

(19) World Intellectual Property
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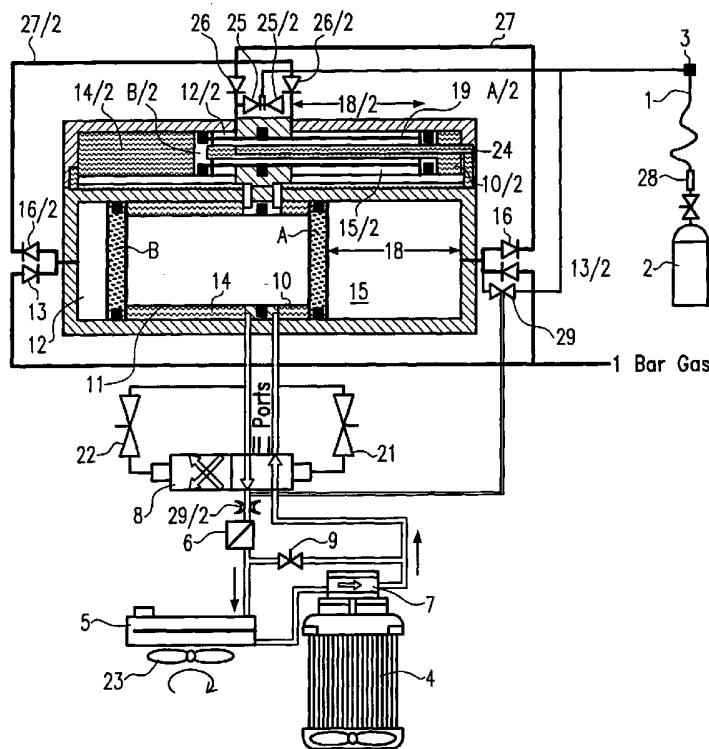
(43) International Publication Date
4 March 2004 (04.03.2004)

PCT

(10) International Publication Number
WO 2004/018873 A2

- (51) International Patent Classification⁷: **F04B** (72) Inventor; and
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- (21) International Application Number:
PCT/EP2003/009348
- (22) International Filing Date: 22 August 2003 (22.08.2003) (81) Designated States (national): AU, BR, CA, IN, JP, MX,
NZ, PH, RU, SG, US.
- (25) Filing Language: English (84) Designated States (regional): European patent (AT, BE,
BG, CH, CY, CZ, DE, DK, EE, ES, FI, FR, GB, GR, HU,
IE, IT, LU, MC, NL, PT, RO, SE, SI, SK, TR).
- (26) Publication Language: English
- (30) Priority Data:
0219595.6 22 August 2002 (22.08.2002) GB Published:
— without international search report and to be republished
upon receipt of that report
- (71) Applicant (for all designated States except US): **LAT-
TICE INTELLECTUAL PROPERTY LTD.** [GB/GB];
1-3 Strand, London WC2N 5EH (GB). For two-letter codes and other abbreviations, refer to the "Guid-
ance Notes on Codes and Abbreviations" appearing at the begin-
ning of each regular issue of the PCT Gazette.

(54) Title: TWO STAGE DOUBLE ACTING HYDRAULIC/GAS COMPRESSOR



(57) Abstract: A two stage gas compressor, comprising: a first compressor stage, comprising a first chamber, a first piston and a second piston, the first and second pistons being rigidly connected by a first connecting member, thereby defining a first volume opposite the first piston, a second volume opposite the second piston and a third volume between the first and second pistons; a second compressor stage, comprising a second chamber, a third piston and a fourth piston, the third and fourth pistons being rigidly connected by a second connecting member, thereby defining a fourth volume opposite the third piston, a fifth volume opposite the fourth piston and a sixth volume between the third and fourth pistons; at least one hydraulic conduit for conveying hydraulic fluid to said first and second compression stages; and at least one gas conduit for conveying said gas between said first compressor stage and said second compressor stage. Each pair of pistons is movable between two extremes under hydraulic force, whereby gas pressurized once in the first compressor stage is fed to the second stage for

further compression. The compressor stages act independently on a common hydraulic supply.

Two stage double acting hydraulic/gas compressor

- 5 The present invention relates to machinery for providing a pressurised gas, and more particularly to a two-stage double acting hydraulic/gas compressor.

There are many situations in industry where it is necessary to provide a compressed fluid, such as natural gas. For example motor vehicles operating with compressed
10 natural gas as an engine fuel require the gas to be compressed to around 200 bar in order to store sufficient quantity in a volume comparable with liquid fuel.

Conventional reciprocating gas compressors have been employed of the type using rotary movement to reciprocate the piston. Such reciprocating gas compressors
15 usually operate with a number of stages in sequence such that the compression ratio in each stage is between 3:1 and 7:1. The operating speed of the piston in this type of compressor may be around 10 Hz and intercooling provided between each compression stage to dissipate the heat generated when the gas is compressed. In these relatively high-speed compressors, designs to achieve gas tight sealing are
20 expensive, particularly at pressures up to 200 bar.

Designs have been proposed to avoid some of these drawbacks: British patent application GB-A-2,346,938 discloses a mains fuel gas reciprocating compressor that includes two hydraulic rams, each comprising a chamber divided by a piston. One
25 side of each ram is alternately supplied with hydraulic fluid by an electric pump, the other side of each alternately receiving mains gas through non-return valves and exhausting through non-return valves to a vehicle fuel reservoir via a pipe and a quick release coupling. In one embodiment, a two-stage compressor with twin, opposed single acting hydraulic pistons, linked by a rod, are used with different gas
30 compressing volumes.

The present invention seeks to provide a compressor that is more efficient, has a simpler and more effective sealing arrangement, and is less complex and less expensive to manufacture than compressor designs advanced heretofore.

- The present invention provides a two stage gas compressor, comprising: a first compressor stage, comprising a first chamber, a first piston and a second piston, the first and second pistons being rigidly connected by a first connecting member, thereby defining a first volume opposite the first piston, a second volume opposite the second piston and a third volume between the first and second pistons; a second compressor stage, comprising a second chamber, a third piston and a fourth piston, the third and fourth pistons being rigidly connected by a second connecting member, thereby defining a fourth volume opposite the third piston, a fifth volume opposite the fourth piston and a sixth volume between the third and fourth pistons; at least one hydraulic conduit for conveying hydraulic fluid to said first and second compression stages; and at least one gas conduit for conveying said gas between said first compressor stage and said second compressor stage.
- Preferably, the first connecting member is movable between a first position, where said first volume is a maximum and said second volume is a minimum, and a second position, where said first volume is a minimum and said second volume is a maximum. Preferably, the second connecting member is movable between a first position, where said third volume is a minimum and said fourth volume is a maximum, and a second position, where said third volume is a maximum and said fourth volume is a minimum. Preferably, in use, the first connecting member is in its first position when the second connecting member is in its second position, and vice versa.
- Preferably, said at least one hydraulic conduit includes a first hydraulic conduit arranged to convey hydraulic fluid from said first compression stage to said second compression stage. Preferably, said first hydraulic conduit is arranged to convey hydraulic fluid between said third volume and said fourth and fifth volumes. Preferably, said at least one hydraulic conduit includes a second hydraulic conduit arranged to convey hydraulic fluid from a pumped supply of hydraulic fluid to said first compression stage. The compressor preferably further including a bistable shuttle valve between said a pumped supply of hydraulic fluid to said first compression stage.

Preferably, said at least one gas conduit includes a first gas conduit arranged to convey pressurised gas from said first volume to said sixth volume when said first connecting member moves between its first position and second position. Preferably, said at least one gas conduit includes a second gas conduit arranged to convey
5 pressurised gas from said second volume to said sixth volume when said first connecting member moves between its second position and first position. Preferably, said at least one gas conduit includes a gas supply conduit arranged to convey unpressurised gas (a) to said second volume when said first connecting member moves between its first position and second position, and/or (b) to said first volume
10 when said first connecting member moves between its second position and first position.

The compressor preferably includes a first sealing member sealingly dividing said third volume into a first section and a second section. The compressor preferably
15 includes a second sealing member sealingly dividing said sixth volume into a third section and a fourth section.

Compressor operation

The system consists of a hydraulic power circuit linked directly and integrally with a
20 gas compression circuit. A flexible hose fuel delivery mechanism with quick release nozzle is provided to deliver compressed gas to an external storage cylinder.

The hydraulic power circuit consists of an electric motor coupled to a hydraulic gear or piston pump. High-pressure fluid output from the pump is connected to a spool
25 type reciprocating shuttle valve, pressure relief valves and a pair of two hydraulically opposed cylinders or rams. The 1st stage ram has two fluid connections, each for flow/discharge to the shuttle valve. The 2nd stage ram also has two fluid connections, each for flow/discharge to the shuttle valve that are ported through the 1st stage ram. The low pressure or discharge from the shuttle valve is connected via a filter and oil
30 cooler to a sump, containing a reservoir of hydraulic fluid. The hydraulic pump intake is connected to a point on the sump that is gravitationally well below the fluid level.

The gas compression circuit consists of two pairs of opposed cylinders or rams, which are integral with the hydraulic rams. Each gas ram has two gas connections.

One is for the gas inlet and the other is for higher-pressure gas discharge. Operational non-return valves (compressor valve) are fitted to both the inlet and outlet connections of each gas ram.

- 5 The high-pressure gas delivery pipe is of a small-bore flexible type fitted with an in-line quick release breakaway coupling. The fuel delivery pipe is equipped with a quick release fuel nozzle for connection to a matching refuelling receptacle to deliver fuel to each high-pressure gas storage vessel. For motor vehicle applications, the storage vessel is usually mounted under the vehicle body. To facilitate easy
- 10 uncoupling, a pressure off-loading mechanism has been developed to automatically off-load gas in the fuel delivery pipe to enable the refuelling nozzle to be safely removed at low fuel pressures. To safeguard a vehicle being inadvertently driven away without disconnecting the fuel delivery pipe, a break-away coupling has been included to prevent damage to the system.

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Further particular embodiments and advantages of the invention are set out below. In the compressor in accordance with aspects of the invention —

- (1) the 1st and 2nd stage pistons operate independently from a common hydraulic supply;
- 20 (2) the 1st and 2nd stage pistons are double acting;
- (3) the 1st and 2nd stage stroke length are set independently to give the desired gas compression ratios;
- (4) the hydraulic gearing can be optimised to minimise electrical power and maximise gas throughput, whilst satisfying the hydraulic gear pump specification (rating) and
- 25 compressor cooling requirements;
- (5) the hydraulic optimisation has three new facets:
- i. optimisation can be with hydraulic pressures below the gas - this was not possible in the previous submission,
- ii. independent 1st and 2nd stage stroke lengths helps to achieve the
- 30 required geometries,
- iii. double acting pistons - doubles the gas throughput and compacts and simplify the design;
- (6) the off-loading gas can be recycled through the compressor moving parts;

- (7) the pistons cannot be stalled by means of a high-pressure intake gas charge (i.e. fuel-line gas off-loaded in to the 1st stage cylinder);
- (8) the 1st stage hydraulic gearing can be biased to overcome 2nd stage dead/residual volumes;
- 5 (9) a 'dip-tube' is used to inter-cool gas carrying parts which otherwise would not have access to hydraulic cooling;
- (10) purposely designed compressor valves have been designed, based on a shuttle valve to give the desired gas hold times up to and in excess of 100 times that of a standard reciprocating compressor;
- 10 (11) the compressor valve design is dual acting (gas intake and exhaust) and operates without a closure spring — having unique gas impulse mechanism to close the valve;
- (12) the use of purposely designed compressor valve seals which can be enhanced with the use of plastic polymers in which the closure force is pressure
- 15 assisted;
- (13) the use of purposely-designed compressor valve seals in which the seal aspect ratio prevents jamming which can accommodate a closure spring if necessary;
- (14) the control (delivery gas pressure) can be achieved from the inlet low-pressure
- 20 gas and hydraulic supplies;
- (15) non intrinsically safe electrical components can be used in construction of the appliance by means of cooling a air-flow and the interaction of the off-loading valve mechanism (i.e. in operation gas leakages diluted to below the LEL whilst in standby the compressor reverts to a naturally ventilated low-pressure gas appliance);
- 25 (16) an air purge can enhance item (13) above; and
- (17) the appliance can be used as a room sealed device for indoor use.

Embodiments of the invention will now be described, by way of example, with reference to the accompanying drawings, in which:

30

Figure 1 provides a schematic illustration of the layout of the compressor according to the invention;

Figure 2 shows the operation of the compressor valve in the compressor of Fig. 1;

Figure 3 depicts the operation of the compressor exhaust valve in the compressor of Fig. 1; and

Figure 4 illustrates another embodiment of the invention in which the compressor arrangement is provided as a compact freestanding appliance within a casing.

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Two stage double acting compressor operation

Operation of the system is described by reference to Fig. 1. The high-pressure gas delivery hose 1 is connected to the inlet of vehicle gas fuel tank 2 by means of the quick release coupling 3 and the break-away coupling 28.

10

The hydraulic pump motor 4 is energised by means of a trip relay switch (not shown).

15

Hydraulic oil is drawn from the oil cooler sump 5 at atmospheric pressure, and oil filter 6, into the hydraulic pump 7. Rotation of the gears within the pump, forces oil to flow into the spool valve 8 at high pressure. If the pressure exceeds a set value, typically 120 bar, then the spill-back relief valve 9 opens to allow oil to bypass the spool valve 8 and flow back to the sump 5. In addition, hydraulic pressure caused by hydraulic oil returning to the sump 5 at point 29/2 maintains the off-loading valve 29 closed whilst hydraulic oil is flowing. As soon as the hydraulic pump motor 4 is de-

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energised, the pressure at point 29/2 decays away to release the off-loading valve 29 mechanism. High pressure gas in the gas delivery hose 1 vents into gas chambers 15 and 12/2 via non return valve 16 to reduce the pressure to a point where the hose 1 can be safely removed without discharging the trapped high pressure gas to atmosphere.

25

The spool valve 8 is a shuttle-operated type whereby oil may flow from one port and return to the other port or vice versa. The direction of flow is determined by the position of the spool (not shown) inside the valve 8. This is a pressure operated bi-stable device. When the discharge pressure at port I reaches a set pressure, typically 100 bar, a relief valve 21 allows oil at this pressure to actuate the spool. This reverses the direction of flow through the outlet ports until the outlet pressure at port II reaches the pressure set by its relief valve 22, whereupon the flow reverses back to the original direction.

30

Low-pressure oil entering the spool valve 8 is returned back to the sump 5 for cooling via the cooling fan 23 and continuous supply to the pump 7 whilst the pump motor 4 is running.

5 The operation will be described with respect to the following eight stages.

(1)

High-pressure oil from the spool valve 8 flows via port I into the oil chamber 10 in hydraulic ram A and through chamber 10 to oil chamber 10/2 through dip-tube 24. This pushes pistons A (Left->Right in Fig. 1) and A/2 (R->L) and simultaneously pulls
10 piston B (L->R) and pushes piston B/2 (R->L) by means of piston sleeves 11 and 19.

(2)

Piston B moves so as to enlarge the volume 12 in the gas chamber B. This induces gas to enter the chamber B via the non-return valve 13 and low-pressure gas supply
15 (1 Bar). At the same time, piston B/2 moves so as to enlarge volume 12/2 in gas chamber B/2 to accommodate compressed gas forced out of chamber 15 via non-return valve 16 as piston A moves to decrease the volume in chamber 15. When pistons A and A/2 reach the end of their permissible strokes 18 and 18/2, respectively, the oil pressure to hydraulic ram A and A/2 rises rapidly, causing the
20 spool valve 8 to change the oil flow direction.

(3)

High-pressure oil now flows from the spool valve 8 via port II into hydraulic rams B (14) and B/2 (14/2). This pushes the pistons B (R->L in Fig. 1) and B/2 (L->R) and
25 simultaneously pulls piston A (R->L) and pushes piston A/2 (L->R) by means of the piston sleeves 11 and 19. Pistons A and A/2 move so as to enlarge the volumes in gas chamber 15 and 15/2 and reduce the oil volume in chambers 10 and 10/2, respectively, of hydraulic rams A and A/2. This causes low-pressure oil to flow back to the spool valve 8 at port I from hydraulic ram A.

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(4)

Piston A moves so as to enlarge the volume in the gas chamber 15. This induces gas to enter the chamber 15 via the non-return valve 13/2 from the low-pressure gas supply at 1 Bar. At the same time, piston A/2 moves so as to enlarge volume in gas

chamber 15/2 to accommodate compressed gas forced out of chamber B via non-return valve 16/2 as piston B moves to decrease volume 12. When pistons B and B/2 reach the end of their permissible strokes 18 and 18/2, respectively, the oil pressure to hydraulic ram B and B/2 rises rapidly, causing the spool valve 8 to change the oil flow direction.

(5)

Piston B moves to reduce volume in gas chamber 12, compressing the volume of gas induced on the previous stroke. The inlet non-return valve 13 prevents gas returning to the low-pressure gas supply. Likewise, the movement of piston B/2 reduces the volume in gas chamber 12/2 compressing the volume of gas induced on the previous stroke from chamber 15, forcing it to flow to discharge through non-return valve 25. The inlet non-return valve 26 prevents gas returning to the inter-cooling tube 27.

(6)

When both pistons B and B/2 reach the end of their permissible stroke 18 and 18/2, respectively, the oil pressure to hydraulic rams B and B/2 rises rapidly to 120 bar causing the spool valve 8 to change direction again. The reversed oil flow pushes pistons A and A/2 and reduces the oil volumes in hydraulic rams 14 and 14/2. This causes low-pressure oil to flow back to the spool valve 8 at port II from hydraulic ram B completing one cycle of the compressor.

(7)

Piston A moves to reduce volume in gas chamber 15, compressing the volume of gas induced on the previous stroke. The inlet non-return valve 13/2 prevents gas returning to the low-pressure gas supply. Likewise, the movement of piston A/2 reduces the volume in gas chamber 15/2, compressing the volume of gas induced on the previous stroke from chamber 12, forcing it to flow to discharge through non-return valve 25/2. The inlet non-return valve 26/2 prevents gas returning to the inter-cooling tube 27/2.

(8)

Pistons A and B move either to induce or compressing low-pressure and are thus a double-acting pair providing 1st stage gas compression to the 2nd stage gas compression A/2 and B/2, respectively. Their respective hydraulic oil and gas chambers are different in size to the 2nd stage double-acting piston pair A/2 and B/2.

5 The 1st stage hydraulic chambers of pistons A and B, (for 10 and 14), are formed with a piston sleeve 11, whilst the gas chambers 12 and 15 are formed in the conventional way with the piston crown moving in a piston cylinder. The 2nd stage hydraulic chambers, 10/2 and 14/2, and the gas chambers 12/2 and 15/2, are reversed in comparison to those of the 1st stage: thus, the hydraulic chamber is
10 associated with the piston crown and the gas chamber is formed with a piston sleeve 19. In the 1st stage, the gas swept volume 18 will always be greater than that of the hydraulic whilst the 2nd stage gas swept volumes 18/2 will always be less than that of the hydraulic.

15 For the two stage compressor to deliver 200 Bar gas to the fuel tank 2, typical compression ratios of 15:1 are required for the 1st and 2nd stage pistons which gives a theoretical delivery pressure of 225 Bar. However, because the 1st and 2nd stage piston pairs operate independently from a common hydraulic drive, the amount of compression at each stage is controlled by the following independent variables: 1st
20 stage piston bore and stroke, 2nd stage bore and stroke, and piston sleeve diameter 19.

In a 1:1 relationship of hydraulic (non-compressible) to Gas (compressible), the hydraulic volume requirement is 200 times that of the gas, and in the case of natural
25 gas typically 240 times. As the power requirement to drive the gear pump is equal to: Pressure*Volume flow, the motor power requirement will be proportional to pressure (200 Bar) and the fluid volume (240).

Hydraulic gear pumps are designed to deliver high pressure — in excess of 200 Bar.
30 Assuming the gear pump 7 is operated at a working pressure of 100 Bar, then 1st stage gas pressure of typically 15 Bar can be achieved with a hydraulic volume 15/100th of the gas volume. During the 2nd stage, the gas pressure is elevated from 15 Bar to 200 Bar which requires a larger hydraulic volume 200/100th of the gas volume to ensure the gear pump 7 continues to operate within the assumed working

pressure. The configuration of the compressor, as described, is therefore designed to achieve this.

To deliver 200 Bar gas to the fuel tank 2, typical geometry, for a gear pump 7 operating at 94 Bar, is given in Table 1. It should be noted that in this configuration, the gearing of hydraulic to gas throughput is 1:4.

	1 st Stage	2 nd Stage
Bore (mm)	120	50
10 Sleeve Diameter (mm)	110	35
Stroke (mm) 180 80	180	121.5
Compression Ratio	15.0	13.28
Hydraulic Pressure (Bar)	94.28	94.32
Gas Swept Volume (litres)	2.0	0.12
15 Hydraulic Swept Volume (litres)	0.325	0.24

Table 1

A feature of the compressor is that the 1st and 2nd stage pistons move independently of one another. Indeed, if one pair of pistons is not free to move, then the other pair is free to do so. In the case where the off-loading valve 29 dumps gas from the gas delivery hose 1 to the 1st stage chamber 15 there may not be sufficient hydraulic pressure to complete the 1st stage stroke because the initial charge of gas is too great. However, the 2nd stage piston is free to batch this excessive 1st stage gas through the system to the point where the 1st stage piston is free to operate as described.

Residual volume bias

Dead volumes or volumes not swept by the compressor pistons A, B, A/2, B/2 reduce the gas volumetric efficiency of the compressor which reduces the designed gas throughput. Dead volumes, internal ports and clearances, act as parallel volume to the main swept volume, 18 and 18/2. During the compression stroke, these parallel volumes take in gas only to release back to the compression cylinder on the gas intake stroke — reducing the intake gas charge. The resulting residual gas trapped in

the compressor can severely affect the gas throughput resulting in an over-sized compressor and motor 4 to deliver the desired throughput. In addition, the effect on throughput worsens at operating temperatures as the residual gas expands.

- 5 The hydraulic compressor sleeve 11 of the 1st stage can be biased to produce a higher hydraulic compression force on the 1st stage gas chambers 12 and 15 such it overcomes the residual gas pressure of the 2nd stage chambers (12/2 and 15/2) by delivering a higher compression pressure-thus to ensure a full 2nd stage intake gas charge. It should be noted that the 2nd stage gas swept volume is typically 6% of the
10 1st stage. As a result, the residual volume from gas port is likely to be a greater percentage and harder to deal with in the 2nd stage. The hydraulic bias technique allows this to be addressed without affecting compressor size.

Compressor valve operation

- 15 In comparison to reciprocating compressor technology, hydraulic compressor variants described herein are slow moving. Conventional reciprocating compressors rotate at typically 1500 rpm — a cycle time of 40 milliseconds. The hydraulic compressor variants have cycle times of typically 1 to 6 seconds. The resulting slow action necessitates that the valve technology (13, 13/2, 16, 16/2, 25, 25/2, 26 and
20 26/2) has gas pressure hold times up to 100 times longer than that of conventional compressor valve technology. As a result, operational compressor valves have been designed specifically to suit the hydraulic application described.

- Referring to Fig. 2(a), the valve assembly comprises: a valve seat 40, an O-ring seal
25 41, a valve-sealing member 42 and a valve retaining sleeve 43. The valve-retaining sleeve 43 is attached to the valve seat 40 whilst the valve sealing member 42 is free to glide within a cavity that limits the seal member lift 42/2 and is of an aspect ratio designed to prevent jamming of the seal member 42 in the cavity. The valve seal is achieved by means of a 'disc-on-seat' mechanism brought about by two ground flat
30 faces; 48 and 49 which mate when the valve seal member 42 glides within its cavity to contact the valve seat 40. Tight gas shut-off is achieved by reverse gas differential pressure acting on the sealing member 42 such that the higher the pressure acting on the sealing member 42 the higher is the force of closure which compresses faces 48 and 49 tightly together.

It was found that it was almost impossible to achieve the holds required with conventional pop-pit valves without resorting to the aid of purpose design seals, which in an operational compressor would severely limit their working life. The ground face valve proved to be very successful with metal to metal finishes on both the valve seat 40 and valve seal member 42 — ensuring a robust design well able to cope with the working environment of an operational compressor valve of temperature, pressure and repetitive use. In practice, it is not possible to manufacture a perfect valve — but the disc-on-seat approach proved to be far more tolerant to the vagaries of manufacture. The key parameters which ensure a good quality valve able to operate over the gas pressure range described are as follows.

- (a) The valve sealing faces must be ground flat with no gullies or troughs — lapping aids this process
- (b) It is not necessary to ensure that all the surface marks resulting from the grinding process are eliminated. The two mating faces (46 and 47) are ground to ensure a labyrinth of crisscrossing grooves of sufficient area that discourage gas from escaping.

The valve sealing ability can be further improved by manufacturing the valve-sealing member 42 from modern polymer materials such as PEEKTM polyetheretherketone resin. This has the mechanical properties that suit the applicant whilst its increased elasticity can provide a tighter gas shut-off. Its additional advantages are as follows.

- (a) The force required to separate the two ground surfaces (stiction) is significantly reduced using a PEEKTM sealing member 42 compared to metal seal — further improving the gas intake charge.
- (b) A PEEKTM sealing member 42 is much lighter than metal one, reducing the inertia force required to move the sealing member 42, which is operationally desirable as the sealing member 42 continually shuttles in sympathy with the motion of the compressor.
- (c) PEEKTM material can be moulded which aids the manufacturing process and improves the valve finish.

The valve assembly shown in Figure 2(a) can either operate as an intake valve (13, 13/2, 26, 26/2) or as an inter-stage exhaust valve (16, 16/2) — the valve size and lift

42/2 are sized to suit the gas flow requirement. To ensure high volumetric efficiencies, the valve has been designed without a spring closure mechanism to keep the valve-sealing member in contact with the ground face 49. The valve-sealing member is therefore free to respond to differential gas pressure at compressor ports

46 and 47 which has two distinct advantages, as follows.

(a) It ensures that the intake cylinder 12 and 15 is fully charged to gas line pressure and is not reduced by the differential gas pressure required to compress a closure spring acting on the sealing member 42 — typically improving the gas charge by 10%.

(b) Similarly, during the inter-stage exhaust stroke a closure spring would act on the sealing member 42 to add to the amount of gas compression required to drive gas out of the compression chamber, increasing the required hydraulic force.

Once the sealing member 42 has moved away from the seat 40 its tendency is to block the gas exit port 46. However, the sealing member cavity prevents this. In addition, the valve sleeve 43 has a reduced diameter, which allows gas to route out into the annular gap and back into the sleeve 43 around the valve-sealing member 42.

Figure 2(b) depicts the closure of the valve. Without a spring to force the sealing member 42 closed at the end of the intake stroke, it is essential to provide an alternate closure mechanism. This mechanism must close the sealing member 42 and ensures that the sealing surfaces 48, 49 are brought together rapidly at the start of the compression stroke.

As is seen in Fig. 2(b), at the start of a gas compression stroke, gas is forced on to the rear surface of the sealing member 42, through port 46, in preference to the annular gap route described. The gas dynamics are therefore configured in such a way as to force the sealing member to rapidly close and block the escape of gas back through the system.

Figure 3 depicts the operation of the compressor exhaust valve in the compressor of Fig. 1. In certain embodiments, the valve design may include the use of a closure

spring to provide a positive closing force on the sealing member 42. Figure 3 shows the arrangement where the sealing member 42 has been modified to include a spring cavity 44 to accommodate a closure spring 45. Whilst the arrangement can be used as an inter-stage valve, it is more specifically designed to provide a positive seal for the 2nd stage exhaust valves 25 and 25/2. The arrangement maintains the aspect ratio of the valve-sealing member 42 to avoid jamming of the seal in its valve-sleeve cavity, which is modified to accommodate the spring 45. In addition, the valve seat 40 has been modified in length so that the overall valve profile remains the same for both types of valve.

High-pressure limit and control

The hydraulic compressor is designed such that it can be controlled from the low-pressure gas supply (1 Bar) to avoid over-pressure sensing of high-pressure gas in the gas delivery line 1. This avoids compromising the high-pressure gas line with the inter-connection of a high-pressure electrical transducer — this is the conventional means of limiting the final pressure (200 Bar).

In the hydraulic compressor described, the hydraulic spool valve 8 sets the final gas pressure via relief settings of valves 21 and 22. These settings are safeguarded by means of an independently set spillback safety valve 9 and the compressor will continue to reciprocate without producing gas throughput when these hydraulic pressure limits are reached. Therefore, the compressor can be controlled by the combined low-pressure gas flow to compressor intake valves 13 and 13/2 whereby gas flow signals that the desired outlet pressure (200 Bar) has not been reached.

Modern low-pressure gas meters of the domestic type, have an electrical output that is proportional to gas flow. Of the two common types, the U6 (0-6 SCMH) has a switched or pulsed output whilst the E6 (0-6 SCMH) has an optical link that can be decoded to obtain the gas flow. To avoid the cramping of the hydraulic settings at the desired gas pressure of 200 bar, the compressor control system is illustrated by way of example for a compressor design to deliver 2 SCMH:

- (a) Assuming a typical domestic U6 gas meter with a pulse output for every 30 standard cubic litres of gas.

- (b) The compressor when pumping at design conditions will cause the U6 gas meter to produce a pulse every 1.1 minutes.
- (c) To allow for reduced volumetric efficiencies of the hydraulic compressor at high delivery pressures the compression operation could be terminated at for instance 50% of design (1.0 SCMH), which would generate a pulse on the U6 meter every 2.2 minutes.
- (d) Conversely, to guard against a low-pressure supply line failure or leakage, an upper limit may be set, for instance 25% above design (2.5 SCMH), which would generate a pulse on the U6 meter every 53 seconds.
- (e) The compressor control module has therefore been configured to accept a start signal and following a permitted start-up delay to only maintain the compressor operation via the electric motor 4 provided the intake gas flow is within its permissible limits. For the example given: a flow pulse received in a period not less than 53 seconds and not greater than 132 seconds. Flow pulses outside this time frame de-energise the electric motor 4 via its trip relay switch.

Safe Operation

The hydraulic compressor (Fig. 1) can be configured such that when pumping flammable gases, the gas carrying components are operated through remote hydraulic lines to ensure their intrinsic safety from proximity to source of potential ignition, i.e. electric motor 4 and control module 54. However, a further embodiment of the compressor is shown in Fig. 4, whereby the whole arrangement is shown as a freestanding appliance within a casing 53. The compressor is attached via wall mountings 60 and includes the compressor stages 62 and hydraulic chamber 64. The machine is fed by a low pressure (LP) gas supply 66. The hydraulic tank 68 is disposed at the back of the device, adjacent suspension mounts 70.

Figure 4 depicts the airflow through the appliance, which is driven by fan 23, with air intake 50 and air outlet 51. Where the compressor is used to pump flammable gases, the airflow serves two purposes:

- (a) Heat generated during gas compression is transferred to the hydraulic oil and subsequently removed by the airflow through the appliance.

- (b) The airflow through the appliance is designed to make safe potential gas leakages by dilution to non-flammable mixtures to neutralise potential sources of ignition, e.g. the electric motor 4 and the control module 54.

- 5 Gas leakage from the high-pressure delivery system. The compressor gas throughput is constant by design. Therefore, potential gas leakage from high-pressure components, e.g. the fuel line 1, cannot exceed the design flow-rate of the compressor, irrespective of the delivery pressure to the fuel tank 2 (not shown).
- 10 This can be illustrated by contriving a gas leakage orifice size that at the design pressure of 200 Bar, leaks gas at the design flow rate from a high pressure component. Under these conditions, the system is balanced in that all the gas compressed passes through the contrived leakage orifice. If the leakage orifice is any greater in size than that contrived, the high-pressure component containing the gas
- 15 leak, will vent and the pressure will decay to 1 Bar (atmospheric) whilst the gas leakage rate will remain the same. Conversely, if the leakage orifice is smaller in size than that contrived the leak rate from the high-pressure component containing the gas leak will fall whilst the pressure remains the same (200 Bar). Thus, the worst-case potential high-pressure gas leakage to atmosphere cannot exceed the
- 20 compressor designed gas throughput at the designed gas pressure.

- In atmosphere, gas dispersion calculations can predict separation distances of potential sources of ignition (e.g. electrical switch-gear) from potential sources of gas leakage — e.g. high-pressure joints. For instance, the separation distance for 200
- 25 Bar high-pressure components (Fig. 1), pumping natural gas and having their gas flow limited by design to 0.5 SCMH is not less than 10 cm — compared to 2m for conventional high pressure components assumed to have unlimited flow capability.

- The dispersion of gas, however, cannot be guaranteed inside the compressor casing arrangement 53 even though it may be possible to achieve 'in atmosphere' gas
- 30 dispersion safe distances from potential sources of ignition. As a result, the appliance relies upon the airflow to dilute potential worst-case flammable gas leakages to non-flammable mixtures, e.g. a safe margin below the LEL — Lower Explosive Limit of the flammable gas in air.

The amount of cooling airflow required to maintain good compressor performance through gas inter-cooling 27 and 27/2 generally exceeds the design gas throughput. For a compressor with a designed natural gas pumping rate of 0.5 SCMH, the cooling airflow through the appliance, driven by the motor fan 23 was measured at 76.9 SCMH. Assuming a worst-case gas leakage, this equates to 0.65% natural gas in air or 13% of the LEL for natural gas.

Mechanism to ensure safe operation. When pumping flammable gasses, the compressor is made safe from gas leakage via two basic mechanisms:

- (a) As described above — there is sufficient cooling air to guarantee dispersion of gas leakage to a safe margin below the LEL.
- (b) Once the compressor stops, high-pressure gas trapped in the compressor components is simultaneously vented to low pressure through the off-loading valve 29. The off-loading valve 29 is inextricably linked to the operation of the compressor through the hydraulic drive. This results in the appliance reverting to a conventional domestic low-pressure appliance whilst not in use with natural ventilation — air intake port 50 and outlet/vent 51.

The appliance therefore is intrinsically safe provided the air intake 50 is connected to an unlimited supply of fresh air, which is not contaminated by outlet air 51. Furthermore, the appliance can be operated indoors provided it is room-sealed — in that the air intake 50 and air outlet 52 are connected to the outside so that to prevent the potential build-up of gas in the building. However, it must be stressed that high pressure joints external to the appliance and the refuelling of the tank 2 via break-away coupling 28, flexible refuelling hose 1 and quick-release coupling 3 are not part of these in-door safety arrangements and must be subject to separate safety considerations on a case-by-case basis.

A further embodiment of the mechanism to ensure safe operation in situations where it is not possible to ensure adequate natural internal ventilation whilst the appliance is not in use is as follows.

The air fan 23 can be arranged to be a freestanding mechanism external to the appliance whereby the electrical supply to the potential sources of ignition, electric motor 4 and control module 54 is remotely isolated. The start-up procedure is as follows.

- 5 (a) Energise the external fan 23 to drive the required airflow through intake 50 and outlet 51.
- (b) Pre-purge: prove by some means that the required air is flowing and allow typically 5 air volumes changes of the internal casing to remove any potential build-up of flammable gasses.
- 10 (c) Energise the remote electrical isolation to power the compressor.

Claims:

1. A two stage gas compressor, comprising:

a first compressor stage, comprising a first chamber, a first piston and a second piston, the first and second pistons being rigidly connected by a first connecting member, thereby defining a first volume opposite the first piston, a second volume opposite the second piston and a third volume between the first and second pistons;

a second compressor stage, comprising a second chamber, a third piston and a fourth piston, the third and fourth pistons being rigidly connected by a second connecting member, thereby defining a fourth volume opposite the third piston, a fifth volume opposite the fourth piston and a sixth volume between the third and fourth pistons;

at least one hydraulic conduit for conveying hydraulic fluid to said first and second compression stages; and

at least one gas conduit for conveying said gas between said first compressor stage and said second compressor stage.

2. The compressor of claim 1, wherein the first connecting member is movable between a first position, where said first volume is a maximum and said second volume is a minimum, and a second position, where said first volume is a minimum and said second volume is a maximum.

3. The compressor of claim 1 or 2, wherein the second connecting member is movable between a first position, where said third volume is a minimum and said fourth volume is a maximum, and a second position, where said third volume is a maximum and said fourth volume is a minimum.

4. The compressor of claim 3, wherein, in use, the first connecting member is in its first position when the second connecting member is in its second position, and vice versa.

5. The compressor of any of the preceding claims, wherein said at least one hydraulic conduit includes a first hydraulic conduit arranged to convey hydraulic fluid from said first compression stage to said second compression stage.

5 6. The compressor of claim 5, wherein said first hydraulic conduit is arranged to convey hydraulic fluid between said third volume and said fourth and fifth volumes.

7. The compressor of claim 5 or 6, wherein said at least one hydraulic conduit includes a second hydraulic conduit arranged to convey hydraulic fluid from a
10 pumped supply of hydraulic fluid to said first compression stage.

8. The compressor of claim 7, further including a bistable shuttle valve between said a pumped supply of hydraulic fluid to said first compression stage.

15

9. The compressor of any of the preceding claims, wherein said at least one gas conduit includes a first gas conduit arranged to convey pressurised gas from said first volume to said sixth volume when said first connecting member moves between its first position and second position.

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10. The compressor of any of the preceding claims, wherein said at least one gas conduit includes a second gas conduit arranged to convey pressurised gas from said second volume to said sixth volume when said first connecting member moves between its second position and first position.

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11. The compressor of any of claims 2 to 10, wherein said at least one gas conduit includes a gas supply conduit arranged to convey unpressurised gas

(a) to said second volume when said first connecting member moves between its first position and second position, and/or

30 (b) to said first volume when said first connecting member moves between its second position and first position.

12. The compressor of any of the preceding claims, including a first sealing member sealingly dividing said third volume into a first section and a second section.

13. The compressor of any of the preceding claims, including a second sealing member sealingly dividing said sixth volume into a third section and a fourth section.

- 5 14. A compressor substantially as hereinbefore described with reference to the accompanying drawings.

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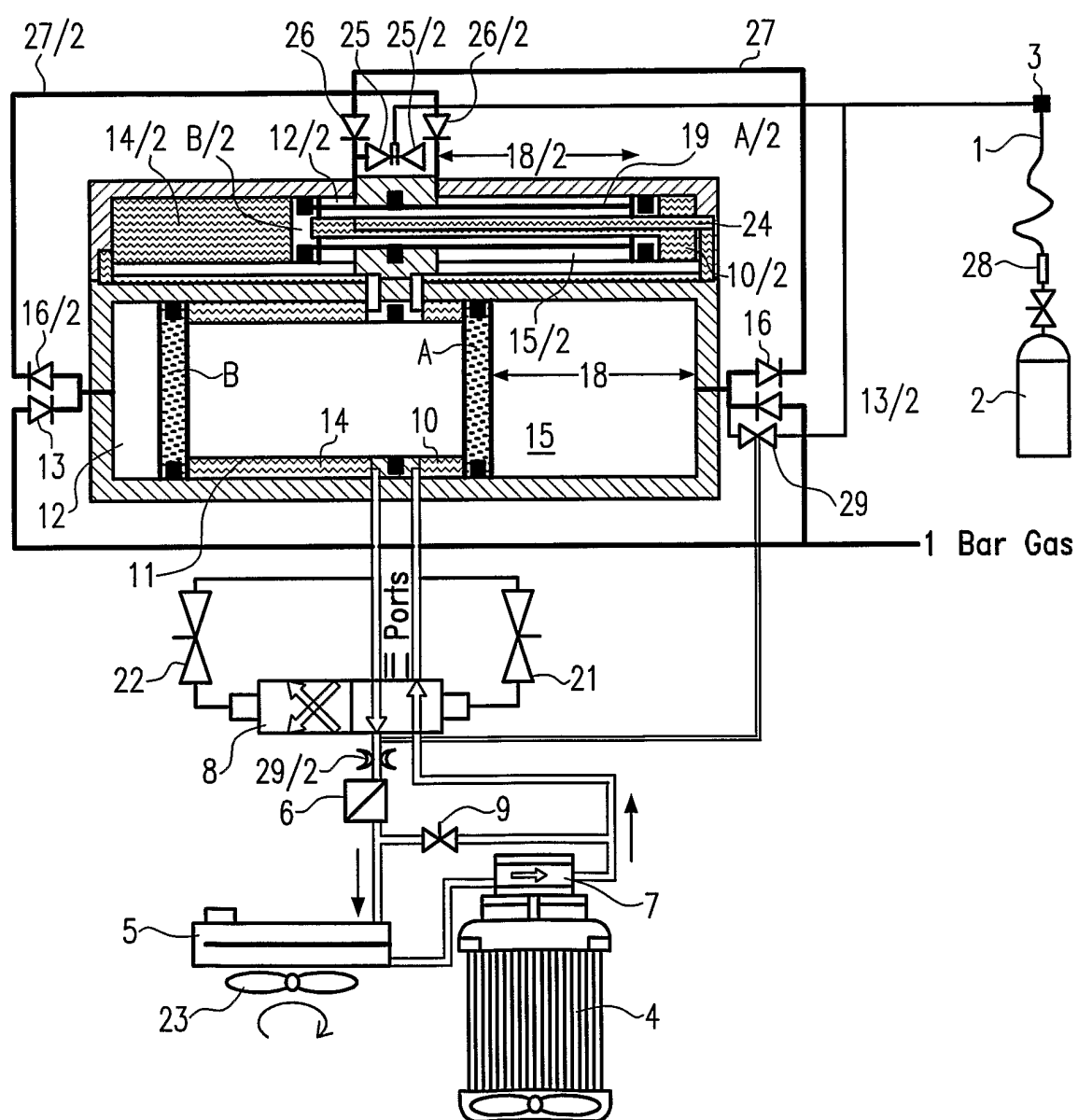


Fig.1

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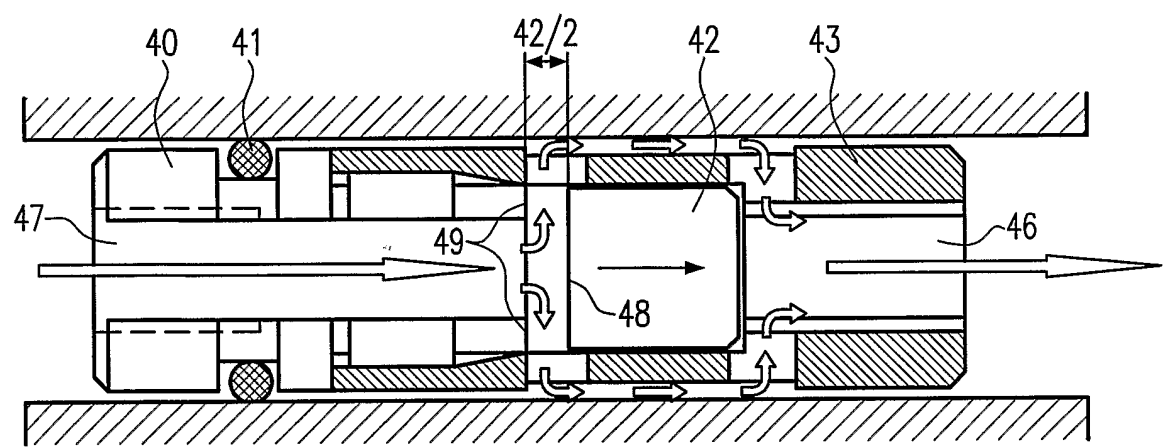


Fig.2(a)

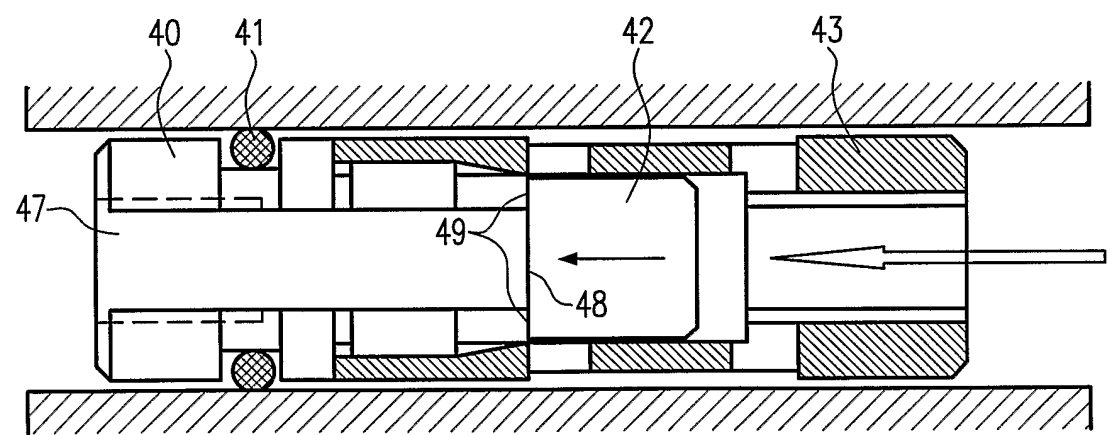
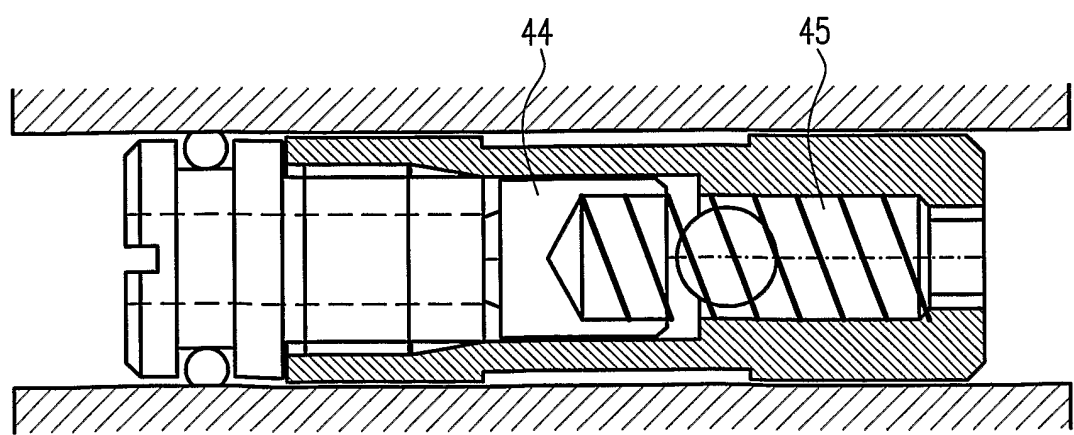


Fig.2(b)



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Fig.3

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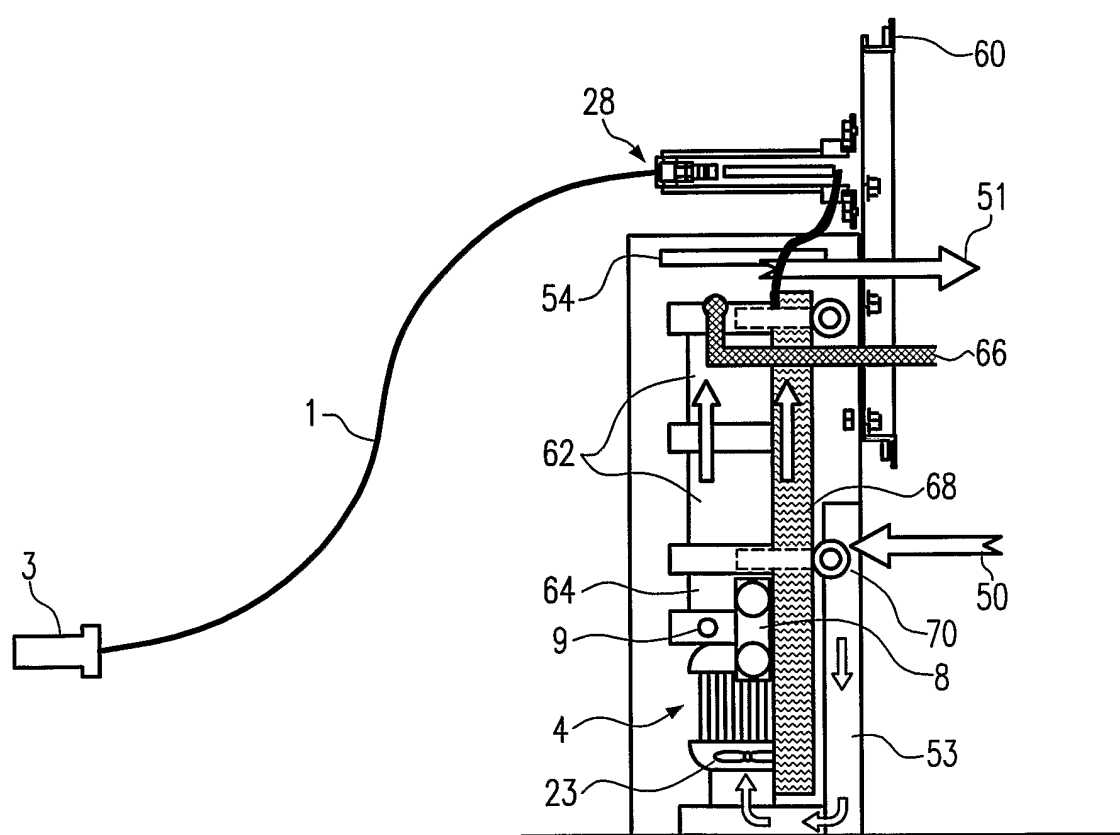


Fig.4

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