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(54) Fluid-operated, programmable, linear actuator.

(57) A fluid operated actuator comprising a cylinder (10) a piston (17) reciprocable in said cylinder (10) a screw shaft (19) rotatably supported in the cylinder (10), said screw shaft (19) axially extending through and being threadly engaged by said piston (17) for rotation during relative axial movement between said piston (17) and said cylinder (10); a centrifugally controlled and fluid actuated braking device (31, 32; 38) is provided for retaining said screw shaft (19) from rotation to lock the relative movement between the piston (17) and the cylinder (10) of the actuator.

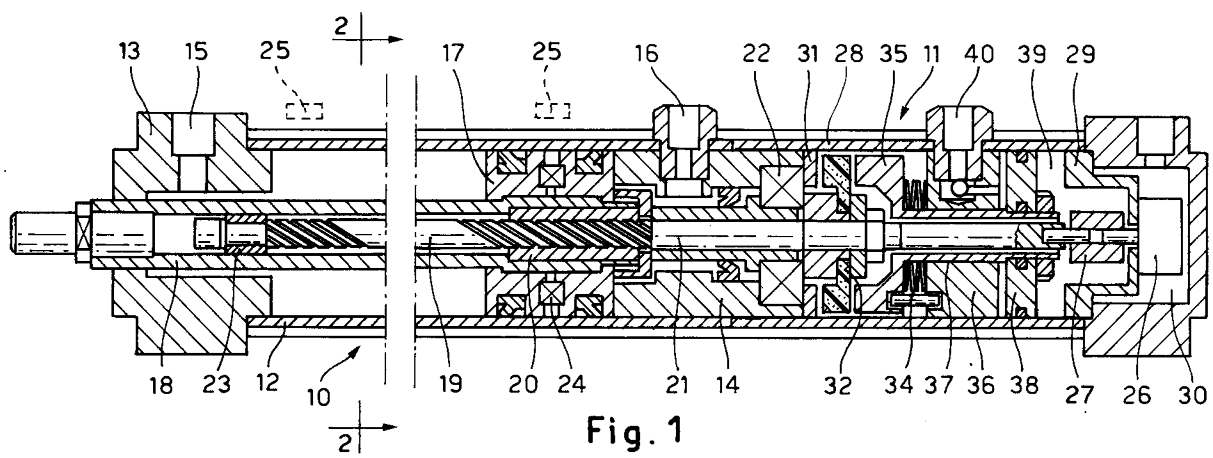

Fig. 1

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The present invention relates to a fluid operated linear actuator comprising a cylinder and a reciprocable piston within said cylinder, and in particular is directed to a pneumatic or hydraulic actuator comprising a centrifugally actuated control means by which a extremely precise control of the stroke and positioning of the piston is possible.

The use of linear actuators, be they pneumatically or hydraulically operated ones, is widespread in many sectors on account of the undoubted advantages which they offer compared to electromechanical or other types of actuators, in particular on account of the high level of power or high working speeds, as well as the flexibility in use which is generally possible with such actuators.

However, in automated or robot-controlled manufacturing systems, which require the use of sophisticated machines or equipment and suitable programming of the working cycles, linear actuators are used to little advantage and are unsuited owing to their insufficient working flexibility and limited positioning stability. For example, the high working speed of a pneumatic linear actuator is offset by a limited degree of precision or low resolution as regards positioning and the difficulty of programming operational cycles, while maintaining close control of the operational parameters, in particular the obtained position.

Therefore, fluid pressure operated actuators of the type referred above have included various types of braking systems for positioning and preventing relative movement between the piston and the cylinder; some examples of said actuators are disclosed in many patents or published patent applications such as: DE-A- 2034826, GB-A- 683633, GB-A- 2154282, US-A- 2804053 and US-A-2801615. In particular, from GB-A- 683633 and GB-A- 2154282 fluid pressure operated actuators are known in which a reciprocable piston member is slidably movable within a cylinder and is threadably, engaged with a screw shaft axially extending through said piston and said cylinder which screw shaft is connected to a pressure operated braking device to held the piston at any desired position along the length of its stroke; in addition in the actuator of GB-A- 2154282, a signal generator is provided within the cylinder and is connected to the screw shaft to provide a required degree of linear accurancy in positioning the piston within the cylinder. All braking or locking systems provided in said known actuator devices make use of rigid friction members which are to be kept sufficiently spaced apart to allow rotational movement of the screw shaft causing a retarded braking action and consequently an inaccurate positioning of the piston in the cylinder.

It has also been proposed to use an electromagnetic braking device with a cylinder, by operativally connecting it to a hollow piston by a ball nut-screw and a screw shaft which actuates an encoder controlled by a microprocessor. This solution, although it allows flexibility of use and control of the speed and position of the piston, nevertheless has considerable limitations and drawbacks resulting from the constructional features of the cylinder and the braking system. In particular the use of a ball screw in combination with an electromagnetic brake does not allow a high degree of positional control owing to the poor resolution which can be achieved. The braking system is, moreover, extremely expensive, when compared to the cost of the cylinder, and does not allow any miniaturization or any substantial reduction in the dimensions of the overall device without adversely affecting the positional control of the piston.

An object of the present invention is to provide a fluid operated linear actuator of the kind mentioned above, provided with a braking system which can be strictly controlled and which allows an extremely high degree of positioning accuracy, with a resolution of the order of hundreths, not possible with other known systems; this allows to more appropriately use the intrinsic features of fluid operated actuators.

A further object of the present invention is to provide a pneumatic or hydraulic cylinder provided with a centrifugally controlled braking device, which enables the overall dimensions to be considerably reduced while maintaining a compact structure and a high degree of stability of the actuator as regards positioning and control of the stroke, at comparatively low cost.

A further object of the present invention is to provide a linear actuator which can be programmed and controlled by an electronic processing unit and designed to allow operation at different speeds and stopping in short time, so as to considerably reduce the downtime and the duration of the working cycle, while maintaining a high degree of reliability.

A further object of the present invention is to provide a pneumatic or hydraulic linear actuator as referred to above, by means of which it is possible to obtain a very high working thrust, while at the same time requiring a relatively small braking and stopping torque which ensure positional stability even when the actuator is under loaded conditions in a slanted or vertical plane.

These and further objects of the invention can be achieved by means of a pressure fluid operated linear actuator comprising the characteristic features of claim 1.

The possibility of connecting an encoder to the screw shaft of the brake or of using, by way of an alternative, magnetically actuated limit switches or proximity switches, enables the cylinder to be associated with electronic or electric control circuits which can also be housed in the same actuator.

The actuator can be realized equally well with a normally active brake or with a passive brake, whilst maintaining all the characteristic features and fundamental principle of the invention; it is important, for the purposes of operational homogeneity, to actuate the brake using the same fluid operating the piston.

Some embodiments of linear actuators according to the invention will be described hereinbelow, by way of example, with reference to the accompanying drawings in which:

Fig. 1 is a longitudinal sectional view of a first embodiment of a linear actuator provided with a passive brake;

Fig. 2 is an enlarged cross-sectional view along the line 2-2 of figure 1;

Fig. 3 is an enlarged detail of Fig. 1;

Fig. 4 is a longitudinal sectional view through an actuator according to a second embodiment of the invention, provided with an active brake;

Fig. 5 shows a first variation of the braking device;

Fig. 6 shows a second variation of the braking device;

Fig. 7 shows the device in figure 1, with a self-aligning nut screw.

With reference to figures 1 to 3, we shall describe a first embodiment of the linear actuator according to the invention.

As shown in figure 1, the linear actuator comprises a cylinder 10 provided with a braking device 11 provided in a rear extension 28 of the cylinder, which braking device is operationally connected by means of a worm-screw system to a piston 17 reciprocable within the cylinder 10.

The cylinder 10 comprises a barrel 12 closed by a front head-piece 13 and by an intermediate head-piece 14. 15 and 16 in figure 1 denote, respectively, the inlet and/or outlet ports for a pressurized fluid, for example compressed air, through the two head-pieces of the cylinder.

Inside the cylinder 10 slides, a piston member 17 having a hollow piston rod 18 axially extending within the cylinder, which rod 18 is guided and protrudes outwardly from the front head-piece 13 of the cylinder. The piston 17 is prevented from rotating, providing means to prevent rotation during sliding movement of the piston 17, for example by giving the said piston 17 and the barrel 12 of the cylinder a polygonal cross-section, or by providing rods or guiding means for the piston inside or outside the cylinder.

The piston 17 and the piston rod 18 are axially hollow so as to allow the passage of a screw shaft 19 which is threadly engaged by a nut screw 20 fixed inside the piston 17.

The screw shaft 19 is connected or forms part of a control shaft 21 which extends through the intermediate head piece 14 of the cylinder, said control shaft 21 being connected to the braking device 11 described below.

The screw shaft 19 and the control shaft 21 for entraining the brake in rotation are rotatably supported, by means a ball bearing 22 in the intermediate head-piece 14 of the cylinder and, respectively, by means of a bush 23 at the front end of the screw shaft 19; the bush 23, in addition to support the screw shaft 19, also serves to guide the rod 18 of the cylinder during its axial movement.

According to a characteristic feature of the invention, the mounting of the screw shaft 19 into the nut screw 20 is of the direct type, that is with a long-pitch thread so as to allow high speed of travel for the piston and high actuating power. The choice of the screw pitch and its diameter may vary in each case; in general they must be such as to allow a high reduction in the torque exerted on the braking device 11, whilst ensuring a high degree of stoppage stability for the piston 17 and a brake with considerably reduced dimensions.

Optionally, the piston 17 may comprise permanent magnets 24 which, during the reciprocating movement, actuate switch devices or magnetic sensors 25, the position of which along the cylinder 10 can be suitably adjusted; these sensors supply signals indicating the position of the piston 17 to an electric or electronic control circuit, not shown.

In place of or in combination with the magnetic switch devices 25, the pneumatic actuator may comprise a signal generator such as an encoder 26 designed to emit control signals for controlling the position of the piston, within the cylinder, which signals are sent to an electronic control unit, not shown. In particular, the encoder 26 is connected, by means of a coupling 27, to the control shaft 21 and screw shaft 19 which operate the braking device 11 so as to provide a direct relationship between the position of the piston 17 within the cylinder and the control signals emitted by the encoder 26.

The braking device 11 according to the present invention is a braking device which is controlled centrifugally and actuated by the same pressurized fluid which causes travel of the piston; the braking device 11 may be equally well of the active type or passive type, as described below.

The braking device 11 is provided inside an extension 28 of the barrel 12 of the cylinder, suitably fixed at the opposite end of the intermediate head-piece 14 or integral with said barrel 12; a third head-piece 29 defining the housing 30 for the encoder 26 closes the brake housing at the rear.

The braking device 11, in the form shown in

Fig. 1, essentially comprises a first fixed braking disc 31, connected to the intermediate head-piece 14 of the cylinder, as well as a second rotating braking disc 32 connected so as to rotate with the control shaft 21 of the screw shaft 19.

This second braking disc 32, as shown in greater detail in Fig. 3, is made of elastically yieldable material , for example elastomeric material or the like, and is provided with a laterally protruding annular braking rim 33 facing towards the fixed braking disc 31. The annular rim 33 may be formed or provided with a suitable layer of frictional material having a high coefficient of friction, designed to cooperate with the fixed disc 31 so as to brake the sliding movement of the piston 17 and to retain the same steady in any desired position of its stroke.

The use of elastically flexible or yieldable material and the special shape of the rotating braking disc 32, owing to the presence of the peripheral braking rim 33 which locally displaces the centre of gravity to one side, towards the fixed braking disc 31, make it possible to use positively the centrifugal forces which are generated on rotation of the braking disc 32 so as to move away the braking rim 33 and keep it disengaged from the fixed disc 31 during actuation of the cylinder at the normal speed of travel of the piston, whilst maintaining a very close arrangement of the rim 33 with respect to the disc 31, so as to allow instantaneous operation of the brake without idle time, so that the piston 17 is rapidly locked in position; in this way it is possible to achieve a high factor of resolution for positioning of the piston, which, as previously mentioned, may be within the order of a few hundreths of a mm. This mode of operation is schematically shown in Fig. 3, where continuous lines and phantom lines denote, respectively, the two positions of engagement and disengagement between the annular rim 33 of the rotating disc 32 and the fixed braking disc 31, owing to the lateral flexion of the disc 32 caused by the moment generated by the centrifugal force as a result of the lateral disposition of the centre of gravity of the abovementioned braking rim, with respect to the plane of symmetry of the disc itself.

The centrifugal force on the rotating braking disc 32, which as a result of the displacement of the centre of gravity of the rim 33 gives rise to a torque tending to move this rim away from the fixed disc, therefore has the function of controlling the respective engaged or disengaged positions of the brake; the actual braking action or force, in the case of figure 1, however, is provided by a set of cup springs 34 or other equivalent elastic means, located between a thrust member 35 which acts against the braking disc 32 and counteracting elements 36 inside the sleeve 28.

In the example of figure 1, when the cylinder is at a standstill, the brake is normally activated in that the springs 34 act on the thrust member 35 so as to move it forwards and cause it to strongly press the rotating braking disc 32 against the fixed braking disc 31.

Deactivation of the brake occurs in a manner which is controlled and coordinated with the movement of the piston 17 of the actuator 10, using the same control fluid. In fact, the thrust member 35 is connected to a hollow stem 37, coaxial with the shaft 21, which extends through the block 36 and is connected to the piston 38 of a single-acting cylinder 39 forming part of the braking device 11. The cylinder 39 is supplied with pressurized fluid through an inlet 40 so as to move the piston 38 and allow the backward movement of the thrust member 35 against the action of the counter springs 34, when the brake must be deactivated.

Operation of the actuator described, in brief, is as follows: let us assume that the piston 17 is at a standstill in the position shown in Fig. 1 and that the solenoid valves of the circuit supplying and controlling the actuator, not shown, cut off circulation of the pressurized fluid; in these conditions, the brake control cylinder 39 will be deactivated and the thrust member 35, owing to the action of the springs 34, will act on the braking disc 32 so as to push it against the fixed braking disc, thereby keeping the piston 17 in a stable position. The stable position of the piston 17 is also ensured in the case where a high axial force acts on the stem 18 of the cylinder, since this force is reduced considerably by means of the direct connection between screw shaft 19 and nut screw 20 as a result of the long pitch of the thread and the relatively small lever arm of the couple exerted by the nut screw, compared to the mean lever arm exerted by the braking device. This makes it possible to operate with comparatively small braking and locking forces and hence realize a braking device 11 of limited dimensions, such that it can be easily accommodated in the extension of the barrel of the cylinder 10. In this way it is possible to realize an actuator with extremely small overall dimensions, suitable even for cylinders of small diameter, which have not been possible in operating and braking systems known hitherto.

If, at this point, pressurized fluid is supplied to the inlet 16 of the cylinder and to the inlet 40 of the braking device, the cylinder 39 operating the brake will allow the backward movement of the thrust member 35 which will release its pressure and move away from the braking disc 32; at the same time, the piston 17 of the actuator will be made to move forward.

With the forward movement of the piston 17, the screw shaft 19 and its control shaft 21 will be made to rotate, entraining the braking disc 32 in rotation. As soon as the braking disc 32 starts to rotate rapidly, owing to the effect of the centrifugal force it will tend to straighten or to flex elastically on the rear side, moving the braking rim 33 away from the fixed braking disc 31. In this way the braking device, after being deactivated, is centrifugally controlled and kept in the disengaged position, allowing the piston 17 to travel freely in one direction since the rim 33 of the braking disc 32 is arranged slightly removed, a short distance away from the disc 31, as shown schematically in broken lines in Figure 3.

When the piston 17 approaches its stoppage position, at a predetermined point it will be made to slow down by supplying a counter-pressure in the opposite chamber of the cylinder so as to reduce rapidly the speed of travel, thereby ensuring that the maximum amount of kinetic energy of the cylinder itself is absorbed.

Since, with slowing down of the piston 17, the speed of rotation of the braking disc 32 and hence the centrifugal effect acting on the latter will be reduced considerably, the braking disc 32, as a result of its elastic action, will move the rim 33 towards the braking disc 31, exerting a slight pressure on the latter. At this point, release of the pressure in the cylinder 39 of the brake will allow the springs 34 to act on the thruster 35, moving it forwards so as to press the braking disc 32 firmly against the fixed braking disc 31. In this way, the piston 17 of the actuator will be stopped in the desired position, controlled by the encoder 26. Given the reversibility of the coupling consisting of screw shaft 19 and nut screw 20, operation of the cylinder 10 and the braking device 11 will still be in the manner illustrated above, but with reversal of the travel of the piston 17.

Figure 4 of the accompanying drawings shows and alternative solution as regards the braking device 10 which in this case allows an active action, opposite to that of the braking device 10 in the preceding example.

In figure 4, parts identical or equivalent to those in figure 1 have been denoted by the same reference numbers. The example in figure 4 differs from that shown in figure 1 in that the rotating braking disc 32 is in the form of a piston member having an annular braking rim on one side and a peripheral lip on the opposite side, which piston member is actuated, during activation of the brake, by the pressurized fluid in place of the elastic means or the counter springs 34 previously mentioned. Therefore, in the case of figure 4, the rotating braking disc 32 is directly designed as means for activating the brake, while centrifugal control of the braking disc 32 is still possible in the deactivated condition of the brake during travel of the piston 17 in either direction. Therefore, in the case of figure 4 also, the forward and backward movement of the piston 17, by means of the worm nut-screw coupling 19, 20, will cause rotation of the control shaft 21 and, therefore, when there is no pressure in the cylinder 19, will allow the disc 32 to move away from the disc 31 as a result of the centrifugal force exerted by rotation of the disc 32 itself. In all cases, a pneumatic or hydraulic actuator is achieved, comprising a centrifugally controlled brake means which is operated, during activation or deactivation, by the same pressurized fluid, so as to ensure a precise stopping action and stable positioning of the piston 17 in any programmable position controlled by means of the encoder 26, while observing very rigorous positioning tolerances of the order of one hundreths of a millimetre. In this way, extreme reliability of operation and stable positioning, not possible with the currently known linear actuators, is ensured. The actual arrangement of the braking device, in addition to achieving the aforementioned advantages, owing to the considerable reduction in the overall dimensions, allows the braking device to be applied even to cylinders of considerably small dimensions. In this way a programmable linear actuator is achieved, which is able to ensure extremely reliable operation and a high degree of positioning accuracy, thus making it extremely flexible and utilizable in several application fields as well as sophisticated technologies. Finally, as a result of the homogeneous nature of the energy sources for the cylinder and the braking device, it is possible to combine features which are difficult to reconcile together in the same actuator, such as fast working speed, efficient braking action and high positioning accuracy.

Figures 5 and 6 show some possible variations of the movable braking disc 32. In the preceding examples, the braking rim 33 protrudes on the same side of the fixed braking ring or disc 31, or equivalent braking element, so as to obtain lateral displacement of the centre of gravity of the abovementioned ring which is directed towards the fixed braking element, so as to exploit the centrifugal effect in order to move one braking element away from the other. The same effect can be achieved with the solution of figure 5 or with that of figure 6.

Both in figure 5 and in figure 6, the disc 32 along the peripheral rim 33 has a braking ring 33a consisting of material with a high coefficient of friction; the braking rim 33 in these cases does not protrude from the plane of the disc; however, the higher weight of the braking ring 33a is such as to cause a forward displacement of the centre of gravity of the entire braking rim, therefore still allowing the centrifugal effect to be used to advantage in the manner previously mentioned; this centrifugal effect which detaches the movable disc from the fixed disc can be accentuated or improved, for example designed so as to be slightly conical with respect to the disc 32, as shown in figure 6.

Figure 7 of the drawings shows a further characteristic feature of the actuator according to the present invention, which allows free orientation and displacement of the piston 17 and its hollow stem 18 with respect to the screw shaft 19 and the nut screw 20; this solution proves to be particularly useful for pneumatic actuators where the piston has a small degree of play with respect to the barrel of the cylinder, which, in the event of unbalanced loads, can cause slight axial displacement between worm screw and hollow stem of the piston.

Therefore, unlike the preceding cases where a fixed connection between hollow piston 17 and screw nut 20 was used, in the case of figure 7 the nut screw is supported by the piston 17, in a floating and self-aligning manner, so as to allow both a different angular orientation and a relative radial displacement, while preventing any displacement between the two parts in the axial direction so as not to adversely affect the positioning control of the said piston.

The above was obtained by providing sufficient radial play between the nut screw 20 and the hollow piston 17, supporting this screw nut by means of two opposing self-aligning bearings arranged on the two sides of a radial flange 42. The nut screw 20 is prevented from rotating with respect to the piston 17, for example by means of diametrically opposite pins 43 slidably inserted in respective holes 44 in the nut screw 20 and in an elongated hole or eyelet 45 in the piston, oriented in the axial direction of the actuator.

Unlike conventional self-aligning bearings, which allow a different angular arrangement of two axes, the use of self-aligning bearings in which the races of the balls consist of an annular surfaces arranged in a radial plane, and the corresponding possibility for the nut screw 20 to perform a translational movement parallel to itself, without any displacement in the axial direction, advantageously also allow a skew arrangement of the screw shaft and the associated nut screw, with respect to the piston of the actuator.

Another possibility for angular adaptation of the hollow rod 18 of the piston also exists by providing this rod with a head 18' delimited by a spherical surface portion 46 inside a seat provided by a sleeve 47 which extends from one side of the hollow piston 17; a pin 48 allows small angular oscillations of the rod 18 with respect to the piston 17, preventing it from rotating.

It can be understood, therefore, that the above explanations and illustrations in the accompanying drawings have been given purely by way of example and that other modifications or variations may be made both to the pneumatic cylinder of the actuator and to the braking or control device without thereby departing from the present invention.

Claims

1. A fluid operated linear actuator, comprising a cylinder (10) ,a piston member (17) reciprocable in said cylinder (10) and a screw shaft (19) rotatably extending in said cylinder (10), said screw shaft (19) threadly engaging said piston member (17) for rotation during relative axial movement between said piston member (17) and said cylinder (10) and a braking device (11) operationally connected to said screw shaft (19), said braking device comprising first frictional member (31) connected to said cylinder (10) and second frictional member (32) rotatably connected to said screw shaft (19), characterized in that said braking device is in the form of a centrifugally controlled and fluid operated braking means (33, 38; 39), said braking means (33, 38;39) comprising a braking annular member (33) rotatably connected to said screw shaft (19) by an elastically flexible connecting member, the centre of gravity of said annular braking member (33) being displaced to one side and towards said first braking member (31).

2. Actuator according to Claim 1, characterized in that said annular braking member (33) and said connecting member are in the form of a disc element (32).

3. Actuator according to Claims 1 or 2, characterized in that said annular braking member (33) protrudes from one side.

4. Actuator according to Claim 2, characterized in that said disc element (32) is in the form of a flat disc.

5. Actuator according to Claim 2, characterized in that said disc element (32) is in the form of a conical disc.

6. Actuator according to Claim 1, characterized in that said fluid operated braking means act in the same direction of activation of the brake.

7. Actuator according to Claim 1, characterized in that said fluid operated braking means act in the opposite direction of activation of the brake and biasing means (34) being provided be-

tween a shoulder (36) and a fluid actuate thrust member (35).

8. Actuator according to Claim 7, characterized in that said fluid operated braking means comprises a thrust member (35) on one side of said second braking member (32), said thrust member (35) being connected to the piston (38) of a control cylinder (39) acting in opposite direction to said biasing means (34).

9. Actuator according to Claim 1, characterized in that said screw shaft (19) is rotatably and slidably supported inside a hollow rod (18) of the piston (17).

10. Actuator according to Claim 1, characterized in that said braking means are integral with the piston member (39) of a control cylinder (11) for actuating the braking means.

11. Actuator according to Claim 1, characterized in that a signal emitter (26) for controlling the movement of the piston member (17) is connected to the screw shaft (19) of the braking device.

12. Actuator according to Claim 1, characterized in that said piston member (17) comprises magnetic means (24) and in that magnetic sensing means (25) are provided along said cylinder (10) for detecting the positions to stop said piston member (17).

13. Actuator according to Claims 1 and 11, characterized in that said braking means (33, 38; 39) and said signal emitter (26) are provided in a rear extension of the cylinder (10).

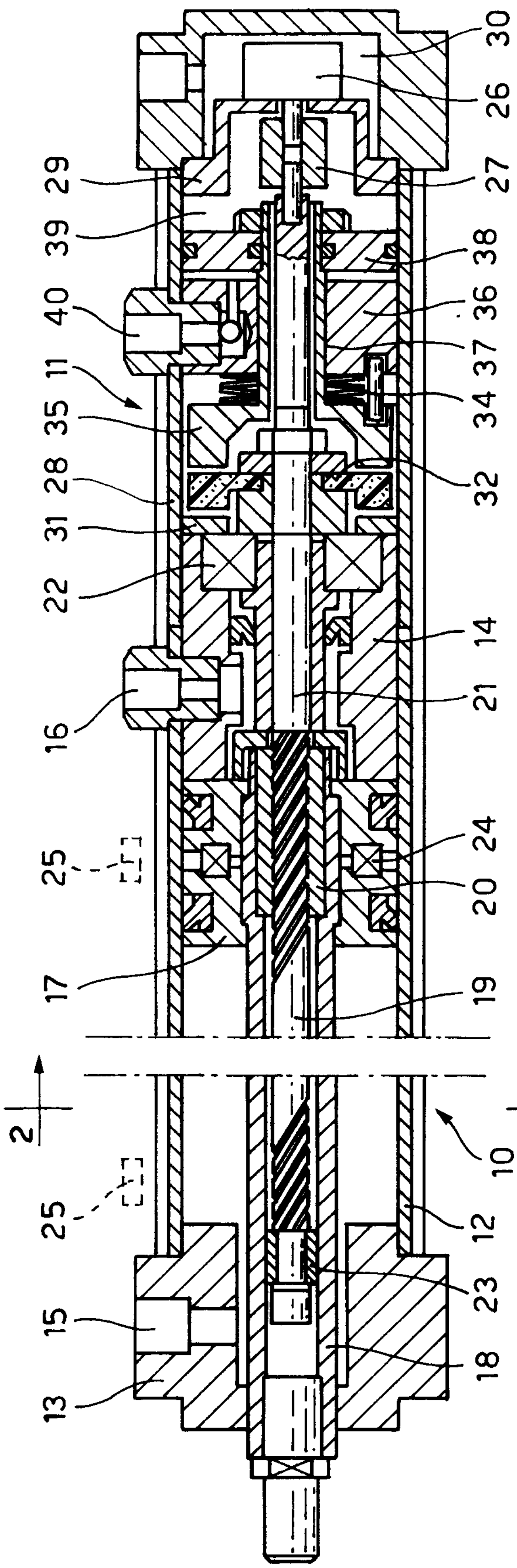

Fig. 1

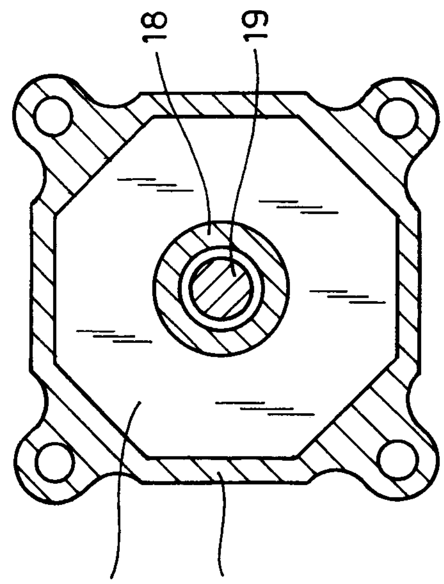

Fig. 2

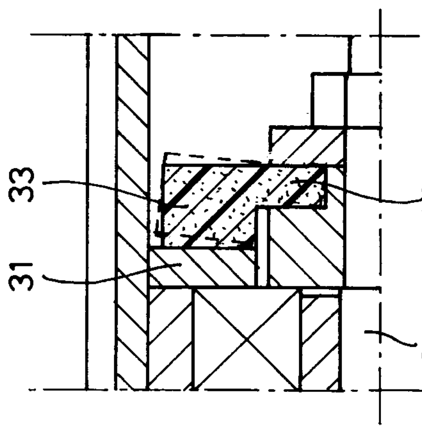

Fig. 3

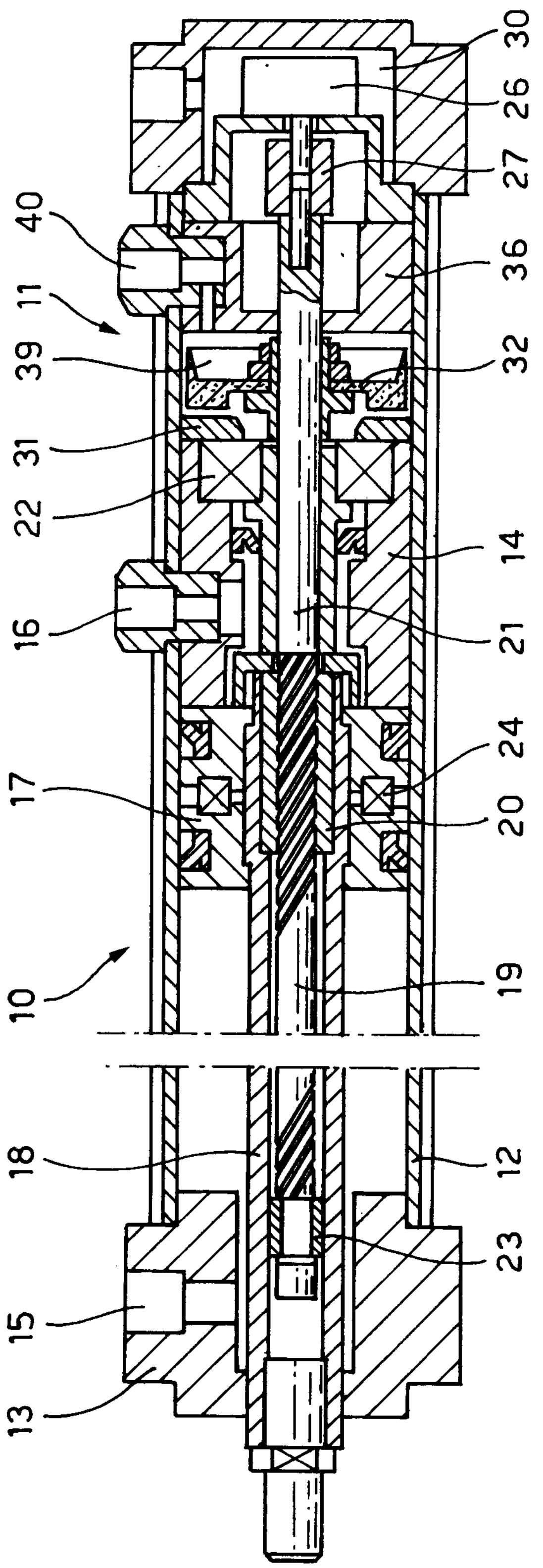

Fig. 4

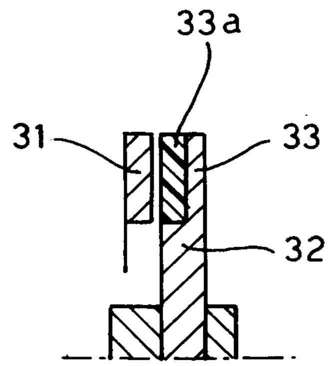

FIG. 5

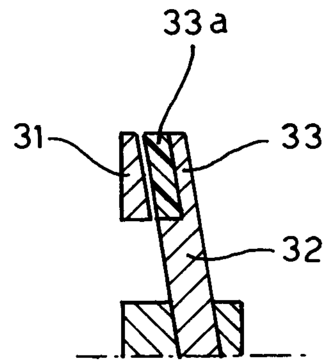

FIG. 6

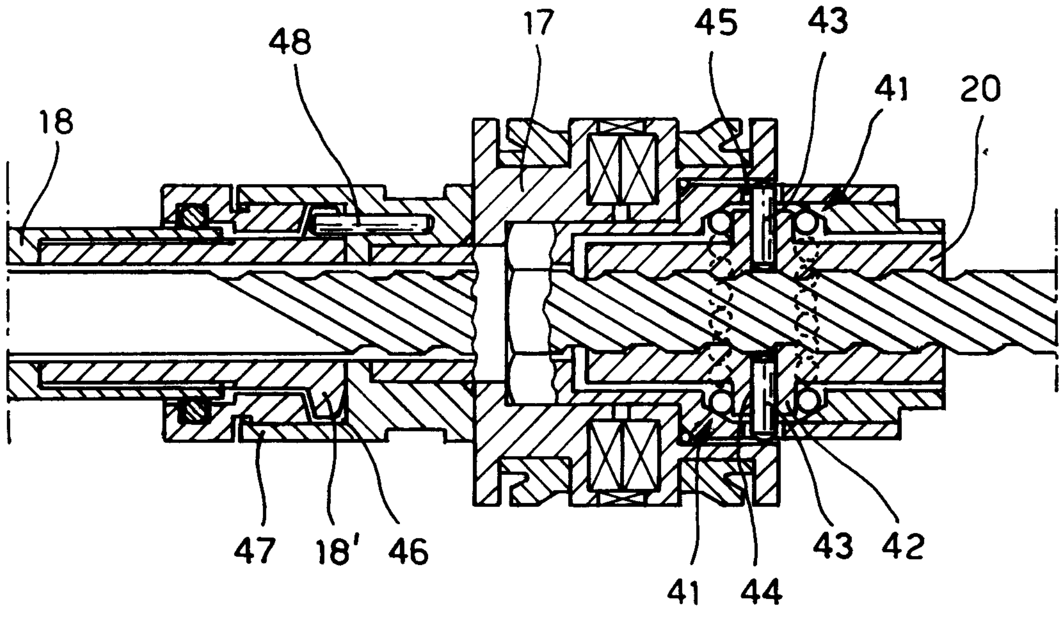

FIG. 7

European Patent Office

EUROPEAN SEARCH REPORT

Application Number

EP 91 10 7470

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int. Cl.5)
A,D	DE-A-2 034 826 (PNEUMO DYNAMICS) * Page 13, lines 11-20 * - - -	1	F 15 B 15/26
A	US-A-2 879 746 (GEYER) - - -		
A,D	US-A-2 804 053 (GEYER) - - -		
A,D	US-A-2 801 615 (GEYER) - - -		
A,D	GB-A-2 154 282 (HCH) - - -		
A,D	GB-A-683 633 (GENERAL MOTORS) - - - - -		

TECHNICAL FIELDS SEARCHED (Int. Cl.5)
F 15 B

The present search report has been drawn up for all claims

Place of search	Date of completion of search	Examiner
The Hague	02 October 91	KNOPS J.

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
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