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(54) Expansion valve

(57) A damper chamber 23 of an expansion device is formed by a cylinder 16 in a housing 10 defining a valve seat 14. A piston 17 is integral with a valve element 15. When the valve element 15 undergoes a sudden pressure change, the volume of a damper chamber 23 changes to accommodate sudden motion of the valve element 15, whereby vibrations of the valve element 15 are suppressed, to reduce generation of untoward noise. The ratio (b/a) between the valve element diameter (b) and a port diameter (a) of a valve hole (13) is set to be less than 1.5 so as to suppress a valve element suction phenomenon and to establish a flow rate corresponding to the value of a differential pressure between primary and secondary pressures.

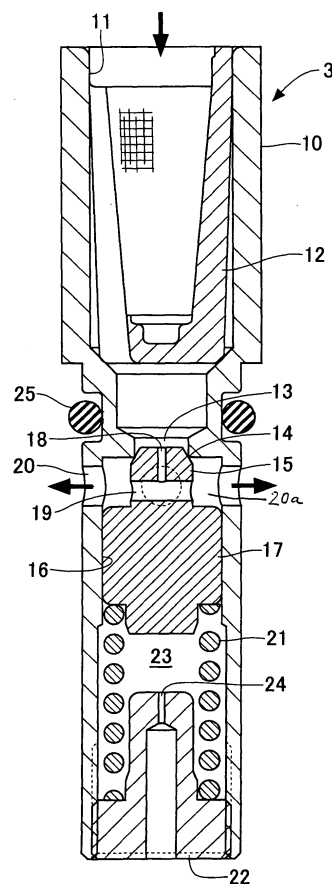


FIG. 2

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Description

[0001] The invention relates to an expansion device according to the preamble of claim 1, and particularly for an automotive air conditioner using carbon dioxide (CO₂).

[0002] Generally, a refrigeration cycle for an automotive air conditioner comprises a receiver for separating condensed refrigerant into gas and liquid, and a thermostatic expansion valve for expanding the liquid refrigerant. Other known refrigeration cycles employ an orifice expansion device for throttling and expanding condensed refrigerant, and an accumulator for separating evaporated refrigerant into gas and liquid. The orifice expansion device orifice tube does not control the flow rate. The thermostatic expansion valve operates with a variable orifice controlling the flow rate and functions as a differential pressure valve the valve element of which is spring loaded in valve-closing direction. When the differential pressure is small, the valve will close, whereas when the differential pressure exceeds a predetermined value, the valve will open to reduce the differential pressure. When the valve element moves in valve-closing direction, the differential pressure rises and tends to move the valve element in valve-opening direction. The differential pressure valve repeatedly performs this operation, and hence the valve element oscillates in opening/closing directions, which can cause untoward noise. Particularly with CO₂ as refrigerant, since a very small valve element stroke is controlled by a very large differential pressure, when the pressure rises and falls sharply, it is difficult to immediately stop the valve element at a balanced position. Therefore, the valve element inevitably tends to oscillate thereby generating untoward noise. When the valve element vibrates or oscillates, the flow rate increases, and the vaporization temperature of the refrigerant in the evaporator becomes high, so that the temperature of air which has passed through the evaporator and is blown into a passenger compartment also becomes high. The instability of the valve operation can cause hunting of the refrigeration cycle, which makes the temperature of blown-in air unstable.

[0003] In an expansion device known from JP 2004-218918 A, published August 8, 2004, Fig. 4, a vibration proof spring is mounted on the valve element such that the valve element slides while the vibration proof spring is pressed against an inner housing wall surface. Thus, the motions of the valve element in opening/closing directions is restricted by increased sliding resistance to suppress vibration of the valve element, and to prevent generation of untoward noise. The vibration proof spring increases the frictional force at the wall and produces a large hysteresis in the valve flow rate characteristic. The valve element may get displaced from a position set as an optimum position, causing degradation of the efficiency of the refrigeration cycle. When applied to a refrigeration cycle using CO₂, the valve element diameter is larger than the port diameter of the valve hole to fully close

the valve hole when the differential pressure is small. A lap margin between the valve hole and the valve element thus becomes large. When high-pressure CO₂ passes through the orifice between the valve hole and the valve element, the flow velocity increases and causes a valve element suction phenomenon. The valve element tends to move in valve-closing direction and makes the flow rate smaller than a set flow rate, which makes it impossible to obtain sufficient cooling power.

[0004] It is an object of the invention to provide an expansion device which both reduces generation of untoward noise and suppresses the occurrence of a valve element suction phenomenon.

[0005] The object is achieved by the features of claim 1.

[0006] With the help of the damper means, it is possible to suppress vibration of the valve element in opening or closing directions, thereby reducing generation of untoward noise, and to prevent that the flow rate increases due to a vibration of the valve element, to thereby suppress an undesirable rise of the temperature of the blown air. The damper means also prevents an occurrence of a hunting effect, which makes it possible to stabilise the temperature of the blown-in air.

As the damper means is a measure for preventing generation of untoward noise, and also allows to reduce hysteresis more dramatically than by the known measure utilising increased mechanical sliding resistance, and to thereby obtain stable characteristics and to operate the refrigeration cycle efficiently.

[0007] Moreover, the valve element diameter is selectively set to be close to the port diameter, such that the ratio between the valve element diameter and the port diameter is set not larger than 12.5, with the effect that, surprisingly, valve element suction phenomenon effects can be suppressed. As a result, the characteristics of the expansion device can be set to some extent by calculation based on only the balance between the pressure received by the valve element and the load of the spring, which greatly facilitates adjustment of the characteristics. Further, due to the suppression of occurrence of the suction phenomenon, it is possible to prevent an accidental reduction of the flow rate of refrigerant, thereby securing the cooling power.

[0008] Embodiments of the invention are explained referring to the drawings.

Fig. 1 is a schematic view of a refrigeration cycle containing an expansion device,

Fig. 2 is a longitudinal section of the expansion device,

Fig. 3 is an enlarged section of essential parts of a differential pressure valve,

Fig. 4 is a diagram showing changes in the suction force acting on a valve element of the differen-

tial pressure valve,

Fig. 5 is a longitudinal section of another expansion device, and

Fig. 6 is a diagram showing a valve-opening characteristic.

[0009] A refrigeration cycle for an automotive air conditioner using CO₂ as refrigerant comprises in Fig. 1 a compressor 1, a gas cooler 2 for cooling compressed refrigerant, an expansion device 3 for adiabatically expanding the cooled refrigerant, an evaporator 4 for evaporating the adiabatically expanded refrigerant, an accumulator 5 downstream of the evaporator 4 for storing surplus refrigerant, and an internal heat exchanger 6 for cooling the refrigerant cooled by the gas cooler 2, using refrigerant delivered from the accumulator 5 to the compressor 1.

[0010] The expansion device 3 is disposed within a hollow cylindrical body 7. The body 7 has an upstream-side end connected to a pipe extending from the internal heat exchanger 6, and a downstream-side end connected to a pipe extending toward the evaporator 4.

[0011] The operation of a refrigeration cycle using CO₂ is substantially the same as when using chlorofluorocarbon. The compressor 1 sucks in gaseous-phase refrigerant from the accumulator 5, and discharges after compression the high-temperature, high-pressure refrigerant in a gaseous phase or supercritical state. The discharged refrigerant is cooled by the gas cooler 2 and supplied via the internal heat exchanger 6 to the expansion device 3. In the expansion device 3, the refrigerant is adiabatically expanded to have the phase state changed from the liquid phase state to a two-phase state of low-temperature, low-pressure gas and liquid. In the evaporator 4, the refrigerant in the two-phase gas-liquid state is evaporated by air within a vehicle compartment, to cool the air by depriving the air of latent heat of vaporization. The evaporated refrigerant is supplied into the accumulator 5 and is temporarily stored. A gaseous-phase portion is returned from the accumulator 5 via the internal heat exchanger 6 to the compressor 1. In the case of CO₂, the internal heat exchanger 6 further cools the high-temperature refrigerant cooled in the gas cooler 2, by the low-temperature refrigerant delivered from the accumulator 5 to the compressor 1, or further heats the low-temperature refrigerant from the accumulator 5 to the compressor 1, by the high-temperature refrigerant from the gas cooler 2.

[0012] The expansion device 3 in Fig. 2 has a hollow cylindrical housing 10 with an upper open end forming a primary-side refrigerant inlet port 11, where a strainer 12 is fitted. An axially central housing portion is formed with a valve hole 13, the lower peripheral edge of which forms a valve seat 14. A valve element 15 is disposed below the valve seat 14. The valve element 15 is integral with a coaxial piston 17 axially slidably fitted in a housing cyl-

inder 16. The valve element 15 contains a fixed orifice 18 communicating with a horizontal through hole 19 extending through a valve element tip perpendicular to the axis of the housing 10. The dimension of the fixed orifice 18 allows a minute amount of refrigerant to pass through when the valve element 15 is seated on the valve seat 14, to enable a circulation of a minimum amount of lubricating oil dissolved in the refrigerant, as required for the lubrication of the compressor 1.

[0013] A secondary-side chamber 20a communicates with a refrigerant outlet port 20 of the housing 10. The piston 17 is urged by a spring 21 in valve-closing direction. The valve seat 14 and the valve element 15 jointly form a differential pressure valve operated by the balance between the differential pressure between a primary pressure on the upstream side of the valve hole 13 and a secondary pressure on the downstream side plus the load of the spring 21. The spring 21 is received by an adjustment screw 22 screwed into the lower end of the housing 10. The load of the spring 21 may be adjusted by adjustment screw 22.

[0014] A space defined by the housing 10, the piston 17, and the adjustment screw 22 forms a damper chamber 23 communicating with the secondary side of the expansion device via a fixed orifice 24 in the adjustment screw 22. Refrigerant flows via a clearance between the housing 10 and the piston 17 into the damper chamber 23 and the fixed orifice 24 to the secondary side of the expansion device. An O-ring 25 seals between the primary side and the secondary side when the expansion device 3 is inserted into the body 7 shown in Fig. 1.

[0015] When the differential pressure between the refrigerant inlet port 11 and the refrigerant outlet port 20 is smaller than a predetermined value as determined by the load of the spring 21, the valve element 15 is seated on the valve seat 14 to close the expansion device. Refrigerant flows from the refrigerant outlet port 20 through the fixed orifice 18 at a minimum necessary flow rate.

[0016] When the primary pressure received by the valve element 15 increases, the valve element 15 is moved away from the valve seat 14 to thereby place the expansion device in the open state. The primary-side refrigerant flows to the secondary side via a variable orifice defined between the valve seat 14 and the valve element 15. The high-temperature, high-pressure gaseous-phase refrigerant is adiabatically expanded into low-temperature, low-pressure refrigerant in a gas-liquid mixture state flowing out from the refrigerant outlet port 20. The valve element 15 is lifted to a position where the differential pressure between the primary and secondary pressures and the load of the spring 21 are balanced, and stops at that position. The expansion device passes refrigerant at a flow rate corresponding to the differential pressure between the primary and secondary pressures. The valve element 15 is able to move without large sliding resistance in accordance with any changes of the primary pressure. When the primary pressure is changing gently, it is possible to reduce hysteresis of the flow rate char-

acteristics.

[0017] When the pressure of the refrigerant inlet port 11 rises sharply, the valve element 15 is urged to move quickly in valve-opening direction. However, the piston 17 tends to move in valve-opening direction to reduce the volume of the damper chamber 23. The pressure within the damper chamber 23 rises and acts to return the piston 17 in valve-closing direction. This restricts a sudden valve-opening motion of the valve element 15 to prevent that the valve element 15 moves in direct accordance with the sharp primary pressure rise. Thereafter, the increased pressure within the damper chamber 23 is progressively released through the clearance between the cylinder 16 and the piston 17 and the fixed orifice 24 such that the force returning the piston 17 in valve-closing direction decreases or is lost. When the pressure in the refrigerant inlet port 11 falls sharply, the valve element 15 and the piston 17 operate opposite to the above case, and for the same reasons, the valve element 15 cannot move in the valve-closing direction in direct accordance with the sharp primary pressure drops. Whenever the primary pressure sharply rises or drops, sudden motions of the valve element 15 will be suppressed, thereby preventing vibrations of the valve element, and dramatically reducing generation of untoward noises.

[0018] In the expansion device in Fig. 3 the variable orifice is formed by the valve seat 14 of the housing 10 and by the valve element 15. The valve element 15 has a diameter "b" larger than a port diameter "a" of the valve hole 13 so that the differential pressure valve can be fully closed, and on the secondary side of the variable orifice, there exists a lap margin of width or the diameters (b - a).

[0019] When high-pressure CO₂ passes through the variable orifice at high speed, turbulences occur on the secondary side tending to cause a suction phenomenon acting on the valve element 15 such that the valve element is drawn toward the valve seat 14. The occurrence of the suction phenomenon moves the valve element 15 in valve-closing direction, and hence when the valve lift is determined depending on the value of the differential pressure, the flow rate becomes lower than the flow rate which would correspond to the value of the differential pressure.

[0020] It turned out that the suction phenomenon is largely related to the ratio (b/a) between the valve element diameter "b" and the port diameter "a". Fig. 4, in which the abscissa represents the ratio (b/a) of the valve element diameter "b" to the port diameter "a", and the ordinate represents the suction force, shows that the suction force is small when the ratio (b/a) is low, and as the ratio (b/a) is higher, the suction force is larger. Presumably, this is because when the ratio (b/a) is low, the lap margin is small, and hence the influence of turbulences of the refrigerant flow upon the valve element 15 is small. It is apparent from Fig. 4 that when the ratio (b/a) of the valve element diameter "b" to the port diameter "a" is not higher than 1.5, the suction force is small, and the effect of the suction phenomenon on the valve element 15 is

slight. In the present preferred embodiment, the valve element diameter "b" is selected close to the port diameter "a" to reduce the lap margin, whereby the ratio (b/a) of the valve element diameter "b" to the port diameter "a" is set to about 1.16.

[0021] Since then the effect of the suction phenomenon on the valve element operation is suppressed, i.e. the attraction to the valve seat 14 due to the flow of refrigerant is weak, it is possible to set the expansion device characteristics to some extent by calculation based on only the balance between the pressure received by the valve element 15 and the load of the spring 21. Further, since the motion of the valve element 15 in valve-closing direction due to the suction phenomenon is prevented during operation of the expansion device, it is possible to maintain the refrigerant flow at a predetermined flow rate and to prevent that the cooling power of the refrigeration cycle becomes insufficient.

[0022] The expansion device 3 described above operates by a variable orifice having a characteristic such that when the valve is closed, refrigerant flows through the fixed orifice 18 in the valve element 15, and such a characteristic that after the valve is opened, the valve element 15 is progressively lifted according to the differential pressure, i.e. a variable orifice having a single characteristic change point.

[0023] Next, an expansion device will be described which has two characteristic change points, i.e. three stages of characteristics. This enhances the degree of freedom in configuring the settings of the expansion device when it is applied to a refrigeration cycle. The expansion device 3a in Fig. 5 is provided with a stopper 26 which is screwed into the adjustment screw 22. The stopper 26 is axially adjustable in the adjustment screw 22 and serves to restrict the stroke of the piston 17 in valve-opening direction to in turn restrict the maximum lift amount of the valve element 15. The maximum lift amount can be determined by adjusting the screw-in amount of the stopper 26 after the load of the spring 21 is adjusted by the adjustment screw 22.

[0024] The stopper 26 has a hollow part which is open to the secondary side. Similar to the refrigerant outlet port 20, the fixed orifice 24 for the damper chamber 23 is formed in an upper end of the stopper 26 opposed to the piston 17. The piston 17 and/or the stopper 26 has in the opposed end faces e.g. a groove intersecting the axis of the fixed orifice 24 so as to prevent the fixed orifice 24 from being closed when the piston 17 abuts at the stopper 26.

[0025] When the piston 17 reaches the valve-opening stroke determined by the screw-in position of the stopper 26, a further stroke of the piston 17 is inhibited. Thus, two change points of the valve-opening characteristic can be set as illustrated in Fig. 6.

[0026] In Fig. 6 the abscissa represents the differential pressure across the valve element 15, and the ordinate represents an opening area formed when the valve element 15 is lifted, which is indicated by an opening diam-

eter as the diameter of a circle having the same area as the opening area. When the differential pressure is small and the valve is closed, even if the differential pressure changes, a constant opening diameter is maintained by the fixed orifice 18. When the differential pressure has risen to a valve-opening point (e.g. 3MPa in the diagram) determined by the adjusted load of the spring 21 the differential pressure valve starts to open, whereafter the differential pressure valve has a characteristic that the opening diameter changes according to the differential pressure. Then, with a further rise of the differential pressure, the valve element 15 is progressively lifted until the piston 17 abuts at the stopper 26 (e.g. 6MPa in the illustrated example). The valve element 15 cannot move any further in valve-opening direction even if the differential pressure further rises, so that the differential pressure valve has a characteristic that the opening diameter does not change further even when the differential pressure rises. The change point on the high differential pressure side can be changed by adjusting the stopper 26 accordingly.

[0027] In the expansion device 3a, the two change points of the valve-opening characteristic can be adjusted separately by the adjustment screw 22 and the stopper 26. This allows to adjust the two change points for the case where the expansion device 3a is applied to a certain refrigeration cycle, and to tailor the characteristic of the expansion device 3a to a point desirable in terms of system efficiency, so that the degree of freedom in configuring the settings of the characteristic of the expansion device 3a can be enhanced. Further, after the differential pressure has exceeded a predetermined value, the opening diameter no longer increases with a rise in the differential pressure, and hence excessive flow of refrigerant is prevented in an operational region of large differential pressure, which makes it possible to adjust the characteristic of the expansion device 3a to a region ensuring excellent efficiency.

[0028] The respective adjustment member for adjusting the load of the spring 21, instead may be constituted by a member press-fitted into the open end of the cylinder 16. In this case, the load of the spring 21 can be adjusted by the press-fitting depth amount. Similarly, the stopper 26 instead may be press-fitted into the adjustment screw 22 or member such that the maximum stroke position of the piston 17 in valve-opening direction can be adjusted by the press-fitting depth amount of the stopper 26.

Claims

1. An expansion device (3, 3a), in particular for a refrigerating cycle of an automatic air conditioner, the expansion device (3, 3a) having a pressure-dependent movable valve element (15) in a refrigerant flow path between primary and secondary sides downstream of a valve seat (14) in a state urged in valve-closing direction by a spring (21), the expansion de-

vice being capable of passing refrigerant at a flow rate corresponding to a differential pressure between a primary pressure in a refrigerant inlet port (11) and a secondary pressure in a refrigerant outlet port (20), **characterised in that** damper means (23, 17, 16, 24) are associated to the valve element (15), for accommodating motion of the valve element (15) in opening and/or closing directions when the valve element (15) undergoes a sudden pressure change.

2. The expansion device according to claim 1, **characterised in that** the damper means (23, 17, 16, 24) comprises a cylinder (16) in a housing (10) formed with the valve seat (14), that the cylinder (16) extends coaxially with the valve element opening and/or closing directions, that a piston (17) is slidably disposed in the cylinder (16) and is integrally formed with the valve element (15), and that a damper chamber (23) is provided the volume of which can be changed by the piston (17).
3. The expansion device according to claim 2, **characterised in that** the damper chamber (23) is bounded opposite to the piston (17) by an adjustment member (22) which is screwed or press-fitted into an open end of the cylinder (16), and that the spring (21) is interposed between the piston (17) and the adjustment member (22).
4. The expansion device according to claim 3, **characterised in that** the adjustment member (22) is formed with a fixed orifice (24) for communicating the secondary side of the expansion device (3) with the damper chamber (23).
5. The expansion device according to claim 3, **characterised by** a stopper (26) screwed or press-fitted into the adjustment member (22) such that the stopper (26) is adjustable back and forth along an axis of and in relation to the adjustment member (22), for restricting the stroke of the piston (17) in valve-opening direction.
6. The expansion device according to claim 5, **characterised in that** the stopper (26) contains a fixed orifice (24) communicating the damper chamber (23) and the secondary side of the expansion device (3a).
7. The expansion device according to claim 1, **characterised in that** a ratio (b/a) between a valve element outer diameter (b) and a port diameter (a) of the valve hole (13) is not larger than about 1.5, preferably is about 1.16.
8. The expansion device according to claim 2, **characterised in that** a secondary side chamber (20a) accommodating the valve element (15) communicates with the damper chamber (23) via a clearance pro-

vided between the piston (17) and the inner wall of the cylinder (16).

9. The expansion device according to claim 1, **characterised by** the integration of the expansion device (3, 3a) into a refrigeration cycle using carbon dioxide as the refrigerant.

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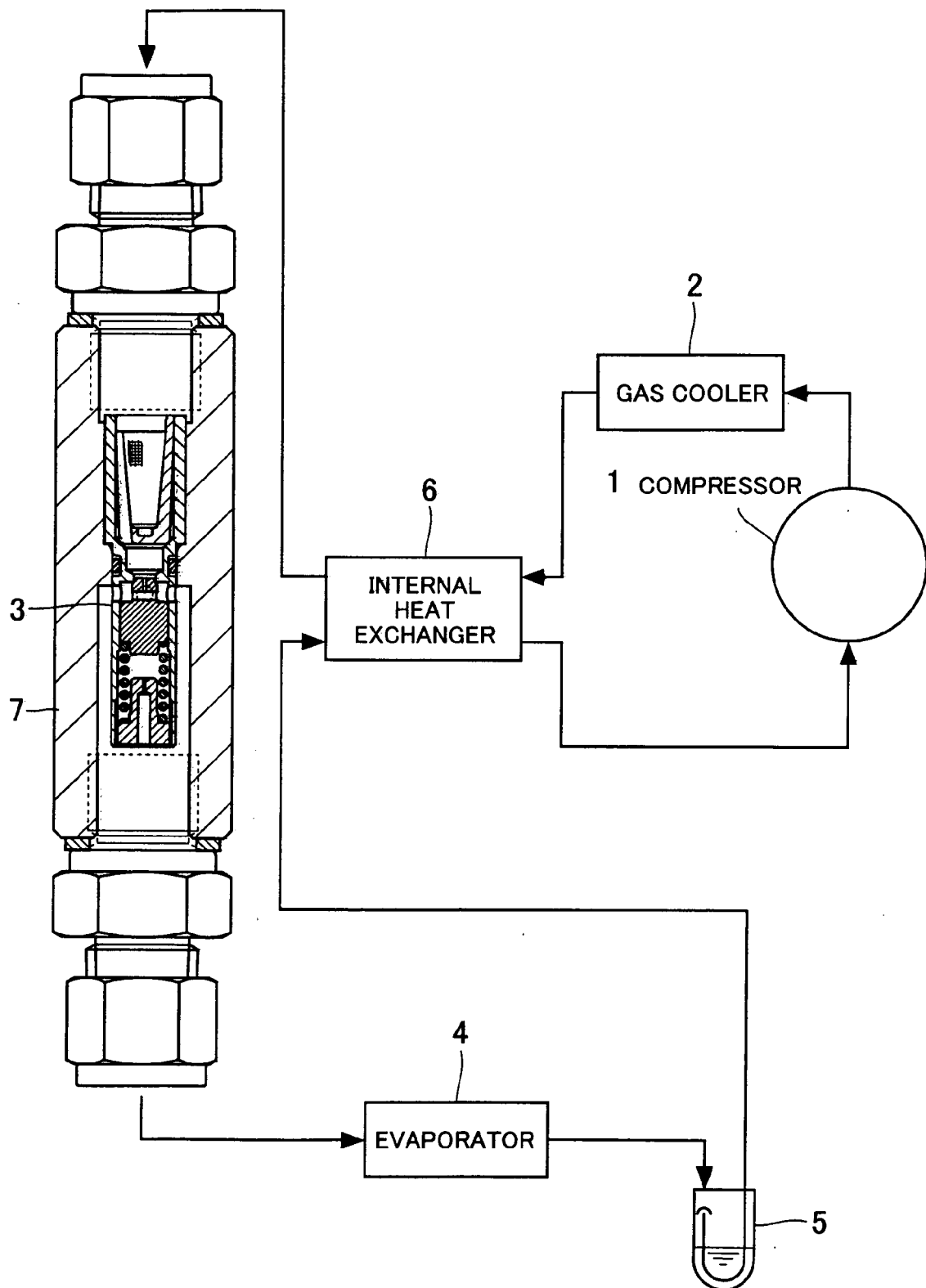


FIG. 1

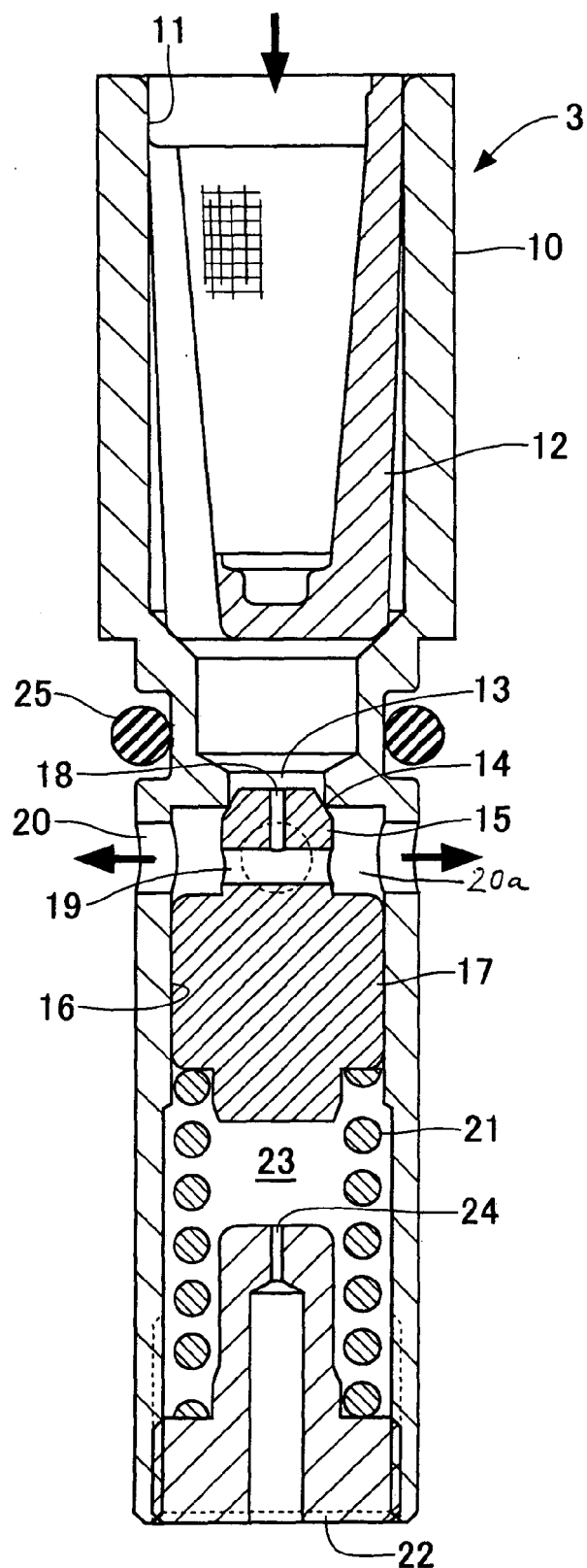


FIG. 2

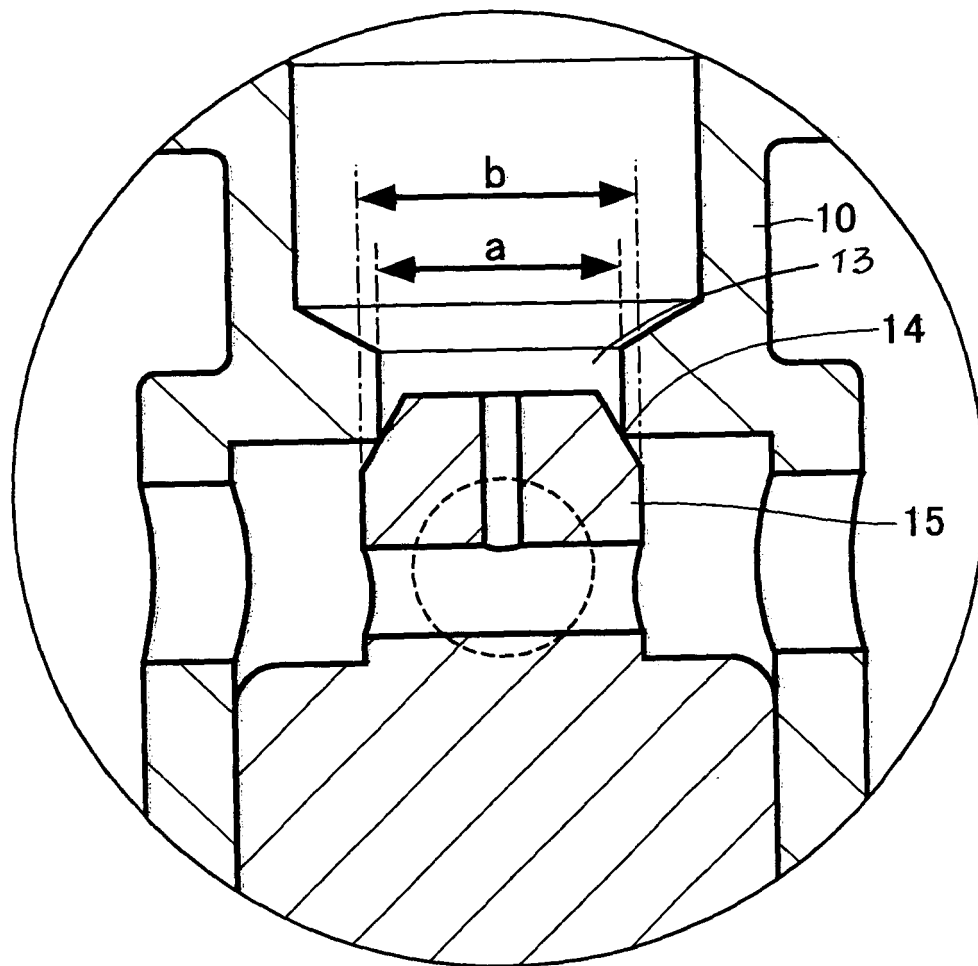


FIG. 3

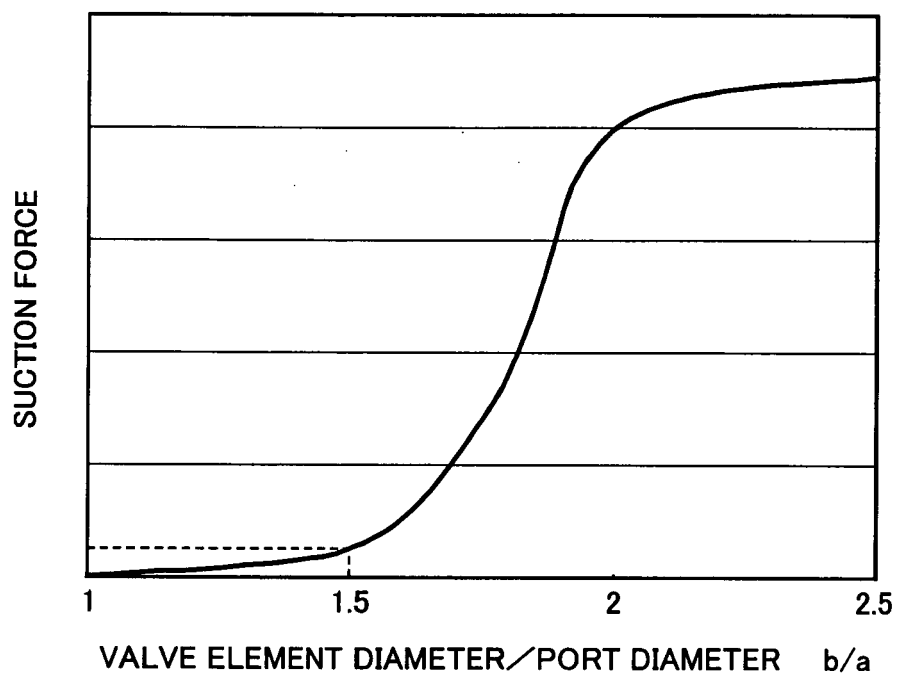


FIG. 4

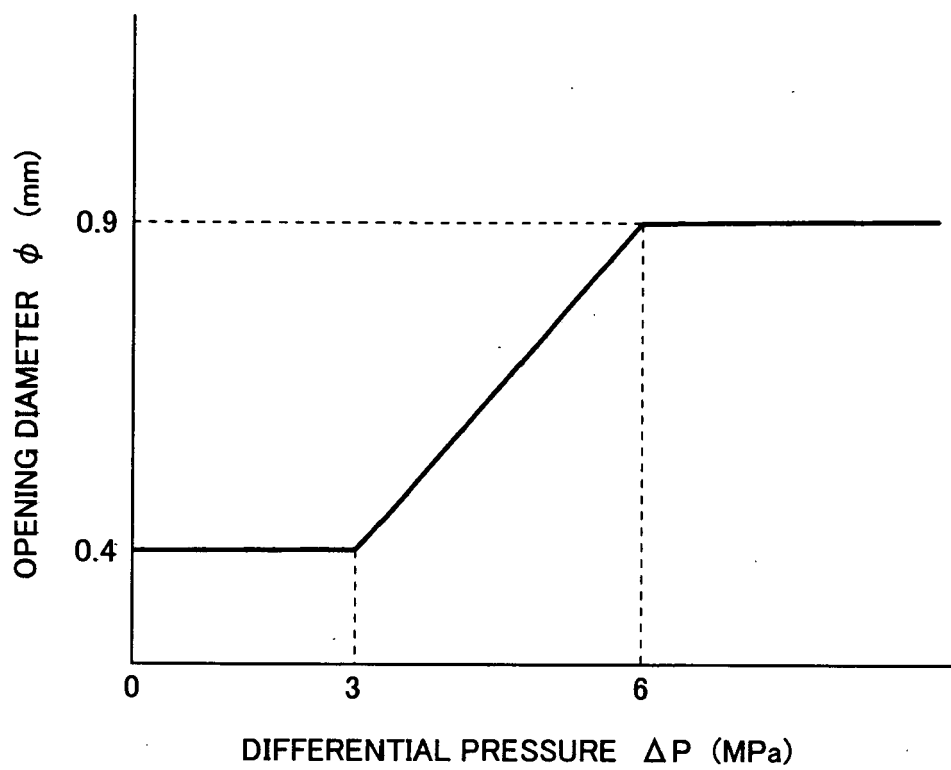


FIG. 6



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Y	* paragraphs [0032], [0036]; figures 8,9 *	7	
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The present search report has been drawn up for all claims			
Place of search The Hague		Date of completion of the search 10 April 2006	Examiner Léandre, A
CATEGORY OF CITED DOCUMENTS X : particularly relevant if taken alone Y : particularly relevant if combined with another document of the same category A : technological background O : non-written disclosure P : intermediate document T : theory or principle underlying the invention E : earlier patent document, but published on, or after the filing date D : document cited in the application L : document cited for other reasons & : member of the same patent family, corresponding document			

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EPO FORM 1503 03/02 (P04C01)

**ANNEX TO THE EUROPEAN SEARCH REPORT
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This annex lists the patent family members relating to the patent documents cited in the above-mentioned European search report. The members are as contained in the European Patent Office EDP file on
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