(12) STANDARD PATENT (11) Application No. **AU 2012240637 B2**
(19) AUSTRALIAN PATENT OFFICE

(54) Title
Device for rock- and concrete machining

(51) International Patent Classification(s)
B25D 9/12 (2006.01) ***E21B 1/24*** (2006.01)
B25D 9/04 (2006.01) ***E21B 4/14*** (2006.01)
B25D 9/14 (2006.01)

(21) Application No: **2012240637** (22) Date of Filing: **2012.04.03**

(87) WIPO No: **WO12/138287**

(30) Priority Data

(31) Number	(32) Date	(33) Country
1100252-4	**2011.04.05**	**SE**

(43) Publication Date: **2012.10.11**
(44) Accepted Journal Date: **2017.06.22**

(71) Applicant(s)
Atlas Copco Rock Drills AB

(72) Inventor(s)
Pettersson, Maria

(74) Agent / Attorney
Watermark Intellectual Property Pty Ltd, L 1 109 Burwood Rd, Hawthorn, VIC, 3122, AU

(56) Related Art
US 4658913 A
US 4282937 A

(12) INTERNATIONAL APPLICATION PUBLISHED UNDER THE PATENT COOPERATION TREATY (PCT)

(19) World Intellectual Property Organization
International Bureau

(43) International Publication Date
11 October 2012 (11.10.2012)

WIPO | PCT

(10) International Publication Number
WO 2012/138287 A1

(51) International Patent Classification:
B25D 9/12 (2006.01) *E21B 1/24* (2006.01)
B25D 9/04 (2006.01) *E21B 4/14* (2006.01)
B25D 9/14 (2006.01)

(21) International Application Number:
PCT/SE2012/050365

(22) International Filing Date:
3 April 2012 (03.04.2012)

(25) Filing Language: Swedish

(26) Publication Language: English

(30) Priority Data:
1100252-4 5 April 2011 (05.04.2011) SE

(71) Applicant *(for all designated States except US)*: **ATLAS COPCO ROCK DRILLS AB** [SE/SE]; S-70191 Örebro (SE).

(72) Inventor; and
(75) Inventor/Applicant *(for US only)*: **PETTERSSON, Maria** [SE/SE]; Bittinge 720, S-715 92 Stora Mellösa (SE).

(74) Agent: **NELANDER, Pontus**; Atlas Copco Rock Drills AB, Patents, S-701 91 Örebro (SE).

(81) Designated States *(unless otherwise indicated, for every kind of national protection available)*: AE, AG, AL, AM, AO, AT, AU, AZ, BA, BB, BG, BH, BR, BW, BY, BZ, CA, CH, CL, CN, CO, CR, CU, CZ, DE, DK, DM, DO, DZ, EC, EE, EG, ES, FI, GB, GD, GE, GH, GM, GT, HN, HR, HU, ID, IL, IN, IS, JP, KE, KG, KM, KN, KP, KR, KZ, LA, LC, LK, LR, LS, LT, LU, LY, MA, MD, ME, MG, MK, MN, MW, MX, MY, MZ, NA, NG, NI, NO, NZ, OM, PE, PG, PH, PL, PT, QA, RO, RS, RU, RW, SC, SD, SE, SG, SK, SL, SM, ST, SV, SY, TH, TJ, TM, TN, TR, TT, TZ, UA, UG, US, UZ, VC, VN, ZA, ZM, ZW.

(84) Designated States *(unless otherwise indicated, for every kind of regional protection available)*: ARIPO (BW, GH, GM, KE, LR, LS, MW, MZ, NA, RW, SD, SL, SZ, TZ, UG, ZM, ZW), Eurasian (AM, AZ, BY, KG, KZ, MD, RU, TJ, TM), European (AL, AT, BE, BG, CH, CY, CZ, DE, DK, EE, ES, FI, FR, GB, GR, HR, HU, IE, IS, IT, LT, LU, LV, MC, MK, MT, NL, NO, PL, PT, RO, RS, SE, SI, SK, SM, TR), OAPI (BF, BJ, CF, CG, CI, CM, GA, GN, GQ, GW, ML, MR, NE, SN, TD, TG).

Published:

— *with international search report (Art. 21(3))*

(54) **Title**: DEVICE FOR ROCK- AND CONCRETE MACHINING

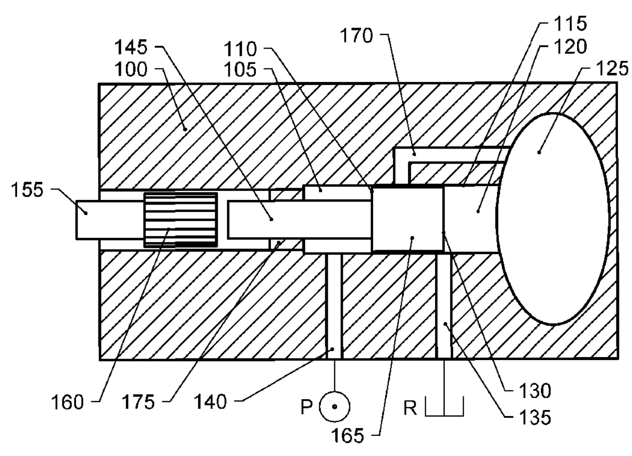

Fig. 2

(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.

WO 2012/138287 A1

2012240637 02 Jun 2017

Device for rock- and concrete machining

1. Field of the Invention

The present invention concerns hydraulic impact mechanisms of the type known as “slideless” or “valveless” to be used in equipment for machining at least one of rock and concrete, and equipment for drilling and breaking comprising such impact mechanisms.

2. Background of the Invention

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

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

2012240637 02 Jun 2017

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 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 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 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.

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 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

2012240637 02 Jun 2017

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 patent 4 282 937 discloses 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.

Russian patent publication SU 1068591 A discloses 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

2012240637 02 Jun 2017

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

It would be beneficial to provide a design of a valveless hydraulic impact mechanism that offers the opportunity of improving the efficiency without at the same time reducing the service interval.

3. Summary of the Invention

The present invention provides, in a first aspect thereof, a valveless hydraulic impact mechanism for use in equipment for at least one of rock and concrete machining, comprising a machine housing with a cylinder bore, and a piston mounted to move within the cylinder bore and arranged to carry out a repetitively reciprocating motion relative to the machine housing during operation, said reciprocating motion delivering impacts directly or indirectly onto a tool connectable to the equipment for machining at least one of rock and concrete, wherein the piston includes a driving part separating a first and a second drive chamber formed between the piston and the machine housing, wherein the first and second drive chambers are arranged to include during operation a driving medium at a predetermined impact mechanism pressure p, wherein the machine housing further includes channels opening out into the cylinder bore and arranged such as to include the driving medium during operation, the channels arranged such that with the

2012240637 02 Jun 2017

aid of the piston, during said reciprocating motion in the cylinder bore, open onto and close from one of the first and second drive chambers such that said one of said first and second drive chambers acquires a periodically alternating pressure for maintaining the reciprocating motion of the piston, wherein positions for the opening of the channels axially in the cylinder bore and for opening and closing of the channels along parts of the piston are adapted to maintain said one of said first and second drive chambers closed for the supply or drainage of the driving medium present in the one of said first and second chambers along a distance between an opening of a first one of said channels in association with a first turning point of the piston and an opening of a second one of said channels in association with a second turning point of the piston, wherein the motion of the piston along said distance continues during compression or expansion of the volume of said one of said first and second drive chambers, wherein said volume has been further adapted in order to achieve a slow change in pressure along said distance, and wherein a total volume V of the first and second drive chambers, including volumes that are in continuous connection with one and the same drive chamber during a complete stroke cycle, is dimensioned to be inversely proportional to the square of the predetermined impact mechanism pressure p and proportional, with a constant of proportionality k having a value in a range of 5.3 to 21.0, to the product of the energy E of the piston in the impact against the tool and the modulus of compressibility β of the driving medium according to the equation $V = k * \beta \;\; E/p^2$.

As will be noted from the above, the effective volume of the drive chambers is defined as the sum of the drive chamber volumes that

2012240637 02 Jun 2017

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 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, 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 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 the tool or a part that interacts with the tool, such as the part known as an "adapter".

As noted above, the relationship is expressed by the equation: $V = k * \beta * E/p^2$, where V is the effective drive chamber volume (by which is meant the sum of the volumes of the two drive chambers, including volumes that are in continuous connection with one and the same drive chamber during a complete stroke cycle). In the case in which

2012240637 02 Jun 2017

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 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 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 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 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 gas accumulators are continuously connected to the drive chambers

2012240637 02 Jun 2017

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 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 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

2012240637 02 Jun 2017

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.

Further preferred features of the present invention may be gleaned from the following description of a non-exhaustive embodiment of the invention, provided with reference to the accompanying drawings.

4. 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

2012240637 02 Jun 2017

braking is starting at the lower position; D - the lower turning point.

5. Detailed description of preferred embodiments

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 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 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 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 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 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 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 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 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

2012240637 02 Jun 2017

achieving high efficiency.

A functioning design involves an effective volume of 3 litres 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 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:

$$k = (3\cdot10^{-3}/200\cdot1687.5\cdot10^{6})\cdot(250\cdot10^{5})^{2} = 5.55.$$

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 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.

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

2012240637 02 Jun 2017

CLAIMS

1. A valveless hydraulic impact mechanism for use in equipment for at least one of rock and concrete machining, comprising a machine housing with a cylinder bore and a piston mounted to move within the cylinder bore and arranged to carry out a repetitively reciprocating motion relative to the machine housing during operation, said reciprocating motion delivering impacts directly or indirectly onto a tool connectable to the equipment for machining at least one of rock and concrete, wherein the piston includes a driving part separating a first and a second drive chamber formed between the piston and the machine housing, wherein the first and second drive chambers are arranged to include during operation a driving medium at a predetermined impact mechanism pressure p, wherein the machine housing further includes channels opening into the cylinder bore and arranged such as to include the driving medium during operation, the channels arranged such that, with the aid of the piston during said reciprocating motion in the cylinder bore, they open into and close from one of the first and second drive chambers such that said one of said first and second drive chambers acquires a periodically alternating pressure for maintaining the reciprocating motion of the piston, wherein positions for the opening of the channels axially in the cylinder bore and for opening and closing of the channels along parts of the piston are adapted to maintain said one of said first and second drive chambers closed for the supply or drainage of the driving medium present in the one of said first and second chambers along a distance between an opening of a first one of said channels in association with a first turning point of the piston and an opening of a second one of said channels in association with a second turning point of the piston, wherein the motion of the piston along said distance continues during compression or expansion of the volume of said one of said first and second drive chambers, wherein said volume is adapted to achieve a slow change in pressure along said distance, and wherein a total volume V of the first and second drive chambers, including volumes that are in continuous connection with one and the same drive chamber during a complete stroke cycle, is dimensioned to be inversely proportional to the

2012240637 02 Jun 2017

square of the predetermined impact mechanism pressure p and proportional, with a constant of proportionality k having a value in a range of 5.3 to 21.0, to the product of the energy E of the piston in the impact against the tool and the modulus of compressibility β of the driving medium according to the equation $V = k * \beta \; E/p^2$.

2. The hydraulic impact mechanism according to claim 1, wherein the constant of proportionality k is within the range $6.2 < k < 11$.

3. The hydraulic impact mechanism according to claim 1, wherein the constant of proportionality k is within the range of $7.0 < k < 9.5$.

4. The hydraulic impact mechanism according to any one of the preceding claims, wherein the volume of one of the first and second drive chambers is much greater than the volume of the other one of the first and second drive chambers.

5. The hydraulic impact mechanism according to any one of the preceding claims, wherein one of the first and second drive chambers has a constant pressure during essentially the complete stroke cycle.

6. The hydraulic impact mechanism according to any one of claims 1 to 3, wherein the first and second drive chambers are alternately set under pressure.

7. The hydraulic impact mechanism according to any one of the preceding claims, wherein the volumes of the first and second chambers extend symmetrically around the cylinder bore.

8. The hydraulic impact mechanism according to any one of the preceding claims, wherein the volumes of the first and second chambers extend concentrically around the cylinder bore.

9. The hydraulic impact mechanism according to claim 5, wherein the drive chamber with alternating pressure extends into the cylinder bore.

10. A rock drill comprising at least one impact mechanism

2012240637 02 Jun 2017

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 at least one impact mechanism according to any one of claims 1 to 9.

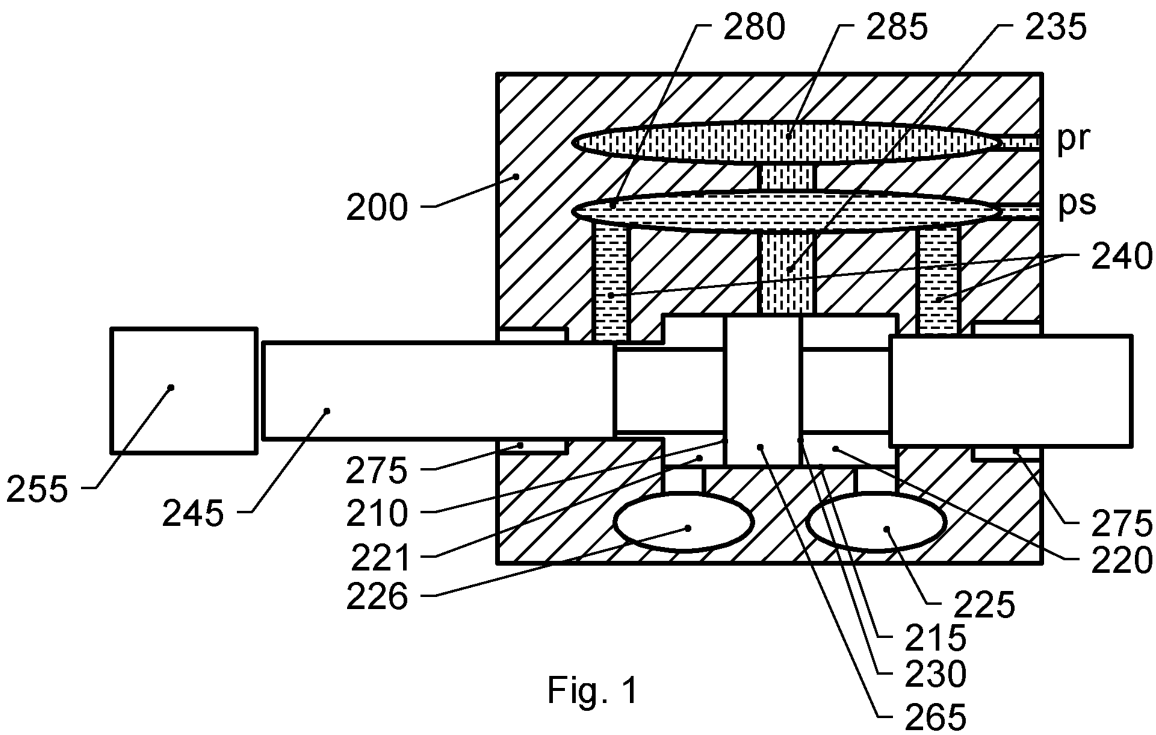
280
285
235
pr
ps
200
240
255
245
275
210
221
226
275
220
225
215
230
265

Fig. 1

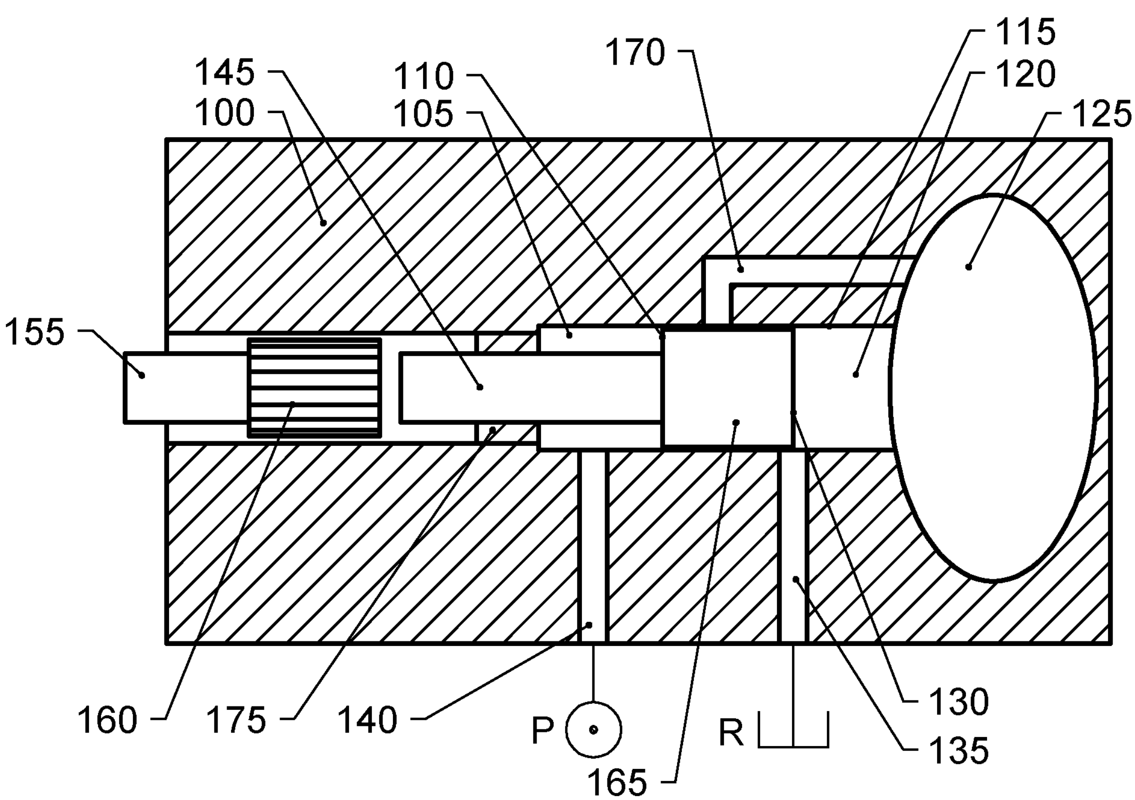
145
100
110
105
170
115
120
125
155
160
175
140
P
R
165
130
135

Fig. 2

Modulus of compressibility for oil-air mixture

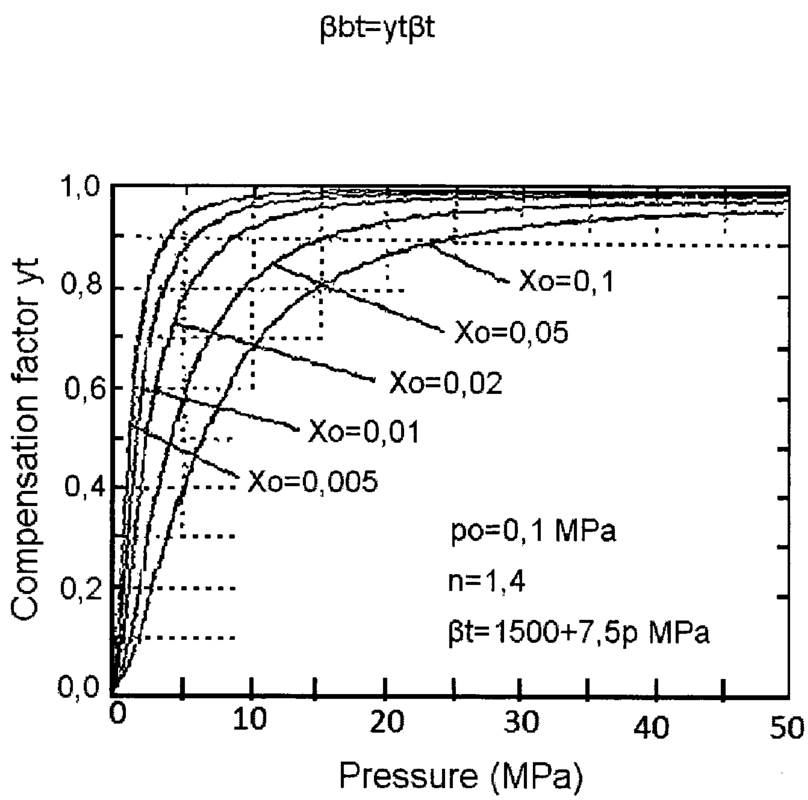

Fig. 3

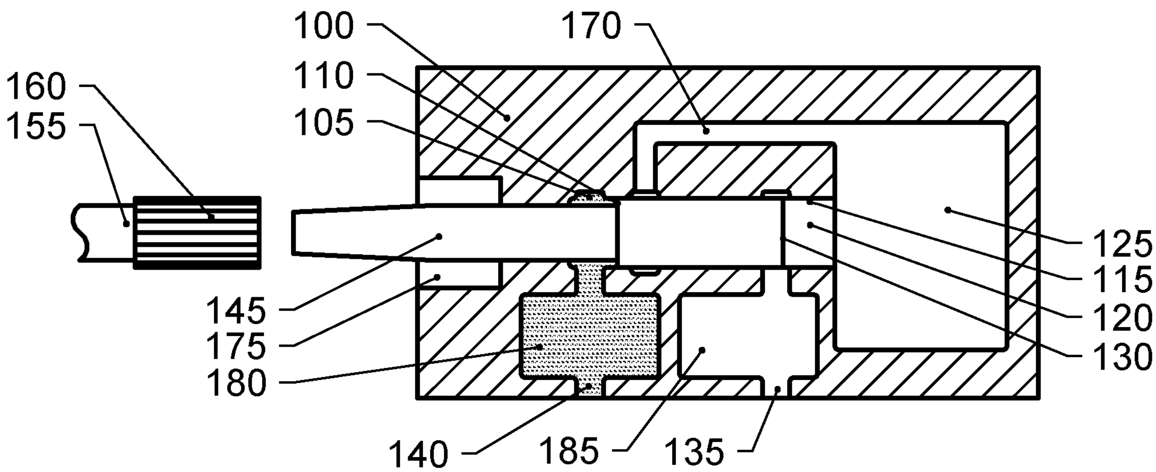

Fig. 4a

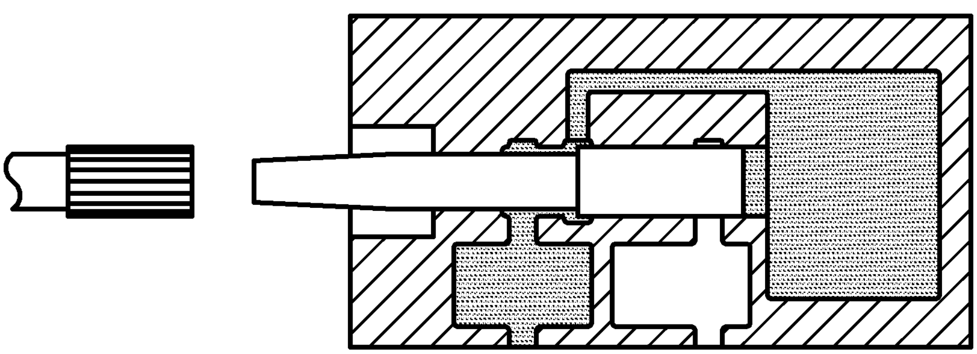

Fig. 4b

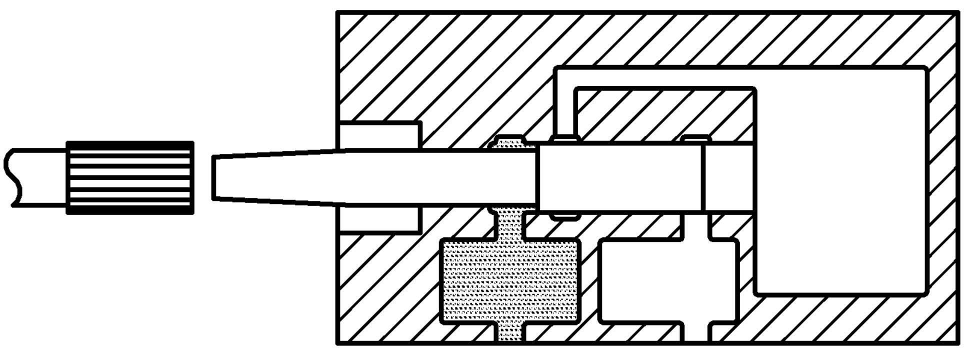

Fig. 4c

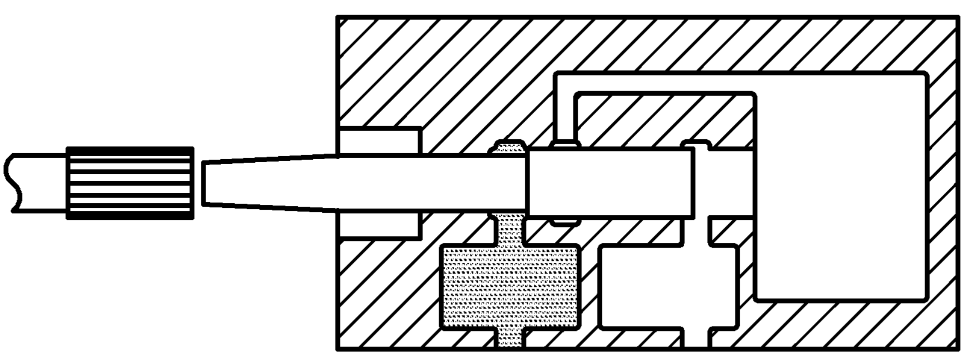

Fig. 4d