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(54) Benævnelse: **FREMGANGSMÅDE TIL DRIFT AF EN STEMPELKOMPRESSOR SAMT STEMPELKOMPRESSOR**

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METHOD FOR OPERATING A PISTON COMPRESSOR AND PISTON COMPRESSOR

Description

The invention relates to a method for operating a piston compressor and to a piston compressor.

Documents DE 10 2007 033 601 B3 and AT 3213 U1 each disclose a method for operating a piston compressor and a piston compressor having a reciprocating piston in a cylinder, an inlet valve and an outlet valve being provided in the cylinder on the side of a medium to be compressed and conveyed, the reciprocating piston being moved back and forth by way of a hydraulic drive with a hydraulic piston using a hydraulic medium in a first volume, with which the reciprocating piston is loaded on the side of the hydraulic drive.

Compressors are used in particular for compressing gaseous media. The efficiency of conventional piston compressors is very strongly influenced by the presence of residual volume or dead space in the top dead center. Gas or medium in this region leads to a re-expansion and a reduction in the inflowing delivery volume possible during the suction cycle. This region is unavoidable in order to compensate for manufacturing tolerances and thermal expansions of the components and consequently to avoid mechanical contact between the reciprocating pistons and the cylinder head in the compressor. Channels for the suction and pressure valves (i.e., inlet and outlet valves) increase this negative effect.

The reduction of this dead space reduces the re-expansion of the remaining medium and thereby increases the delivery volume or the efficiency of the compression process. So-called ionic compressors, in particular piston compressors, are operated hydraulically on the one hand and, on the other hand, compress a two-phase mixture consisting of a medium or gas and a liquid lubricant (i.e., an ionic liquid), which cannot be evaporated and for this reason can be completely separated again by a simple separation process. The advantage of such an ionic compressor is that by using a liquid phase in the compression space, the dead space in the cylinder can be reduced to a minimum and the efficiency of the compression process is thereby optimized.

The resulting additional component loads and thereby increased sound emissions caused by the characteristics similar to a liquid impact are disadvantageous. In addition, in the case of hydraulically driven compression concepts, an oil filling amount must be monitored due to internal leakage in the hydraulic circuit and must be compensated as required. Major problems arise from mechanical contacts at the reversal points of the reciprocating pistons and thereby increased component load and noise emissions, which are to be regarded as problematic especially in the case of an installation in the vicinity of residential areas and require additional sound insulation.

The concept of damping is known, for example, from the field of automotive engineering, in particular in shock absorbers. In this case, kinetic energy is dissipated in the form of vibrations. In addition to reducing undesired fluctuation or vibration of the vehicle, this also enables the reduction of component loads and sound emissions.

Against this background, the object is to provide a possibility of improving the operation of a piston compressor, in particular with regard to component load and sound emission.

Disclosure of the invention

This object is achieved by a method for operating a piston compressor and a piston compressor having the features of the independent claims. Preferred embodiments are the subject-matter of the dependent claims and the following description.

Advantages of the invention

The invention starts with a method for operating a piston compressor having a reciprocating piston in a cylinder, wherein an inlet valve and an outlet valve (or a suction valve and a pressure valve) are provided in the cylinder on the side of a medium to be compressed and conveyed (i.e., in the cylinder head). With a hydraulic drive, which comprises a hydraulic piston, the reciprocating piston is moved back and forth (or up and down) using a hydraulic medium in a first volume, with which the reciprocating piston is loaded on the side of the hydraulic drive.

The reciprocating piston oscillates in the cylinder between two reversal points, the so-called bottom dead center (BDC) and the so-called top dead center (TDC). When moving in the direction of the top dead center, medium present in the cylinder or cylinder head is compressed and then discharged through the outlet valve; when moving in the direction of the bottom dead center, medium is drawn in through the inlet valve.

In principle, the reciprocating piston would be moved synchronously with the hydraulic piston in this case. However, due to leakage effects in the circuit of the hydraulic medium (i.e., the first volume mentioned), it may happen that the reciprocating piston is no longer moved synchronously with the hydraulic piston. This means that, for example, at the bottom dead center, the reciprocating piston strikes the cylinder bottom, but the hydraulic piston moves further downward. This produces an underpressure in the first volume or in the hydraulic circuit. The reciprocating piston can also strike the cylinder head, while the hydraulic piston moves further upward. This produces an overpressure in the first volume or in the hydraulic circuit.

It is now provided that hydraulic medium is supplied into the first volume and/or discharged from the first volume as required depending on a position of the hydraulic piston and/or a rotational angle of a shaft provided for moving the hydraulic piston in relation to a position of the reciprocating piston and/or a pressure in the first volume. The corresponding variables can be determined by means of one or more suitable measuring devices.

The position of the hydraulic piston and the rotational angle of the shaft provided for moving the hydraulic piston are linked to one another and indicate a current position of the hydraulic drive. The position of the reciprocating piston and the pressure in the first volume are also linked to one another, namely in that the pressure increases or decreases when there is a strike of the reciprocating piston in the cylinder. If these variables are now determined, they can be set in relation to one another so that it can be detected whether a strike of the reciprocating piston occurs or, if applicable, whether such a strike of the reciprocating piston is imminent. Accordingly, hydraulic medium can then be supplied into the first volume or discharged from the first volume.

When or before the reciprocating piston strikes the cylinder bottom at the bottom dead center, the underpressure arising in the first volume or in the hydraulic circuit can thus be counteracted by supplying hydraulic medium. The strike can strike can thus be reduced or even prevented, which causes a reduction in sound emissions and component loading.

Accordingly, a strike at the top dead center can be reduced or even prevented by the discharge of hydraulic medium, which likewise causes a reduction in the sound emissions and component loading. Suitable valves can be provided for this purpose, which are correspondingly actuated, i.e., opened or closed. For a more detailed description of such valves, reference is made at this point to the description of the piston compressor or the description of the figures.

Preferably, a movement of the reciprocating piston is limited as required on the side of the hydraulic drive by way of a hydraulic damping unit using the hydraulic medium and forming a second volume which is at least partially delimited by the reciprocating piston. Such a damping unit can be used not only for further damping the movement of the reciprocating piston but also for adjusting a compression ratio.

For this purpose, the second volume is preferably connected to the first volume in order to reduce an amount of medium to be conveyed by means of the piston compressor. This is accompanied by an increase in dead space in the cylinder head. For this purpose, excess hydraulic medium (i.e., in order to reduce hydraulic medium in the second volume) is discharged from the first volume into a reservoir. It is also preferred if the first volume is connected to the reservoir for the hydraulic medium in order to increase an amount of medium to be conveyed by means of the piston compressor. This is accompanied by a reduction of dead space in the cylinder head. Required hydraulic medium (i.e., in order to increase the amount of hydraulic medium in the second volume or to fill up the second volume) is supplied from the reservoir.

The second volume can thus be filled with more or less hydraulic medium as required. Since the movement of the reciprocating piston in the direction of the bottom dead center (i.e., in the direction of the hydraulic drive) can be limited by the hydraulic medium in the second volume, the volume available for compression at the cylinder head or at the top dead center can thus be changed. Accordingly, the compression ratio can be changed.

A multi-stage piston compressor having at least two reciprocating pistons and corresponding cylinders is advantageously used as the piston compressor. However, a movement of these reciprocating pistons in the corresponding cylinders can still take place with the one hydraulic drive, then with a corresponding plurality of such first volumes. It goes without saying that a corresponding plurality of such damping units can then also be provided. Depending on the situation, the individual cylinders can then be arranged in series or in a star shape, for example. The compression then takes place in such a way that medium discharged from a cylinder is supplied to another cylinder and is further compressed there.

It is particularly preferred if an ionic liquid is used as the working liquid. In this context, the compressor is also referred to as a so-called ionic compressor. As already mentioned above, such ionic compressors offer advantages, such as a reduced dead volume. Through the supply or discharge of hydraulic medium proposed here, the remaining disadvantages of sound emission or component wear can now also be reduced.

The invention furthermore relates to a piston compressor having a reciprocating piston in a cylinder, wherein an inlet valve and an outlet valve are provided in the cylinder on the side of a medium to be compressed and conveyed. In addition, the piston compressor has a hydraulic drive with a hydraulic piston, by means of which the reciprocating piston can be moved back and forth using a hydraulic medium in a first volume, with which the reciprocating piston can be loaded on the side of the hydraulic drive. At least one measuring device is provided, by means of which a position of the hydraulic piston and/or a rotational angle of a shaft provided for moving the hydraulic piston and a position of the reciprocating piston and/or a pressure in the first volume can be determined. The piston compressor is now configured to supply hydraulic medium into the first volume and/or discharge hydraulic medium from the first volume as required depending on the position of the hydraulic piston and/or the rotational angle of the shaft provided for moving the hydraulic piston in relation to the position of the reciprocating piston and/or the pressure in the first volume.

Preferably, the piston compressor furthermore has a hydraulic damping unit by means of which a movement of the reciprocating piston can be limited as required on the side of the hydraulic drive using the hydraulic medium and forming a second volume which is at least partially delimited by the reciprocating piston. Advantageously provided in this case are a first valve, by means of which the second volume can be connected to the first volume, and a second valve, by means of which hydraulic medium can be discharged from the first volume into a reservoir for the hydraulic medium. An amount of medium to be conveyed by means of the piston compressor can thus be reduced. It is also preferred if a third valve is provided, by means of which the first volume can be connected to the reservoir for the hydraulic medium. Hydraulic medium can thus be supplied to the first volume. This third valve can preferably also be designed in such a way that hydraulic medium can be supplied automatically from the reservoir to the first volume if a lower pressure is present on the side of the first volume than on the side of the reservoir. For this purpose, the third valve can be designed, for example, as a check valve.

The piston compressor is advantageously designed as a multi-stage piston compressor having at least two reciprocating pistons and corresponding cylinders. An ionic liquid is expediently provided in the piston compressor as the working liquid.

With regard to the detailed explanation as well as further preferred embodiments and advantages of the piston compressor according to the invention, reference is made to the above explanations, which are correspondingly applicable here, concerning the method according to the invention, which is explained with reference to a piston compressor, in order to avoid repetitions.

The invention is schematically represented in the drawing using an exemplary embodiment and is described below with reference to the drawing.

Brief description of the drawing

Figure 1 schematically shows a piston compressor according to the invention in a preferred embodiment, which is suitable for carrying out a method according to the invention.

Detailed description of the drawing

Figure 1 schematically shows a piston compressor 100 according to the invention in a preferred embodiment, which is suitable for carrying out a method according to the invention.

Here, the piston compressor 100, also referred to in the form shown as a reciprocating piston compressor, comprises a cylinder 110 in which a reciprocating piston 111 can be moved back and forth or up and down. In principle, such a piston compressor can have a multi-stage design, i.e., a plurality of the shown cylinders 110 with reciprocating pistons 111 can be present. The following description with respect to the cylinder with reciprocating piston then correspondingly also applies to further cylinders with reciprocating pistons.

The piston compressor 100 is driven by a hydraulic drive, which here comprises a hydraulic piston 120. The hydraulic piston 120 is driven by a rotary wheel with shaft 121 with suitable linkage (hydraulic crank drive). Rotation of this shaft 121, as indicated by an arrow, leads to up-and-down movement of the reciprocating piston 110 by the use of hydraulic medium a (hydraulic oil) in a first volume 141, as likewise indicated by an arrow. The reciprocating piston 111 oscillates in the cylinder 110 between two reversal points, referred to as bottom dead center (BDC) and top dead center (TDC).

The frequency at which the shaft rotates and the reciprocating piston correspondingly moves up and down can be, for example, between 0.5 Hz and

12 Hz (but is generally kept constant); the stroke of the reciprocating piston can be, for example, between 30 mm and 100 mm. The stroke or stroke volume of the hydraulic piston is generally also constant.

The hydraulic medium a is thus conveyed here to the bottom side of the reciprocating piston 111. The reciprocating piston 111 is moved correspondingly upward and compresses a two-phase mixture in the so-called gas cylinder 114, i.e., the upper region of the cylinder. This two-phase mixture here comprises, on the one hand, a medium b to be compressed and conveyed and, on the other hand, an ionic working liquid. If the pressure in the cylinder 110 exceeds the counter pressure at the pressure valve or outlet valve 113, the latter is opened and the medium b is conveyed approximately isobarically into the pressure region until the top dead center is reached.

As soon as the hydraulic piston 120 moves downward, the pressure falls below the counter pressure in the cylinder 110 and the outlet valve 113 is closed. The downwardly moving reciprocating piston 111 reduces the pressure in the cylinder 110 until the applied pressure at the suction valve or inlet valve 112 sinks below the level in the suction region.

The smaller the residual volume or dead space, the earlier the inlet valve 112 can open and, proportionally thereto, the amount drawn in can be increased. Due to the system, the position of the reciprocating piston 111 can deviate from the stroke position of the hydraulic piston 120. The leakage-laden compression in the hydraulic drive subsequently results in the hydraulic medium a conveyed past the hydraulic piston 120 initially entering a reservoir 130 (or a tank).

From there, it can be conveyed back into the hydraulic circuit or the first volume 141 via a pump 131 and a heat exchanger 132, and finally a check valve 153, in order to compensate for the positional deviations between the hydraulic piston 120 and the reciprocating piston 111.

Within the scope of the present invention, the required amount can now be calculated and corrected, for example, via a data comparison between a measuring device 161 (for example, a displacement measuring system) and a measuring device 160 (for example, a rotational angle sensor). For example, while the position x of the reciprocating piston 111 can be determined by means of the measuring device 161, a rotational angle φ of the shaft 121 can be determined with the measuring device 160. In addition, a pressure p in the first volume 141 can, for example, also be detected by means of a suitable measuring device 162.

A strike of the reciprocating piston 111 against the cylinder head is now detected by an excess pressure increase at the end of the actually isobaric exhaust phase, as a result of which the excess hydraulic medium is conveyed back into the reservoir 130 via a pressure-limiting valve 154.

Mechanical contact of the reciprocating piston 111 with the piston bottom or cylinder bottom is detected by an insufficient pressure during the inflow phase. Here, the missing amount of hydraulic medium is drawn or supplied via the check valve 153 (which is also referred to as the third valve within the scope of the invention) in order to move the reciprocating piston 111 into the standard range.

Thus, for example, by now taking a pressure measurement in the first volume as a function of the rotational angle of the shaft 121, it can be detected if a strike of the reciprocating piston 111 against the cylinder head or cylinder bottom occurs or is imminent, and hydraulic medium can be discharged or supplied, for example, by suitable actuation of the valves 153 or 154.

Furthermore, a damping unit 140 is provided by means of which an adaptive damping system can be realized, irrespective of the frequency, of the pressure ratios in the piston compressor and of the leakage in the hydraulic region. This damping unit 140 can be used to dampen the downward movement of the reciprocating piston 111 and thus to reduce the sound emissions and mechanical loads during the movement in the direction of the bottom dead center.

If a leakage in the oil circuit, i.e., in this case in the first volume 141, is detected via the pressure measurement resolved by rotational angle in the hydraulic system, i.e., in this case also in the first volume 141, this can be compensated. In addition, it is possible by means of the adaptive damping system to adjust the top and the bottom dead center and thus the compression ratio or the conveyed amount of medium. For this purpose, additional hydraulic medium is supplied or excess hydraulic medium is discharged or pushed back into the reservoir as required or depending on the demand for the required amount.

If the conveyed amount is to be increased or the existing dead space is to be reduced, a first valve 150 is closed. As a result, the reciprocating piston 111 is prevented from moving in the direction of the bottom dead center or in the direction of the hydraulic drive, even though the hydraulic piston 120 moves downward. As a result, the required amount of hydraulic medium is drawn into the circuit or into the first volume 141 via the check valve 153 by the underpressure thus produced. If the hydraulic piston 120 moves upward again, the system is closed and the reciprocating piston 111 has been raised by a defined volume (by the additionally conducted amount of hydraulic medium), whereby the dead space is reduced and the conveyed amount of medium to be compressed is increased.

If the amount of hydraulic medium is raised too much and the reciprocating piston 111 is about to collide with the cylinder head, the third valve 155 can be opened and the filling amount reduced.

If the conveyed amount is to be reduced or the available dead space is to be increased, the first valve 150 is opened in order to not influence the downward movement of the reciprocating piston 111. At the same time, the second valve 155 is opened in order to reduce or discharge a defined amount of hydraulic medium, which takes place by the upward movement of the hydraulic piston 120. When the required position of the reciprocating piston is reached, the first valve 150 can be closed again.

If this takes place before the hydraulic piston 120 reaches the bottom dead center, the hydraulic piston additionally conveys the necessary amount of hydraulic medium via the check valve 153.

These adjustments can be brought closer to the required operating point in a control loop by repeated iteration. If a change in the intermediate circuit pressures of the individual stages is required (i.e., in the case of a multi-stage piston compressor), this can take place in the same way as just described. Only the pressure is used as the controlled variable and is adjusted in the circuit with the other stages.

If an operating point has been adjusted, a change in the pressure in the system is directly proportional to the position of the reciprocating piston 111. A deviation of the position of the position x of the reciprocating piston 111 associated with the rotational angle φ of the shaft 121 due to leakage can be effected by blocking the reciprocating piston 111 by means of the damping unit 140; hydraulic medium is additionally conveyed via the check valve 153 and the incorrect position is compensated.

This adaptive damping system makes it possible to optimize hydraulically driven piston compressors with regard to the usable stroke volume. The piston travel can thus be made variable and optimized with regard to the delivery volume, pressure and effectiveness depending on the requirements for the system.

The residual volume between the reciprocating piston and the cylinder head, the so-called dead space, expands during the downward movement of the reciprocating piston and, depending on the size, influences the opening time of the generally spring-actuated suction valve. The larger the dead space, the later said valve opens and the smaller the delivery volume that can be drawn in.

This directly influences the efficiency of the respective compressor stage. The intermediate circuit pressures can thus be adapted to the other stages, which may be required in certain operating ranges. It is thus possible using the proposed method or the piston compressor to realize a high degree of variability with regard to the required delivery volume and the applied pressures in the intermediate circuit.

As a result of the possibility of damping the reciprocating piston and the stroke optimization, it is possible to reduce mechanical loads over the long term, which on the one hand leads to an increase in the possible service life of the compressor components and, on the other hand, makes it possible to use materials that have a lower quality and thereby allow greater freedom in terms of cost optimization with regard to raw material and production costs.

The aforementioned effects involve the reduction of the vibrations and noise emissions, as a result of which savings can also be made with regard to previously necessary damping measures and the operation of, for example, a gas station system (in which hydrogen is compressed with such a piston compressor, for example) in residential areas can consequently be made easier.

As a result of the described configuration, such a piston compressor is very variable and thereby simplifies the use of a modular system, even with regard to different operating media, limit values and requirements, and thus enables simpler series production due to the now possible structural sameness of the individual components despite different applications.

An expansion of existing or already delivered piston compressors to a proposed piston compressor allows optimization of the operating parameters and an increase in efficiency. In addition, it is possible to increase existing service lives and to reduce vibrations and sound emissions.

Incorporation into an existing system (i.e., an existing piston compressor or a plurality of them) can be carried out in the course of a regular maintenance process. For this purpose, a displacement measuring system and a rotary encoder (within the meaning of the aforementioned measuring devices) can additionally be mounted, and the automation programming of the system can be expanded by the necessary control routines.

A further embodiment consists in that it is possible to implement an expander system in conjunction with controllable suction and pressure valves, for example on a piezoelectric basis, for example as used in automotive applications. Such a system utilizes the expansion work during expansion of the gas, for example in a dispenser system in which gas of a lower pressure level is required, and can thereby be used for power generation on the basis of an energy recovery system.

PATENTKRAV

1. Fremgangsmåde til drift af en stempelkompressor (100) med et frem- og tilbagegående stempel (111) i en cylinder (110), idet der er anbragt en indløbsventil (112) og en udløbsventil (113) i cylinderen (110), for så vidt angår et medium (b), der skal komprimeres og transporteres,

hvor det frem- og tilbagegående stempel (111) bevæges frem og tilbage ved hjælp af et hydraulisk drev (120, 121) med et hydraulikstempel (120) under anvendelse af et hydraulisk medium (a) i et første volumen (141), med hvilket det frem- og tilbagegående stempel (111) fyldes, for så vidt angår det hydrauliske drev (120, 121),

kendetegnet ved, at det hydrauliske medium efter behov (a) tilføres det første volumen (141) og/eller udtømmes fra det første volumen (141), afhængigt af en position af hydraulikstemplet (120) og/eller en rotationsvinkel (φ) af en aksel (121), der er tilvejebragt til bevægelse af hydraulikstemplet (120), i forhold til en position (x) af det frem- og tilbagegående stempel (111) og/eller et tryk (p) i det første volumen (141).

2. Fremgangsmåde ifølge krav 1, hvor en bevægelse af det frem- og tilbagegående stempel (111) begrænses efter behov, for så vidt angår det hydrauliske drev (120, 121), ved hjælp af en hydraulisk dæmpningsenhed (140) under anvendelse af det hydrauliske medium (a) og dannelse af et andet volumen (142), der mindst delvist begrænses af det frem- og tilbagegående stempel (111).

3. Fremgangsmåde ifølge krav 2, hvor det andet volumen (142) forbindes med det første volumen (141) med henblik på at reducere en mængde medium (b), som skal transporteres ved hjælp af stempelkompressoren (100), hvor der udtømmes overskydende hydraulisk medium (a) fra det første volumen (141) ind i et reservoir (130), og/eller

hvor det første volumen (141) forbindes med reservoiret (130) til det hydrauliske medium med henblik på at forøge en mængde medium (b), som skal transporteres ved hjælp af stempelkompressoren (100), hvor der tilføres nødvendigt hydraulisk medium fra reservoiret (130).

4. Fremgangsmåde ifølge et hvilket som helst af de foregående krav, hvor der anvendes en flertrinsstempelkompressor med mindst to frem- og tilbagegående stempler og tilsvarende cylindre som stempelkompressor (100).

5. Fremgangsmåde ifølge et hvilket som helst af de foregående krav, hvor der anvendes en ionisk væske som arbejdsvæske.

6. Stempelkompressor (100) med et frem- og tilbagegående stempel (111) i en cylinder (110), idet der er anbragt en indløbsventil (112) og en udløbsventil (113) i cylinderen (110), for så vidt angår et medium (b), der skal komprimeres og transporteres,

med et hydraulisk drev (120, 121) med et hydraulikstempel (120), ved hjælp af hvilket det frem- og tilbagegående stempel (111) kan bevæges frem og tilbage under anvendelse af et hydraulisk medium (a) i et første volumen (141), med hvilket det frem- og tilbagegående stempel (111) kan fyldes, for så vidt angår det hydrauliske drev (120, 121),

kendetegnet ved mindst én måleindretning (160, 161, 162), ved hjælp af hvilken en position af hydraulikstemplet (111) og/eller en rotationsvinkel (φ) af en aksel (121), der er tilvejebragt til bevægelse af hydraulikstemplet (120), og en position af det frem- og tilbagegående stempel (x) og/eller et tryk (p) i det første volumen (141) kan bestemmes,

hvor stempelkompressoren (100) er udformet til at tilføre hydraulisk medium (a) til det første volumen (141) og/eller at udtømme det fra det første volumen (141) efter behov, afhængigt af hydraulikstemplets (120) position og/eller rotationsvinklen af den (φ) aksel (121), der er tilvejebragt til bevægelse af hydraulikstemplet (120), i forhold til positionen (x) af det frem- og tilbagegående stempel (111) og/eller trykket (p) i det første volumen (141).

7. Stempelkompressor (100) ifølge krav 6, der endvidere har en hydraulisk dæmpningsenhed (140), ved hjælp af hvilken en bevægelse af det frem- og tilbagegående stempel (111) kan begrænses efter behov, for så vidt angår det hydrauliske drev (120, 121), under anvendelse af det hydrauliske medium (a) og

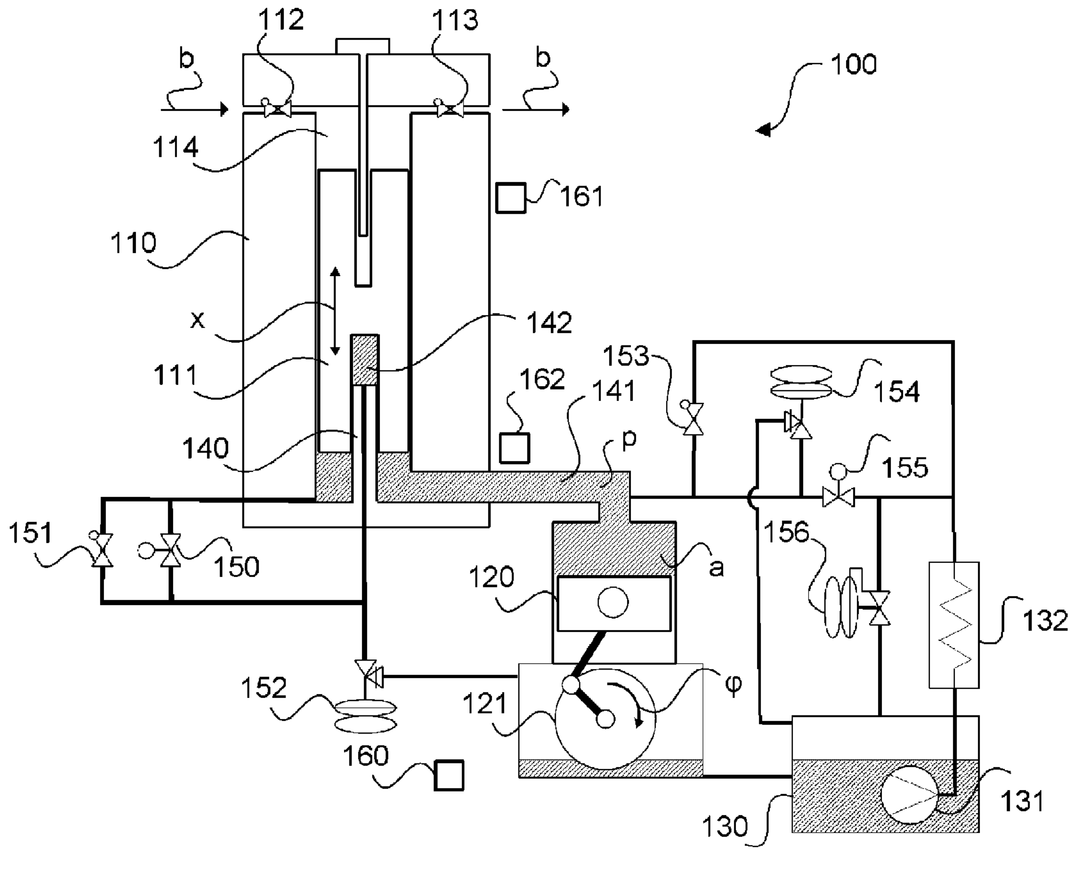
dannelse af et andet volumen (142), der mindst delvist begrænses af det frem- og tilbagegående stempel (111).

8. Stempelkompressor (100) ifølge krav 7, der har en første ventil (150), ved hjælp af hvilken det andet volumen (142) kan forbindes med det første volumen (141), og har en anden ventil (155), ved hjælp af hvilken det hydrauliske medium (a) kan udtømmes fra det første volumen (141) ind i et reservoir (130) til det hydrauliske medium, og/eller

har en tredje ventil (153), ved hjælp af hvilken det første volumen (141) kan forbindes med reservoiret (130) til det hydrauliske medium.

9. Stempelkompressor (100) ifølge et hvilket som helst af kravene 6 til 8, der er udformet som en flertrinsstempelkompressor med mindst to frem- og tilbagegående stempler og tilsvarende cylindre.

10. Stempelkompressor (100) ifølge et hvilket som helst af kravene 6 til 8, hvor der er tilvejebragt en ionisk væske som arbejdsvæske.



112
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Fig. 1