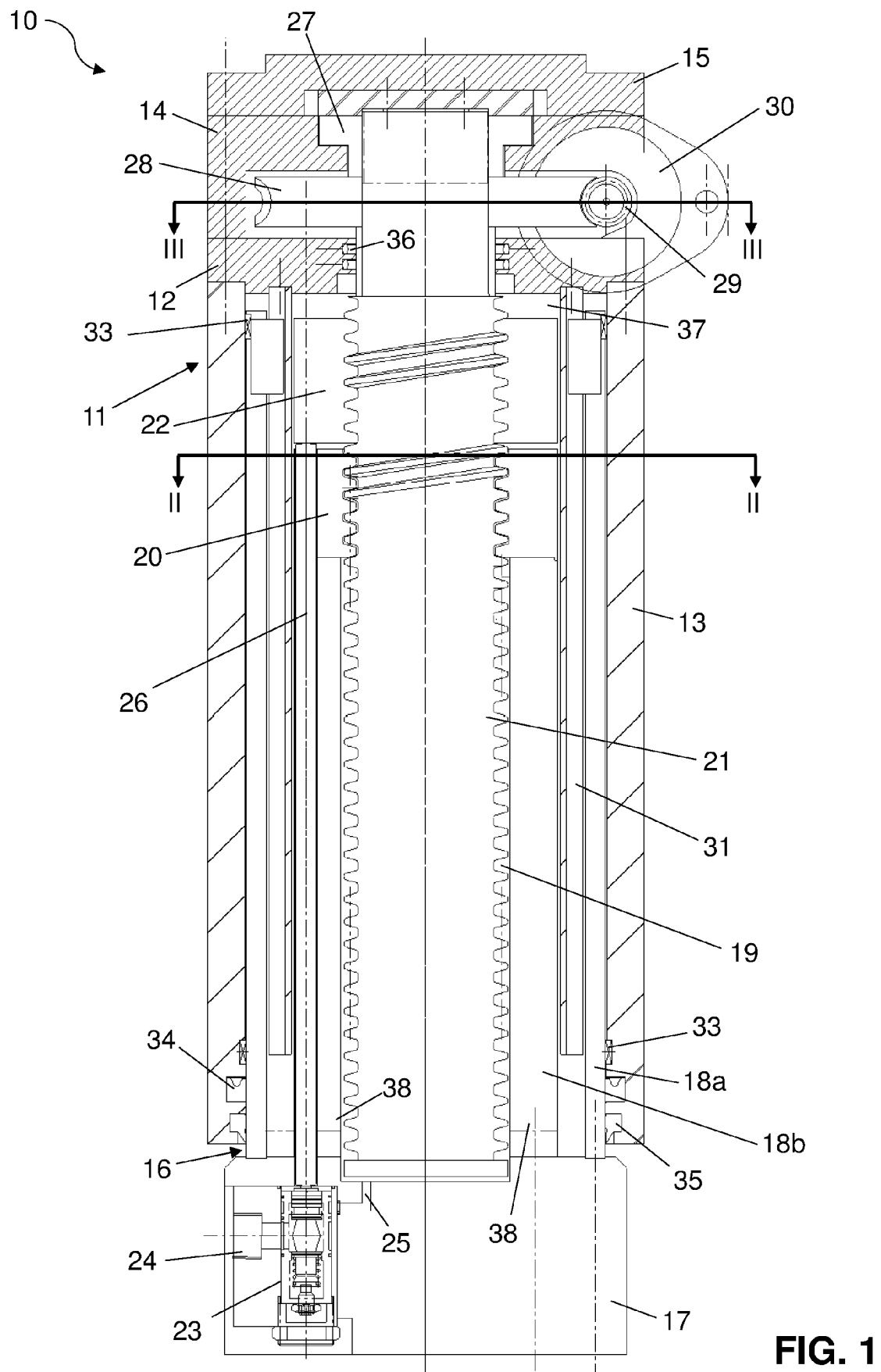
1. A fluid pressure actuator (10) comprising:

a cylinder (11) and a plunger (16) slidably mounted in the cylinder and operable by application of a pressurized fluid to extend axially relative to the cylinder,
a closed-loop speed control system for adjusting an extension speed of the plunger while moving, comprising:

a screw shaft (21) rotatably supported by the cylinder and extending axially in a hollow interior (19) of the plunger,
an internally threaded member (22) disposed in the cylinder and threadably receiving the screw shaft in close fitting relation thereto, the internally threaded member being restrained from rotation such that a rotary motion of the screw shaft translates into an axial motion of the internally threaded member relative to the cylinder,
a proportional valve (23) rigidly connected to the plunger for adjusting a volume flow rate of the pressurized fluid, when applied to operate the plunger,

wherein the internally threaded member is operatively coupled to the valve such that a cross-

- sectional area of a fluid passageway through the valve is reduced or increased in order to compensate for a difference in speed magnitudes of the moving plunger and the oppositely moving internally threaded member in respect of the cylinder.
2. A fluid pressure actuator according to claim 1, wherein an interior surface of the plunger, delimiting the hollow interior thereof, comprises a threaded portion for threadably receiving the screw shaft in loose fitting relation thereto, for supporting the cylinder in the absence of the pressurized fluid and for converting a rotary motion of the screw shaft into an axially directed extension of the plunger relative to the cylinder when the pressurized fluid is applied, wherein an axial backlash associated with the loose fitting relation is greater than a full stroke associated with the proportional valve.
 3. A fluid pressure actuator according to any claim 2, wherein the full stroke associated with the proportional valve, between a completely closed position and a fully open position of the valve, ranges between 70 % and 90 % of the axial backlash.
 4. A fluid pressure actuator according to claim 2 or 3, wherein the plunger comprises a base member (17) with at least one circumferentially projecting sidewall (18), and wherein the threaded portion on the plunger interior surface is provided by a nut (20) rigidly connected to said projecting sidewall, and/or wherein the internally threaded member (22) is a nut.
 5. A fluid pressure actuator according to claim 4, wherein the base member of the plunger has a circumferentially projecting outer sidewall (18a) and inner sidewall (18b), separated by an annular gap, and wherein rotation hindering means for preventing a rotation of the cylinder relative to the plunger are arranged in said annular gap.
 6. A fluid pressure actuator according to any of the preceding claims, wherein the proportional valve comprises a sleeve (41) and a valve member (42) slidably supported in the sleeve, and a connecting member (26) is extending between the internally threaded member and the valve member for operatively coupling the internally threaded member to the valve.
 7. A fluid pressure actuator according to claim 6, wherein the proportional valve further comprises a resilient biasing member (45) for urging the valve member into physical contact with the connecting member.
 8. A fluid pressure actuator according to any of the preceding claims, further comprising a worm drive for driving a rotary motion of the screw shaft, the worm drive comprising a gear (28) connected to the screw shaft and a worm (29) meshing with the gear.
 9. A fluid pressure actuator according to claim 8, further comprising a motor (30) having a motor shaft coupled to the worm of the worm drive.
 10. A fluid pressure actuator according to claim 9, wherein the worm drive and the motor are located at a base member (17b) of the plunger.
 11. A fluid pressure actuator according to any of the preceding claims, wherein an end portion of the screw shaft is hollow and receiving an elongated member (65) with longitudinally extending grooves on an outer surface thereof, said end portion of the screw shaft further comprising an annular disc (67) with a plurality of teeth arranged on an inner rim thereof, the disc teeth operatively engaging with the grooves of the elongated member.
 12. A fluid pressure actuator according to any of the preceding claims, wherein a load-bearing member (15) with a flat outer surface for receiving a load is provided at one end of the cylinder.
 13. A fluid pressure actuator according to any of the preceding claims, wherein the plunger further comprises a fluid channel (38) fluidly connecting the hollow interior (19) of the plunger with a fluid pressure chamber (37) extending between the internally threaded member (22) and an end cap (12) of the cylinder.
 14. A fluid pressure actuator according to any of the preceding claims, further comprising at least one elongated member rigidly connected to the cylinder and extending vertically in a sidewall bore of the plunger, for preventing the cylinder from rotating relative to the plunger.
 15. A lifting system for heavy loads comprising at least one fluid pressure actuator according to any of the preceding claims, a pump for supplying pressurized fluid to the at least one fluid pressure actuator, and driving means for controlling the rotary motion of the screw shaft of each fluid pressure actuator.



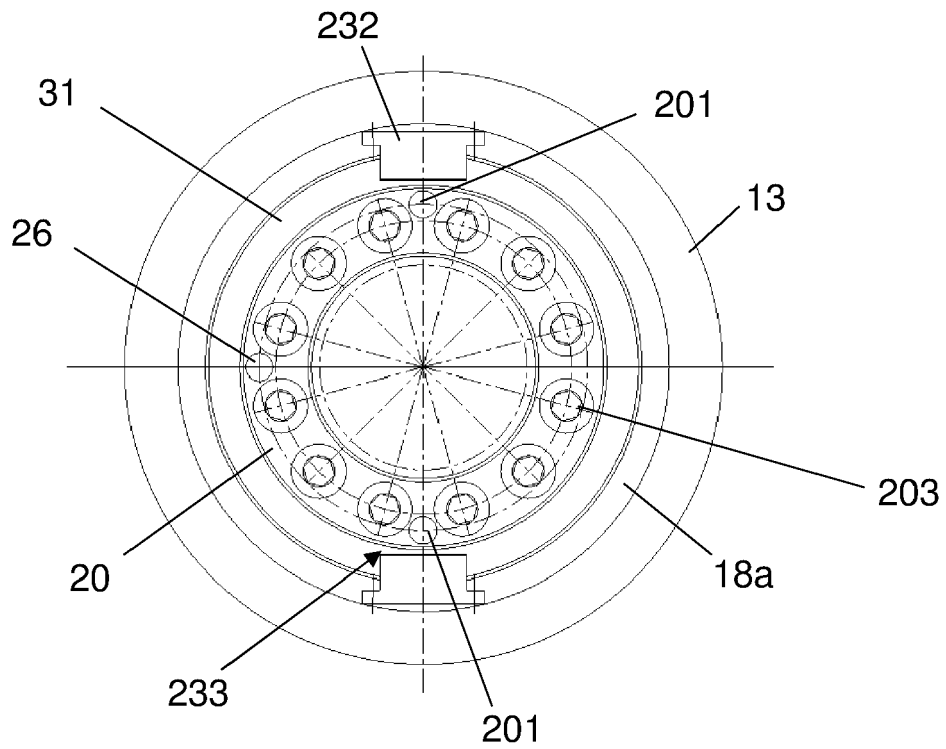


FIG. 2

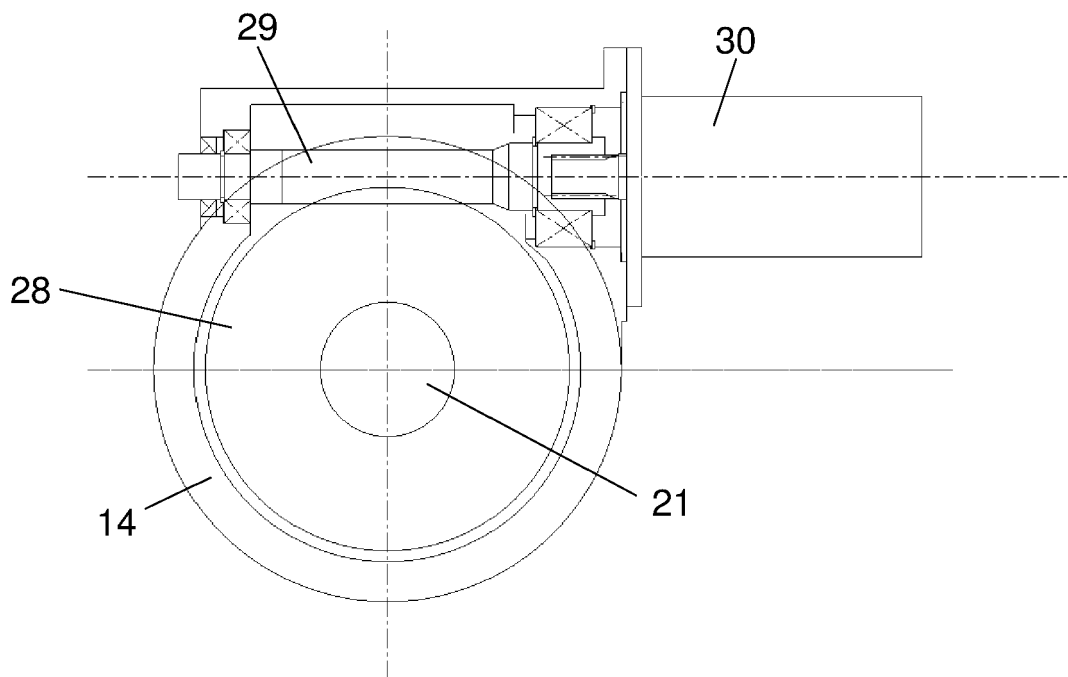


FIG. 3

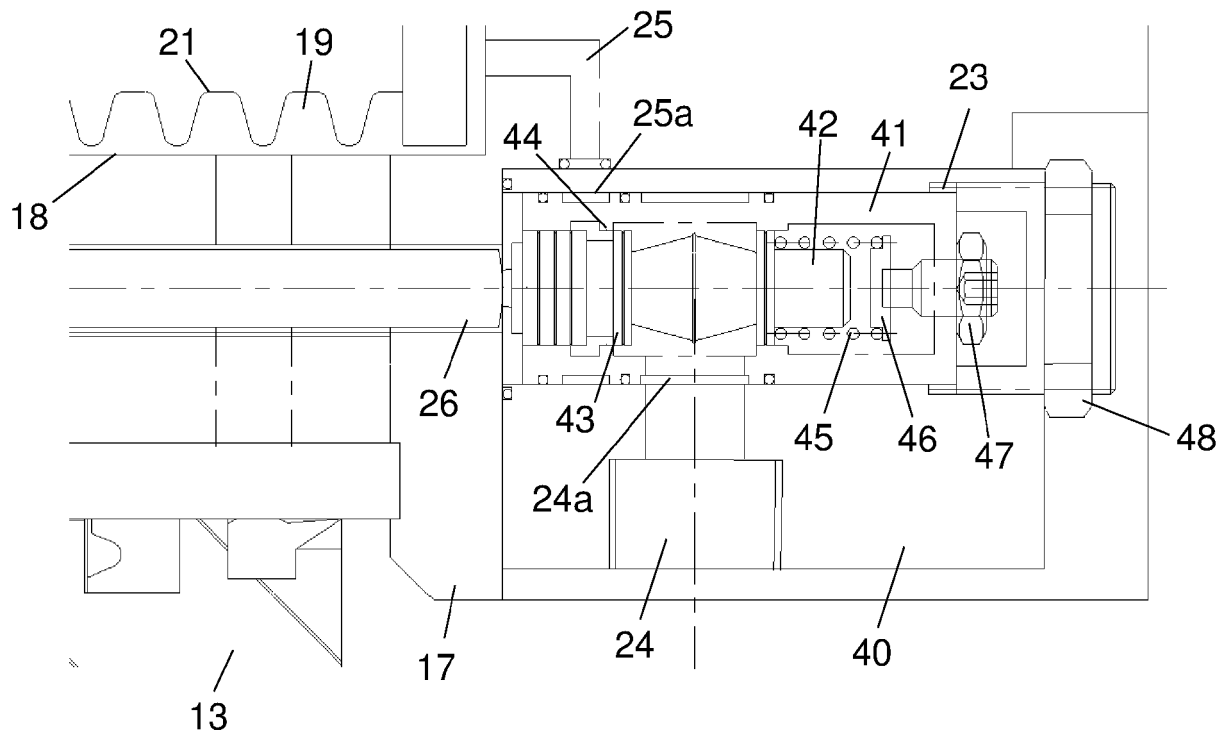


FIG. 4

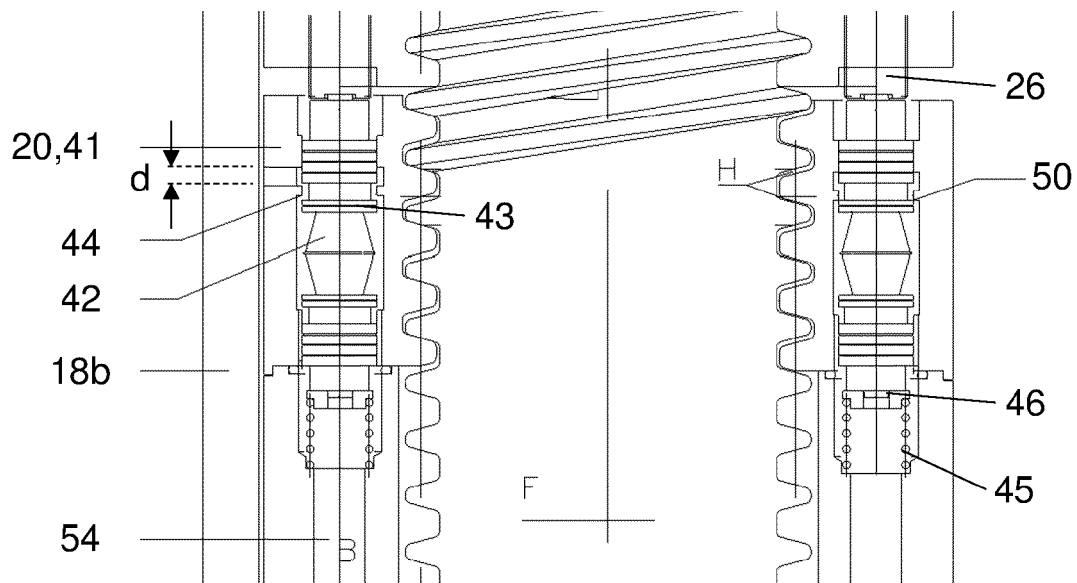


FIG. 5

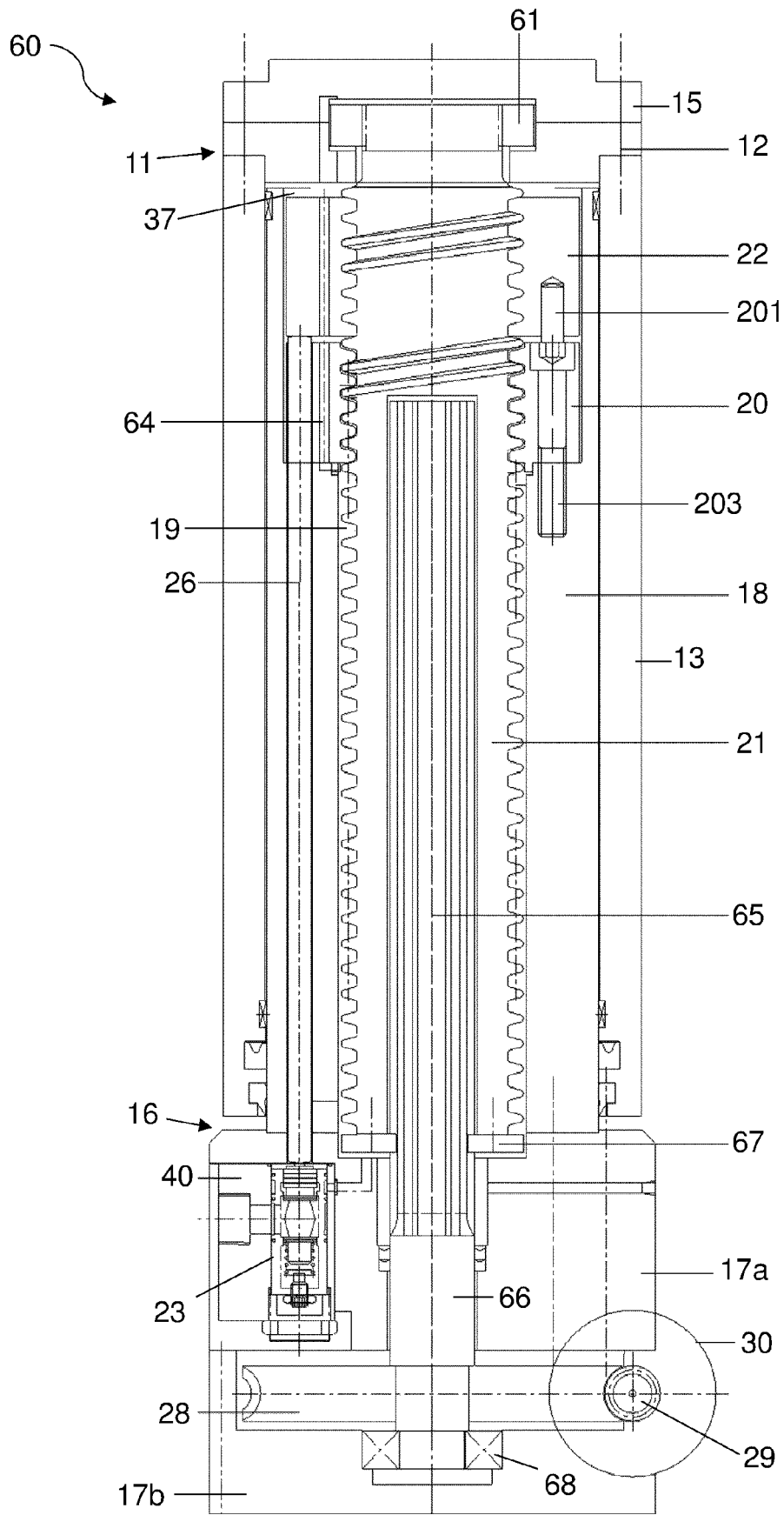


FIG. 6

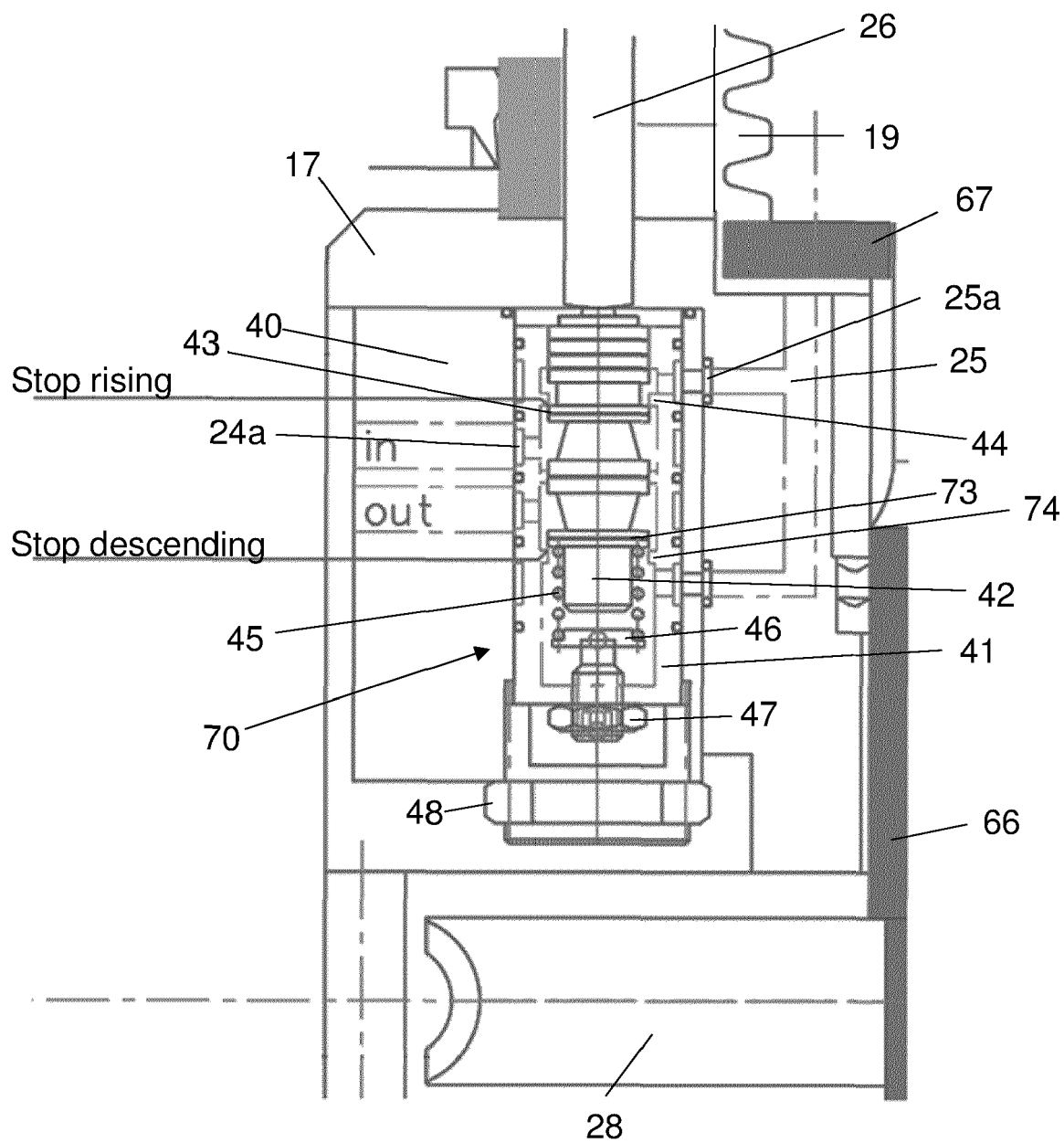


FIG. 7



EUROPEAN SEARCH REPORT

Application Number
EP 20 19 9455

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DOCUMENTS CONSIDERED TO BE RELEVANT			
Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (IPC)
A	DE 23 33 098 A1 (KRAEMER MANFRED DIPL-ING) 16 January 1975 (1975-01-16) * paragraphs [0014], [0015], [0024], [0030], [0039]; figure 1; examples 18,110,17,12,132 * -----	1-15	INV. F15B11/05 F15B15/08 F15B15/14 F15B15/20 F15B15/22 F15B9/09
			TECHNICAL FIELDS SEARCHED (IPC)
			F15B
The present search report has been drawn up for all claims			
Place of search Munich		Date of completion of the search 11 March 2021	Examiner Deligiannidis, N
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EPO FORM 1503 03.82 (P04C01)

11-03-2021

Patent document cited in search report	Publication date	Patent family member(s)	Publication date
DE 2333098	A1	16-01-1975	NONE

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For more details about this annex : see Official Journal of the European Patent Office, No. 12/82

REFERENCES CITED IN THE DESCRIPTION

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