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(54) Title: DEVICE FOR ROCK- AND CONCRETE MACHINING
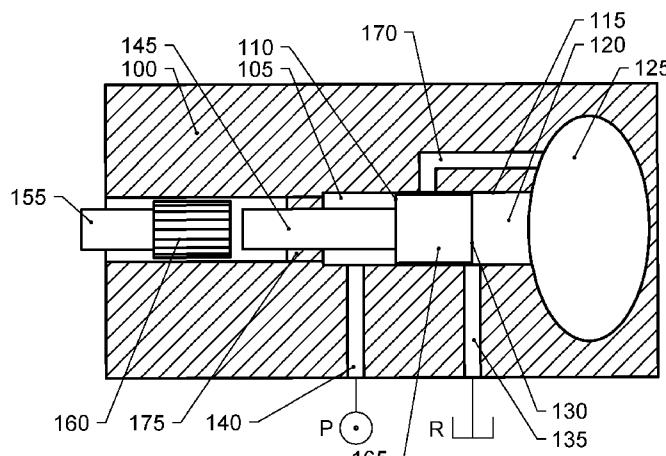
Fig. 2

(57) Abstract: The invention concerns a hydraulic striking tool for application in rock and/or concrete cutting equipment containing a machine housing (100;200) with a cylinder (115;215) with a moveably mounted piston (145;245) which during operation performs a repetitive forward and backward movement relative to the machine housing (100;200) and directly or indirectly strike a rock and/or concrete cutting tool (155;255), and where the piston (145;245) includes a driving part (165;265) which separates a first (120;220) and a second (105;221) driving chamber formed between the piston (145;245) and the machine housing (100;200) and where these driving chambers are arranged to include a pressurised working fluid during operation. The total volume V of the first and second driving chambers is inversely proportional dimensioned to the square of a for the striking tool recommended maximal pressure p, as well as proportional, by a proportionality constant k within the interval 5.3-21.0, to the product of the pistons energy E during the strike against the tool and compression module β of the working fluid.

Device for rock- and concrete machining

Technical area

The present invention concerns hydraulic impact mechanisms of the
type known as "slideless" or "valveless" to be used in equipment

5   for machining at least one of rock and concrete, and equipment for
drilling and breaking comprising such impact mechanisms.

Background

Equipment for use in rock or concrete machining is available in
variants with percussion, rotation, and percussion with simultaneous

10   rotation. It is well-known that the impact mechanisms that are
components of such equipment are driven hydraulically. A hammer
piston, mounted to move within a cylinder bore in a machine housing,
is then subject to alternating pressure such that a reciprocating
motion is achieved for the hammer piston in the cylinder bore. The

15   alternating pressure is most often obtained through a separate
switch-over valve, normally of sliding type and controlled by the
position of the hammer piston in the cylinder bore, alternately
connecting at least one of two drive chambers, formed between the
hammer piston and the cylinder bore, to a line in the machine

20   housing with driving fluid, normally hydraulic fluid, under
pressure, and to a drainage line for driving fluid in the machine
housing. In this way a periodically alternating pressure arises that
has a periodicity corresponding to the impact frequency of the
impact mechanism.

25   It is also known, and has been for more than 30 years, to
manufacture slideless hydraulic impact mechanisms, also known

sometimes as "valveless" mechanisms. Instead of having a separate
switch-over valve, the hammer pistons in valveless impact mechanisms
perform also the work of the switch-over valve by opening and
closing the supply and drainage of driving fluid under pressure

5   during the motion of the piston in the cylinder bore in a manner
that gives an alternating pressure according to the above
description in at least one of two drive chambers separated by a
driving part of the hammer piston. A precondition for thus to work
is that channels, arranged in the machine housing for the

10   pressurisation and drainage of a chamber, open out into the cylinder
bore such that the openings are separated in such a manner that
direct short-circuited connection between the supply channel and the
drainage channel does not arise at any position during the
reciprocating motion of the piston. The connection between the

15   supply channel and the drainage channel is normally present only
through the gap seal that is formed between the driving part and the
cylinder bore. Otherwise, major losses would arise, since the
driving fluid would be allowed to pass directly from the high-
pressure pump to a tank, without any useful work being carried out.

20   In order for the piston to continue its motion from the moment at
which a channel for drainage of a drive chamber is closed until the
moment at which a channel for the pressurisation of the same drive
chamber opens, or vice versa, it is required that the pressure in
the drive chamber change slowly as a consequence of a change in

25   volume. This may take place through the volume of at least one drive
chamber being made large relative to what is normal for traditional
impact mechanisms of sliding type. It is necessary that the volume
be large since the hydraulic fluid that is normally used has a low

compressibility. We define the compressibility κ as the ratio

between the relative change in volume and the change in pressure: κ

= (dV/V)/dP. It is, however, more common to use the modulus of

compressibility, β, as a measure of compressibility. This is the

inverse of the compressibility as defined above, i.e. β = dP/(dV/V).

The units of the modulus of compressibility are Pascal. The

definitions given above will be used throughout this document.

US 4 282 937 reveals a valveless hydraulic impact mechanism with two

drive chambers, where the pressure alternates in both of these

chambers. Both drive chambers have a large effective volume through

them being placed in permanent connection with volumes that lie

close to the cylinder bore. One disadvantage of the prior art

technology revealed in this way is that it has turned out to give a

surprisingly low efficiency, given that one mobile part has been

removed compared with conventional impact mechanisms with a switch-

over valve. In this document we define "efficiency", unless

otherwise stated, as the hydraulic efficiency, i.e. the impact power

of the piston divided by the power supplied to the hydraulic pump.

SU 1068591 A reveals a valveless hydraulic impact mechanism

according to a second principle, namely that of alternating pressure

in the upper drive chamber and a constant pressure in the lower,

i.e. the chamber that is closest to the connection of the tool. What

is aspired to here is improved efficiency through the introduction

of a non-linear accumulator system working directly against the

chamber in which the pressure alternates. This is shown with two

separate gas accumulators, where one of these has a high charging

pressure and the other has a low charging pressure.

One disadvantage of being compelled to introduce accumulators that act directly at a chamber where the pressure alternates at the impact frequency between full impact mechanism pressure and a low return pressure during operation is that the service interval

5   becomes shorter due to the moving parts in the accumulators being subject to heavy wear.

Purpose of the invention and its most important distinguishing features

One purpose of the present invention is to demonstrate a design of a

10   valveless hydraulic impact mechanism that offers the opportunity of improving the efficiency without at the same time reducing the service interval. This is achieved in the manner that is described in the independent claims. Further advantageous embodiments are described in the non-independent claims.

15   We define the effective volume of the drive chambers as the sum of the drive chamber volumes that have an alternating pressure during one stroke cycle, including volumes that are in continuous connection with one and the same drive chamber during a complete stroke cycle. It has proved to be the case that the effective volume

20   of the drive chambers, according to the definition given above, is of crucial significance for the efficiency of the impact mechanism with respect to valveless impact mechanisms. There are, of course, many factors that influence the efficiency, such as play and the length of gap seals, friction in bearings, etc. It is not possible,

25   however, to achieve the desired efficiency without a correctly adapted effective volume of the drive chambers, no matter how such play and bearings are designed.

Factors that influence the optimal effective volume of the drive

chambers with respect to efficiency are: the impact mechanism

pressure used, the compressibility of the driving medium and the

energy of the piston in its impact against the tool or against a

5   part that interacts with the tool. To be more precise, the effective

volume of the drive chambers is influenced in inverse proportion to

the square of the impact mechanism pressure and proportionally to

the product of the effective modulus of compressibility of the

driving medium and the energy of the hammer piston when it impacts

10   the tool or a part that interacts with the tool, such as the part

known as an "adapter".

The relationship can be expressed by the equation: $V = k * \beta * E/p^2$,

where V is the effective drive chamber volume (by which we mean the

sum of the volumes of the two drive chambers, including volumes that

15   are in continuous connection with one and the same drive chamber

during a complete stroke cycle). In the case in which alternating

pressure is present in only one of the drive chambers, the volume of

this chamber is normally totally dominating in comparison with that

of the chamber that has a constant pressure. It then becomes

20   possible to regard the effective drive chamber volume as the volume

solely of the drive chamber that has alternating pressure together

with the volume that is continuously connected to this. β in the

equation constitutes the effective modulus of compressibility of the

driving medium as it has been previously defined. If the driving

25   medium consists of several components each of them having an

individual compressibility, the effective modulus of compressibility

is calculated as the resultant ratio between the change in pressure

and the relative change in volume. Figure 3 presents values of β for

hydraulic fluids with different levels of air content. Figure 3 has
been taken from a collection of equations in hydraulic and pneumatic
engineering, and thus constitutes prior art technology. It will be
apparent to one skilled in the arts that $\beta = 1500 + 7.5p$ MPa when the
air content of the fluid is zero. In the case in which gas
accumulators are directly connected to the effective volumes, as is
described in, for example, SU 1068591 A, these are also to be
included in the calculation of effective volume. Thus, the existing
gas volume that is present in these, normally consisting of nitrogen
gas, will be included in the calculation of the effective modulus of
compressibility. It is appropriate in this case that the gas volumes
of the accumulators when the impact mechanism is in its resting
condition, i.e. the condition that normally prevails before the
impact mechanism is started, be used. The said gas accumulators here
are not to be confused with those that are normally connected to the
supply line and return line for the impact mechanism. Such
accumulators are connected to the drive chamber only intermittently,
and are thus not to be included in the calculation of the effective
volume or the effective modulus of compressibility.

Furthermore, E denotes the impact energy of the piston in its impact
with the tool or with a part that interacts with the tool. Finally,
p is the impact mechanism pressure that is used. The impact
mechanism pressure is normally between 150 and 250 bar. Finally, k
is a constant of proportionality, that it has become apparent most
suitably lies in the interval $7.0 < k < 9.5$, but where a good effect
for the efficiency can be achieved in the larger interval $6.2 < k <$
11.0 and even up to the interval $5.3-21.0$.

When the volumes have been dimensioned according to the description above, it is possible to achieve an efficiency that exceeds 75% in the case in which the effective drive chamber volumes are limited by walls of non-flexible material, i.e. when the driving medium

5   consists of pure fluid or fluid that has been mixed to a certain extent with gas while, in contrast, no gas accumulators are continuously directly connected to the drive chambers. It is possible to achieve such efficiencies without requiring extremely low play between the piston and the cylinder bore, and thus without

10   the subsequent extremely high demands on manufacturing precision needing to be used. An appropriate play may be 0.05 millimetre. This form of impact mechanism is that which gives the longest service interval of all, since so few moving parts are included.

Very much smaller effective drive chamber volumes can be achieved if

15   gas accumulators are continuously connected to the drive chambers and in this way are included in the calculation of effective volumes, as previously described. Furthermore, even higher efficiencies can be achieved in the impact mechanism if two gas accumulators with different specifications are connected to one and

20   the same drive chamber in such a manner that one is pre-charged with a high gas pressure, i.e. equal to the impact mechanism pressure or the system pressure, and one is pre-charged with a low gas pressure, normally atmospheric pressure. When the dimensioning of volumes takes place as described earlier, an efficiency that exceeds 85% can

25   be achieved with a play of the same magnitude as that previously mentioned. The service interval is increased also in this case, through the volumes not being made larger than necessary. The need for motion of the membrane of the accumulators can in this way be

reduced.

One preferred embodiment constitutes an impact mechanism, where the volume (by which we refer to the effective volume as defined above) of one of the drive chambers is much larger than that of the second drive chamber, i.e. that the volume of the second drive chamber is negligible, for example 20% or less than the volume of the first drive chamber, and where the smaller drive chamber has essentially constant pressure during the complete stroke cycle. Constant pressure in this chamber is normally achieved by the chamber being connected to a source of constant pressure during the complete stroke cycle, or at least during essentially the complete stroke cycle, most often being directly connected to the source for the system pressure or alternatively impact mechanism pressure.

Impact mechanisms of the type that has been described above can be an integrated component of equipment for the machining of at least one of rock and concrete, such as rock drills and hydraulic breakers. These machines or breakers during operation should most often be mounted onto a carrier that can comprise means for their alignment and position together with means for the feed of the drill or breaker against the rock or concrete element that is to be machined, and further, means for the control and monitoring of the process. Such a carrier may be a rock drilling rig.

Brief description of drawings

Figure 1 shows a sketch of the principle of a valveless hydraulic impact mechanism with alternating pressure in drive chambers not only on the upper surface of the piston but also on its lower

surface.

Figure 2 shows a sketch of the principle for a corresponding impact mechanism with alternating pressure on only one surface, and with constant pressure on the second.

Figure 3 shows a diagram, actually known, for the calculation of the effective modulus of compressibility for a pressure medium that consists of gas and hydraulic fluid.

Figure 4 shows an impact mechanism according to Figure 2 with the hammer piston at four different positions: A – the braking is starting at the upper position; B – the upper turning point; C – the braking is starting at the lower position; D – the lower turning point.

Detailed description of preferred embodiments

A number of designs of the invention will be described as examples below, with reference to the attached drawings. The protective scope of the invention is not to be regarded as limited to these embodiments, instead it is defined by the claims.

Figure 1 shows schematically a hydraulic impact mechanism with alternating pressure not only on the upper surface of the piston but also on its lower surface.

In a similar manner, Figure 2 and Figure 4 show an impact mechanism with constant hydraulic pressure throughout the stroke cycle on the lower surface of the piston, i.e. on that surface that is located most closely to the tool 155, 255 onto

which the hammer piston is to transfer impact energy, and with

alternating pressure during the stroke cycle on the upper

surface of the piston.

Hydraulic fluid at impact mechanism pressure is supplied to

5   the impact mechanism through supply channels 140, 240, which

pressure often lies within the interval 150-250 bar. The

system pressure, i.e. the pressure that the hydraulic pump

delivers, is often equal to the impact mechanism pressure.

The hydraulic fluid is set in connection with a hydraulic tank

10 through return channels 135, 235, in which tank the oil

normally has atmospheric pressure.

The hammer piston 145, 245 executes a reciprocating motion in

a cylinder bore 115, 215 in a machine housing 100, 200. The

hammer piston comprises a driving part 165, 265 that separates

15 a first driving area 130, 230 from a second driving area 110,

210. The pressure that acts on these driving areas causes the

piston to execute reciprocating motion during operation. The

piston is controlled radially by piston guides 175, 275. In

order to avoid pulsation in connecting lines, gas accumulators

20 180, 280 and 185, 285 may be arranged on supply channels 140,

240 and return channels 135, 235, respectively, which gas

accumulators even out rapid variations in pressure.

In order for it to be possible for the hammer piston 145, 245

to move sufficiently far into a drive chamber 120, 220, 221

25 with alternating pressure, with the aid of its kinetic energy,

after the driving part 165, 265 has closed the connection to

the return channel 135, 235, such that a connection between

the supply channel 140, 240 and the chamber 120, 220, 221 can

be opened, it is necessary that the chamber have a

sufficiently large volume that the increase in pressure in the

chamber as a consequence of the compression by the piston of

5   the volume of fluid that has now been enclosed within the

chamber is not so large that the piston reverses its direction

before a supply channel 140, 240 has been opened into the

chamber, such that the pressure can now rise to the full

impact mechanism pressure, and the piston in this way be

10   driven in the opposite direction. The drive chamber for this

purpose is connected to a working volume 125, 225, 226. Since

this connection between the drive chamber and the working

volume is maintained throughout the stroke cycle, we will

denote the sum of the volume of the drive chamber and the

15   working volume as the "effective drive chamber volume". It has

proved to be the case, as has been described earlier in this

application, that this volume is critically important to

achieving high efficiency.

A functioning design involves an effective volume of 3 litres

20   for a system pressure of 250 bar, impact energy of 200 Joules,

a hammer piston weight of 5 kg, an area of the first drive

surface 130 of 16.5 cm^2 and an area of the second drive surface

110 of 6.4 cm^2. The length of the driving part 70 mm and the

distance between the supply channel and the return channel for

25   the drive chamber 120 at their relevant connections to the

cylinder bore is 45 mm.

At an impact mechanism pressure or system pressure of 250 bar,

giving a β value, as is made clear by Figure 3, equal to 1500 + 7.5 x 25 = 1687.5 MPa. These values together with an effective volume of 3 litres and impact energy of 200 Joule give, as an example, the constant of proportionality:

5       $k = (3 \cdot 10^{-3}/200 \cdot 1687.5 \cdot 10^6) \cdot (250 \cdot 10^5)^2 = 5.55$.

The drive chamber volume and, in particular, the working volume with its large volume can be located in the machine housing in various ways.

It is advantageous that the volumes be placed symmetrically

10   around the cylinder bore.

It is further advantageous that they be placed concentrically around the cylinder bore.

It may be advantageous, as an alternative, that they be placed in the extension of the cylinder bore.

15   It is appropriate that an impact mechanism according to the principles described above be integrated in a rock drill or, alternatively, in a hydraulic breaker.

A rock drilling rig with equipment for the positioning and alignment of such a rock drill or hydraulic breaker should

20   comprise at least one rock drill or at least one hydraulic breaker according to the invention.

CLAIMS

1. A hydraulic impact mechanism for use in equipment for at least
one of rock and concrete machining comprising a machine housing
(100, 200) with a cylinder bore (115, 215), a piston (145, 245)
5   mounted to move within this bore and arranged to carry out
repetitively reciprocating motion relative to the machine housing
(100, 200) during operation and in this way to deliver impacts
directly or indirectly onto a tool (155, 255) connectable to the
equipment for machining at least one of rock and concrete, and
10   where the piston (145, 245) includes a driving part (165, 265)
that separates a first (120, 220) and a second (105, 221) drive
chamber formed between the piston (145, 245) and the machine
housing (100, 200) and where these drive chambers are arranged
such that they include during operation a driving medium under
15   pressure, and where, further, the machine housing (100, 200)
includes channels that open out into the cylinder bore (115, 215)
and that are arranged such that they include the driving medium
during operation, and that with the aid of the piston (145, 245),
during its motion in the cylinder bore (115, 215), open onto and
20   close from one of the drive chambers such that this drive chamber
acquires a periodically alternating pressure for the maintenance
of the reciprocating motion of the piston, and that positions for
the opening of the channels axially in the cylinder bore (115,
215) and for opening and closing along the extent of the piston
25   parts are adapted to maintain this drive chamber closed for the
supply or drainage of driving medium that is present in the
chamber along a distance between an opening of a first channel in
association with a first turning point of the piston (145, 245)

and an opening of a second channel in association with a second

turning point of the piston (145, 245) and that the motion of the

piston along this distance continues during the compression or

expansion of the volume of this drive chamber, where this volume

5   has been further adapted in order to achieve slow change in

pressure along the said distance, characterised in that the total

volume V of the first and second drive chamber has been

dimensioned to be inversely proportional to the square of a

maximal pressure p, recommended for the impact mechanism, and

10   further proportional, with a constant of proportionality k, that

has a value in the interval 5.3-21.0, to the product of the energy

E of the piston in the impact against the tool (155, 255) and the

modulus of compressibility β of the driving medium.

2.   The hydraulic impact mechanism according to claim 1, with the

15   constant of proportionality k in the interval 6.2 > k < 11.

3.   The hydraulic impact mechanism according to claim 1, with the

constant of proportionality k in the interval 7.0 > k < 9.5.

4.   The hydraulic impact mechanism according to any one of the

preceding claims, where the volume of one of the drive chambers is

20   much greater than the volume of the second drive chamber.

5.   The hydraulic impact mechanism according to any one of the

preceding claims, where one of the drive chambers has a constant

pressure during essentially the complete stroke cycle.

6.   The hydraulic impact mechanism according to any one of claims 1-

25   3, where the drive chambers are alternately set under pressure.

7.   The hydraulic impact mechanism according to any one of the

preceding claims, where the volumes of the chambers extend

symmetrically around the cylinder bore (115, 215).

8. The hydraulic impact mechanism according to any one of the preceding claims, where the volumes of the chambers extend concentrically around the cylinder bore (115, 215).

9. The hydraulic impact mechanism according to claim 5, where the drive chamber with alternating pressure extends in the extension of the cylinder bore.

10. A rock drill comprising impact mechanisms according to any one of the preceding claims.

11. A rock drilling rig comprising the rock drill according to claim 10.

12. A hydraulic breaker comprising impact mechanisms according to any one of claims 1-9.
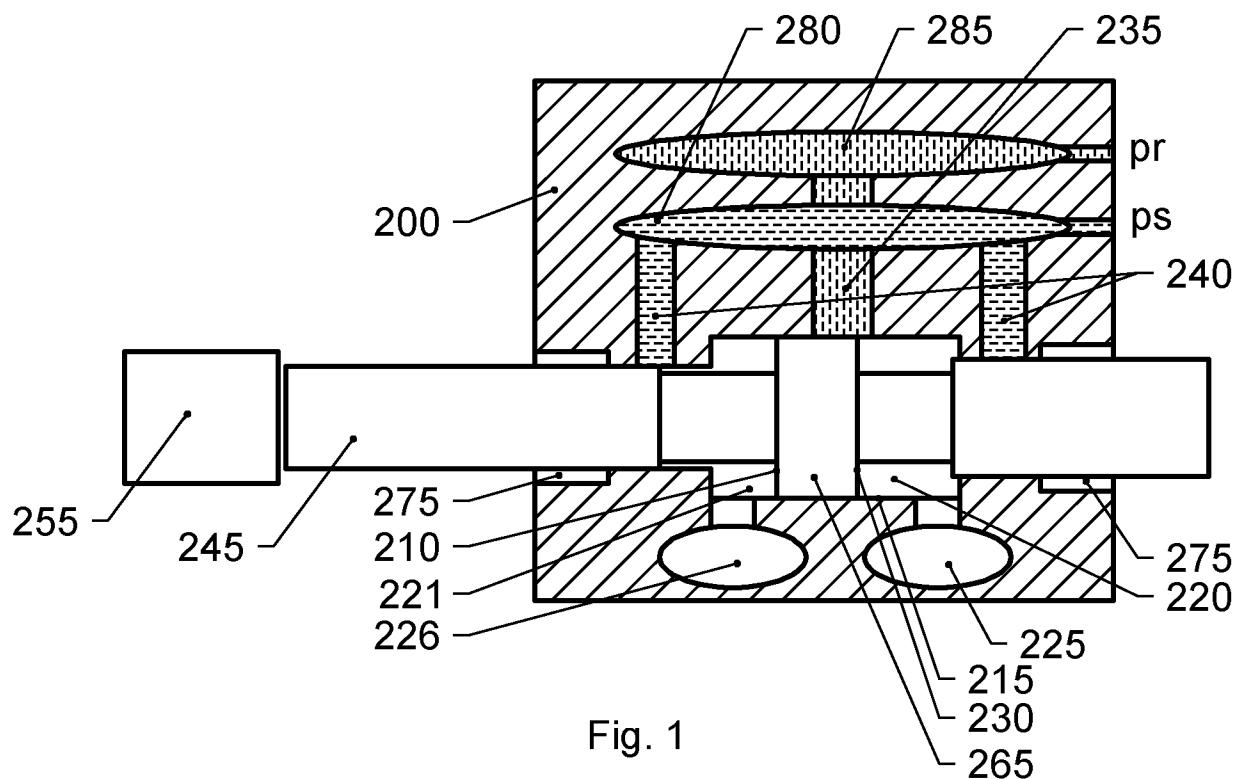
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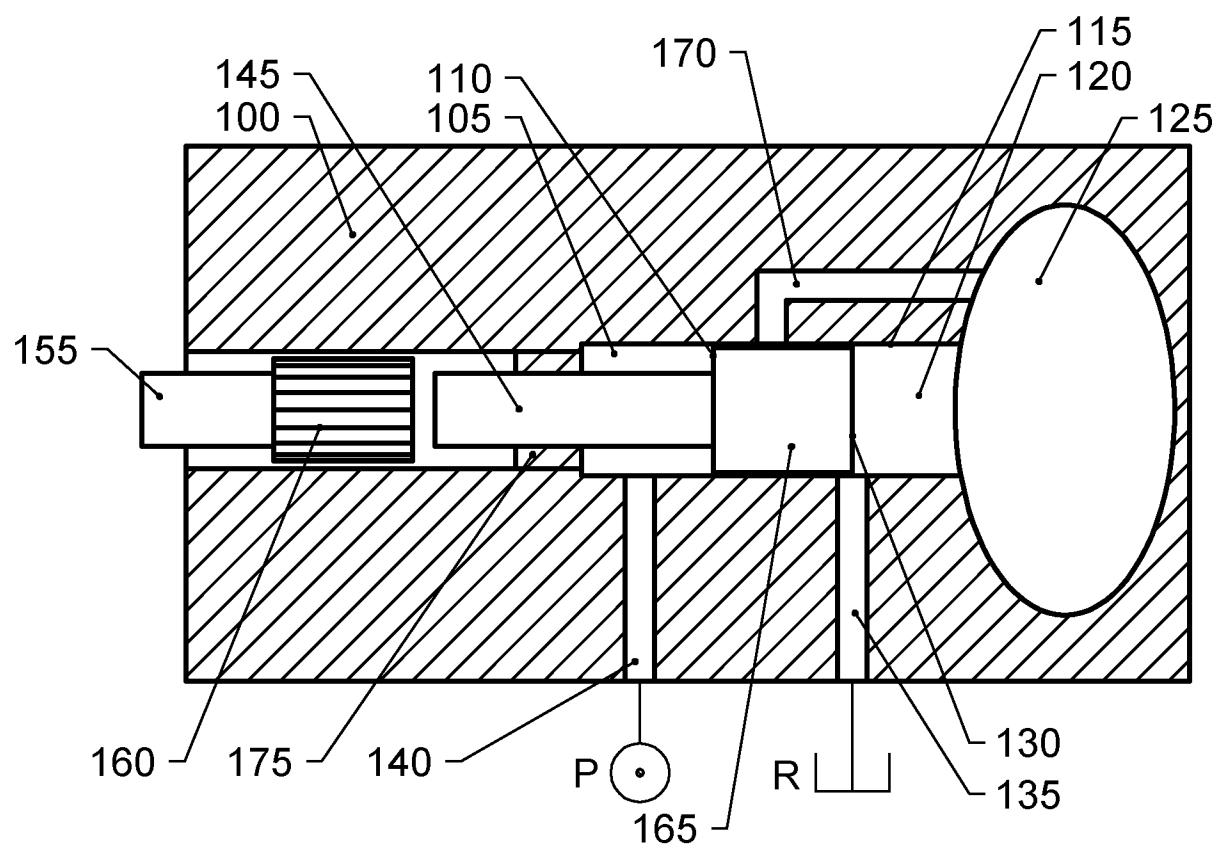
Fig. 2

Modulus of compressibility for oil-air mixture

$$\beta bt = yt\beta t$$
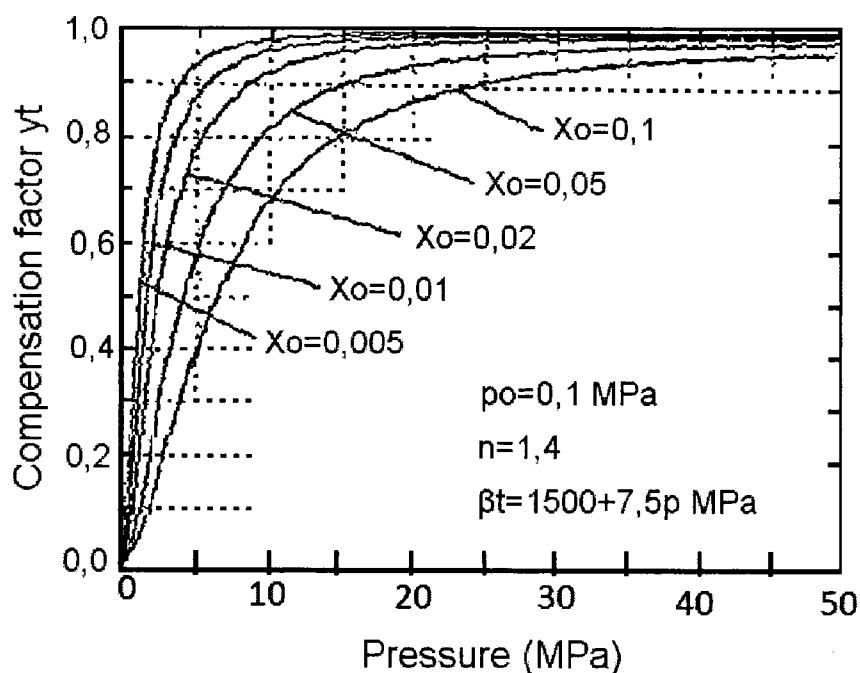
Fig. 3

3/4
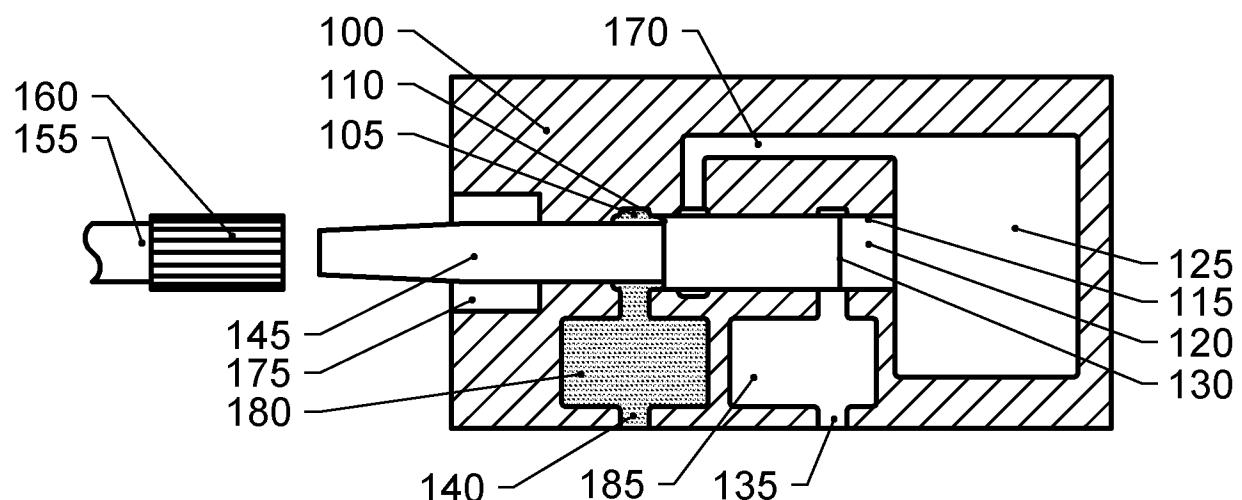
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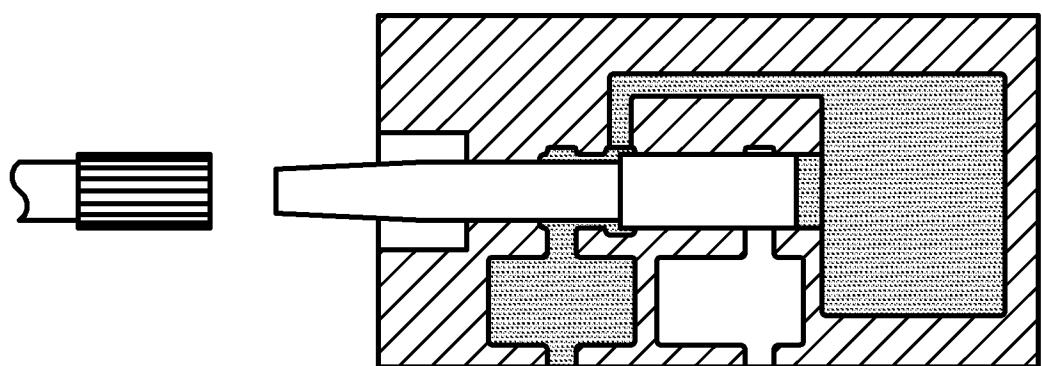
Fig. 4b
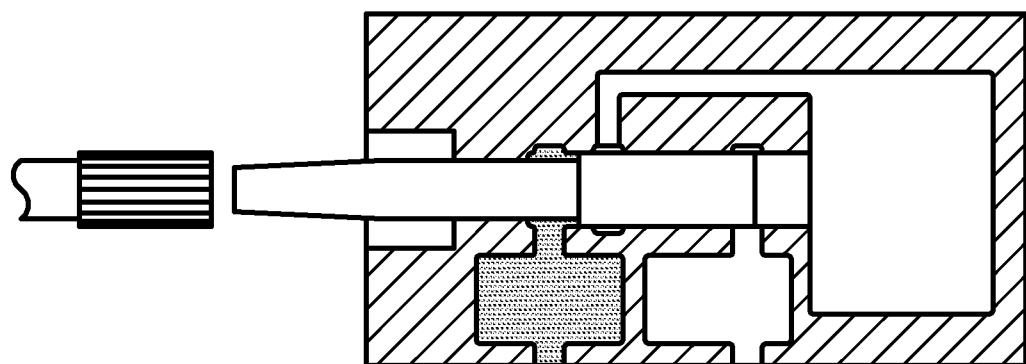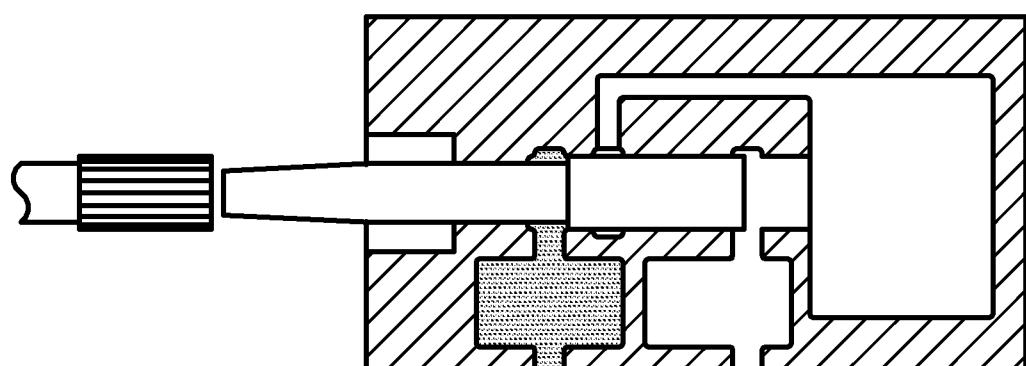
Fig. 4c

Fig. 4d

INTERNATIONAL SEARCH REPORT

A. CLASSIFICATION OF SUBJECT MATTER

IPC: see extra sheet

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

IPC: B25D, E21B, F03C

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

SE, DK, FI, NO classes as above

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

EPO-Internal, PAJ, WPI data

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	US 4282937 A1 (HIBBARD GEORGE A), 11 August 1981 (1981-08-11); whole document --	1-12
A	FR 701725 A (INGERSOLL RAND CO), 21 March 1931 (1931-03-21); whole document --	1-12
A	FR 716440 A (CIE PARISIENNE OUTIL AIR COMPR), 21 December 1931 (1931-12-21); whole document -- --------	1-12

☐ Further documents are listed in the continuation of Box C. ☒ See patent family annex.

* Special categories of cited documents: "A" document defining the general state of the art which is not considered to be of particular relevance "E" earlier application or patent but published on or after the international filing date "L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified) "O" document referring to an oral disclosure, use, exhibition or other means "P" document published prior to the international filing date but later than the priority date claimed	"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention "X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone "Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art "&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
28-06-2012	03-07-2012

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International Patent Classification (IPC)

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E21B 1/24 (2006.01)
E21B 4/14 (2006.01)

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Cited literature, if any, will be enclosed in paper form.

US	4282937 A1	11/08/1981	NONE
FR	701725 A	21/03/1931	NONE
FR	716440 A	21/12/1931	NONE