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(54) **Expansion valve**

(57) The stroke of a valve element 8 of an expansion valve is set such that refrigerant flows at a flow rate 1.0 to 1.4 times the flow rate corresponding to the set tonnage. This means a limit of the valve lift to fully open state during start-up and prevents an unnecessary and excessive refrigerant flow and reduces flow noise of refrigerant during start-up and allows to save driving force for the compressor.

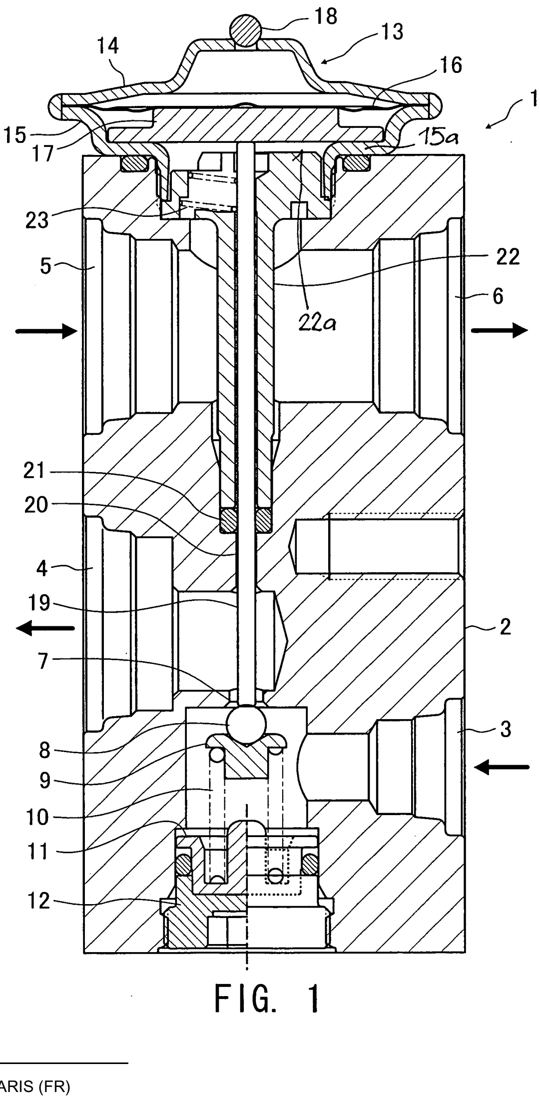


FIG. 1

EP 1 380 801 A2

Description

[0001] This invention relates to an expansion valve according to the preamble of claims 1 and 5, and more particularly to a thermostatic expansion valve in a refrigeration cycle of an automotive air-conditioning system, for allowing a high-temperature and high-pressure liquid refrigerant to expand into a low-temperature and low-pressure refrigerant to supply the refrigerant to an evaporator, and at the same time controlling a flow rate of the refrigerant such that the refrigerant at an outlet of the evaporator is placed in a predetermined degree of superheat.

[0002] In known automotive air-conditioning systems, a high-temperature and high-pressure gaseous refrigerant is condensed in a condenser. The expansion valve allows the condensed liquid refrigerant to undergo adiabatic expansion to be changed into a low-temperature and low-pressure refrigerant, which is evaporated in an evaporator, and then returned to the compressor. The evaporator exchanges heat with air from the vehicle compartment, thereby performing cooling.

[0003] A thermostatic expansion valve known from JP-A-2002-310539 senses the pressure and temperature of refrigerant at an outlet of an evaporator, and controls the flow rate of refrigerant supplied to the evaporator such that the refrigerant is in a predetermined degree of superheat.

[0004] Each respective automotive air-conditioning system has a different required refrigerating capacity depending on the vehicle to which it is applied, and the capacity demanded of the expansion valve is also different. The capacity of the expansion valve is expressed in tonnage. From the tonnage set depending on the vehicle, the flow rate of refrigerant flowing through the expansion valve is determined, and therefore, the expansion valve is designed such that at least the flow rate corresponding to the set tonnage is guaranteed. In this case, the maximum valve lift is unconditionally set to a sufficiently larger value than that corresponding to the set tonnage, for whatever tonnage the expansion valve may be designed.

[0005] However, the conventional expansion valve is fully open when the air conditioner is started, and the valve lift at that time is larger than that providing a required flow rate, causing a large amount of refrigerant to flow. This increases flow noise generated when the refrigerant passes through the valve, and what is worse, since the expansion valve is excessively opened, this causes the refrigerant to flow at a superfluous flow rate, also resulting in an increase in the driving force for the compressor.

[0006] It is an object of the invention to provide an expansion valve that suppresses noise generated during start-up and that saves driving force for the compressor.

[0007] According to the invention, the valve lift in its full-open state is restricted to only 1.0 to 1.4 times the flow rate corresponding to the set tonnage, and therefore, it is possible to prevent the refrigerant from flowing at an unnecessary and excessive flow rate during start-up. This makes it possible to suppress flow noise of the refrigerant during start-up, and at the same time contributes to saving of the driving force for the compressor due to the reduced excessive flow rate of the refrigerant.

[0008] The maximum valve lift may be defined by seating the center disk of the power element on the inner wall of the housing toward the valve portion. This causes the stroke of the center disk toward the fully open state side to be restricted by the housing, whereby the position of the abutment surface of the center disk at which the center disk is brought into abutment with the housing is determined only by the thickness of the housing with reference to the mounting reference surface of the power element, which allows to significantly reduce the number of factors which may cause a dispersion of tolerances of the valve stroke.

[0009] The center disk may be guided in the direction of driving displacement of the diaphragm by a holder holding an end of the shaft on a side opposite to the valve portion. The holder positions the center disk on the same axis as that of the shaft, which prevents the periphery of the center disk from being brought into contact with the inner side wall of the housing when the diaphragm is displaced to move the center disk forward and backward in the direction of displacement thereof, thereby providing stable flow rate characteristics.

[0010] The valve seat may be tapered such that an amount of tapering or the length of the tapered portion of the valve seat is equal to or larger than an amount of axial motion or the length of the valve stroke of the valve element. This makes it possible to prevent the valve element from moving out of the tapered portion of the valve seat when the valve is fully opened or when the helical spring urging the valve element should become inclined.

[0011] By designing the expansion valve such that the maximum opening lift stroke of the valve element is limited exclusively by the mechanical co-action between the lower abutment surface of the center disk and the inner wall of the housing part of the lower housing of the power element detrimental influences of the holder having a thermal coefficient of expansion different from the thermal coefficient of expansion of the expansion valve body and the power element housing as well as of assembly tolerances in the positioning of the holder are avoided for the amount of the opening lift stroke, as the holder does not gain influence on the positioning of the axially maximally displaced center disk. Furthermore, the axial distance over which the center disk is allowed to travel until it is seated on the inner wall of the housing part of the lower housing of the power element is selected such that the actual flow rate of refrigerant passing through the opened valve portion will be limited to 1.0 to 1.4 times the flow rate corresponding to the respectively set tonnage. This means that the valve element cannot be lifted excessively from the value seat which could cause an excessively high flow rate generating extreme noise during the start of operation of the expansion valve and which would require an excessive driving power for the compressor.

[0012] Embodiments of the present invention and prior art examples will be described with reference to the drawings. In the drawings is:

Fig. 1	a longitudinal cross-section of an expansion valve according to the invention,
Fig. 2	a diagram of the relationship between the valve stroke and a refrigeration ton or tonnage, for the invention and the prior art,
Fig. 3	a diagram of the relationship between a factor by which the refrigerating capacity is increased and noise generated during start-up,
Fig. 4	a diagram of changes in noise immediately after the expansion valve has started to operate, for the invention and the prior art,
Fig. 5 (A)	a diagram explaining tolerance dispersion in the case of the conventional expansion valve (prior art),
Fig. 5 (B)	a diagram explaining tolerance dispersion in the case of the expansion valve according to the present invention,
Fig. 6	a longitudinal cross-section of another expansion valve according to the invention, and
Fig. 7	a longitudinal cross-section of the conventional expansion valve according (prior art).

[0013] The expansion valve 1 according to the invention has in Fig. 1 a body block 2 having side portions formed with a refrigerant conduit connection hole 3 to which is connected a high-pressure refrigerant piping for receiving a high-temperature and high-pressure refrigerant from a receiver/dryer through the piping, a refrigerant conduit connection hole 4 to which is connected a low-pressure refrigerant piping for supplying a low-temperature and low-pressure refrigerant expanded and reduced in pressure by the expansion valve 1 to the evaporator, a refrigerant conduit connection hole 5 connected to refrigerant piping from an evaporator outlet, and a refrigerant conduit connection hole 6 connected to a refrigerant piping leading to the compressor.

[0014] Further, in a fluid passage between the refrigerant conduit connection hole 3 and the refrigerant conduit connection hole 4, a valve seat 7 is integrally formed with the body block 2, and a ball-shaped valve element 8 is disposed in a manner opposed to the valve seat 7 from the upstream side. A gap between the valve seat 7 and the valve element 8 forms a variable orifice for reducing the flow of the high-pressure refrigerant, and the high-pressure liquid refrigerant is adiabatically expanded when it flows through the variable orifice. The valve seat 7 is tapered such that the axial amount of the tapered portion is equal to or more than the maximum amount of axial motion (stroke length) of the valve element 8. More specifically, a portion of the valve hole opposed to the valve element 8 has its edge cut to form a tapered or conical hole portion, and the axial length (height) of the tapered hole portion is equal to or larger than the length of the stroke of the valve element 8. Here, a smallest-diameter portion of the tapered hole portion or the transition between the conical portion and the cylindrical valve hole sets the seating position of the valve element 8, and even if the valve element 8 is moved to a farthest position therefrom to make the valve fully open, part of the valve element 8 is positioned within the tapered hole portion, which prevents the valve element 8 from moving out of the tapered hole when the valve is fully opened. The cone angle of the conical portion is adapted to the ball diameter of the valve element 8 such that the valve element 8 will be seated on the transition between the conical portion and the cylindrical valve hole.

[0015] Further, in the fluid passage on the side of the refrigerant conduit connection hole 3, there are disposed a valve element receiver 9 for receiving the valve element 8, and a helical compression spring 10 for urging the valve element 8 via the valve element receiver 9 in the direction of seating the valve element 8 on the valve seat 7. The helical compression spring 10 is received by a spring receiver 11 and an adjustment screw 12 screwed into the body block for adjustment of load of the helical compression spring 10.

[0016] At an upper end of the body block 2, there is provided a power element 13 which comprises an upper housing 14 and a lower housing 15, made of a thick metal, a diaphragm 16 made of a thin metal plate having flexibility and disposed in a manner dividing the space surrounded with the housings, and a center disk 17 disposed below the diaphragm 16. The space surrounded with the upper housing 14 and the diaphragm 16 forms a temperature-sensing chamber which is filled with two or more kinds of refrigerant gas and inert gas, and is sealed with a metal ball 18 by resistance-welding. The center disk 17 has a lower part formed with an increased diameter such that the part radially protrudes outward, and the underside thereof is formed to have a flat surface. The inner wall surface of a housing part 15a of the lower housing 15 opposed to the underside of a protruding portion of the center disk 17 is formed to have a flat surface portion. The flat portion of the inner wall surface functions as a stopper limiting the downward motion of the center disk 17, thereby defining the maximum valve lift of the valve element 8 of the expansion valve 1.

[0017] Below the center disk 17, a shaft 19 is disposed for transmitting displacement of the diaphragm 16 to the valve element 8. The shaft 19 is inserted through a through hole 20 formed in the body block 2.

[0018] The through hole 20 has an upper part thereof expanded, and an O ring 21 is disposed at a stepped portion thereof. The O ring 21 seals a gap between the shaft 19 and the through hole 20, thereby preventing refrigerant from leaking into the fluid passage between the refrigerant conduit connection holes 5 and 6.

[0019] Further, the upper end of the shaft 19 is held by a holder 22 which has a hollow cylindrical portion extending downward across the fluid passage communicating between the refrigerant conduit connection holes 5, 6. The lower end of the holder 22 is fitted in the expanded portion of the through hole 20 and the lower end surface restricts the motion of the O ring 21 toward the upper open end of the through hole 20.

[0020] A coil spring 23 is disposed with the upper end portion of the holder 22, for urging the shaft 19 from a radial direction. This configuration of applying lateral load to the shaft 19 with the coil spring 23 prevents the axial motion of the shaft 19 from sensitively reacting to changes in pressure of the high-pressure refrigerant in the refrigerant conduit connection hole 3. That is, the coil spring 23 forms a vibration suppressing mechanism for suppressing generation of untoward vibration noise caused by vibrations of the shaft 19 in the axial direction.

[0021] Further, the top portion of the holder 22 has a passage formed therethrough for communicating the fluid passage communicating between the refrigerant conduit connection holes 5, 6 and the space below the diaphragm 16, and at the same time, the underside of the center disk 17 is formed with a plurality of not shown ventilation grooves in a radially extending manner, except a central portion with which the shaft 19 is in abutment, thereby allowing the refrigerant returned from the evaporator to enter the chamber below the diaphragm 16. A top surface 22a of the holder is situated at a lower level than the lower abutment surface of the center disk 17 when the center disk 17 is in stroke limiting contact with the inner wall of housing part 15a.

[0022] Before the air conditioner is started, the power element 13 detects a sufficiently higher temperature than that during operation of the air conditioner. The pressure in the temperature-sensing chamber of the power element 13 is increased, which causes the diaphragm 16 to be displaced downward as shown in Fig. 1, whereby the center disk 17 abuts against the stopper of the lower housing 15. This displacement of the diaphragm 16 is transmitted to the valve element 8 via the shaft 19, thereby making the expansion valve 1 fully open. Therefore, when the air conditioner is started, the expansion valve 1 starts its operation from the fully open state, and therefore, the expansion valve 1 supplies refrigerant to the evaporator at the maximum flow rate.

[0023] As the temperature of the refrigerant returned from the evaporator is lowered, the temperature in the temperature-sensing chamber of the power element 13 is lowered, whereby the refrigerant gas in the temperature-sensing chamber is condensed on the inner surface of the diaphragm 16. This causes pressure in the temperature-sensing chamber to be reduced to displace the diaphragm 16 upward, so that the shaft 19 is pushed by the helical compression spring 10, to move upward. As a result, the valve element 8 is moved toward the valve seat 7, whereby the passage area for the high-pressure liquid refrigerant is reduced to decrease the flow rate of refrigerant sent into the evaporator. Thus, the valve lift of the expansion valve is set to a value dependent on the cooling load.

[0024] Fig. 2 is a diagram showing the relationship between the stroke of a valve and refrigeration ton.

[0025] The expansion valve 1 has its capacity determined according to the refrigerating capacity demanded by the system, and in general, there are a 1.0-ton type, a 1.5-ton type, and a 2.0-ton type of expansion valves. In all of these types, the valve element 8 has its valve lift controlled within a range of stroke corresponding to the associated refrigeration ton. For conventional expansion valves, the maximum valve lift during start-up is set as also shown in Fig. 2, irrespective of the type, by a certain sufficiently large value of stroke A, e.g. 0.8 mm. However, in the expansion valve 1 according to the present invention, the maximum valve lift is set by such a stroke as will cause the refrigerant to flow at 1.0 to 1.4 times the flow rate corresponding to the designated tonnage, and not more. For example, in the case of the 1.0-ton type expansion valve, the maximum stroke is set to a value within a range between a stroke position B allowing the refrigerant to flow at a flow rate satisfying the capacity of 1 ton and a position B' of the maximum valve lift allowing the refrigerant to flow at 1.4 times the above flow rate.

[0026] Fig. 3 shows how the noise generated during start-up of the expansion valve 1 varies with a change in the scale factor of the refrigerating capacity. According to Fig. 3, noise is steeply increased when the refrigerating capacity is in the vicinity of 1.4 times or exceeds the same. The expansion valve is configured such that the refrigerating capacity can only be increased by a factor of 1.4 at the maximum, in the fully-open state of the expansion valve, which makes it possible to suppress generation of excessive noise during start-up.

[0027] Further, due to the refrigerating capacity limited up to a scale factor of 1.4, the noise immediately after start-up is made much smaller than in the prior art, as shown in Fig. 4. With the lapse of time, the refrigeration cycle becomes stable, causing the expansion valve 1 to enter the control region, so that the noise becomes equal in magnitude to the prior art.

[0028] The expansion valve according to the present invention is required to make the maximum stroke of the shaft 19 in Figs 1 and 6 and 5B smaller than that of the conventional expansion valve of Figs 5A and 7. For example, in the case of 1.0-ton type expansion valve, the maximum stroke of the shaft is reduced from the conventional value of 0.8 mm to a value of 0.3 mm. Therefore, tolerance dispersion in the sizes of members determining the stroke has a large influence on the valve, and therefore the dispersion is required to be made small. The expansion valve according to the present invention solves this problem by changing the stopper of the center disk 17 from the holder 22 (as conventional) to the lower housing 15 of the power element 13.

[0029] A conventional expansion valve 101 includes in Figs 7, 5A a body block 102 having side portions formed with a refrigerant conduit connection hole 103 for introducing refrigerant, a refrigerant conduit connection hole 104 for delivering refrigerant, and refrigerant conduit connection holes 105, 106 for being intervened in piping leading from an evaporator to a compressor.

[0030] In a fluid passage between the refrigerant conduit connection hole 103 and the refrigerant conduit connection hole 104, a valve seat 107 is integrally formed with the body block 102, and a ball-shaped valve element 108 is disposed in a manner opposed to the valve seat 107 from the upstream side, and refrigerant undergoes adiabatic expansion when it flows through a gap between the valve seat 107 and the valve element 108. Further, the valve element 108 is urged by a helical compression spring 110 via a valve element receiver 109 for receiving the valve element 108 in a direction of being seated on the valve seat 107. The helical compression spring 110 is received by a spring receiver 111 and an adjustment screw 112.

[0031] A power element 113 is provided at an upper end of the body block 102. The power element 113 comprises an upper housing 114, a lower housing 115, a diaphragm 116, and a center disk 117. A temperature-sensing chamber surrounded with the upper housing 114 and the diaphragm 116 is filled with refrigerant, and sealed by a metal ball 118.

[0032] The upper end of a shaft 119 is in abutment with the center disk 117. The shaft 119 is inserted through a through hole 120 formed in the body block 102, and has a lower end thereof in abutment with the valve element 108.

[0033] The through hole 120 has an upper part thereof expanded, and an O ring 121 is disposed at a stepped portion thereof, for sealing a gap between the shaft 119 and the through hole 120.

[0034] Further, the upper end of the shaft 119 is held by a holder 122 which has a hollow cylindrical portion extending downward across a fluid passage communicating between the refrigerant conduit connection holes 105, 106. The lower end of the holder 122 is fitted in the expanded portion of the through hole 120 and retains the O ring 121.

[0035] A coil spring 123 is disposed at the upper end of the holder 122, for suppressing axial vibrations of the shaft 119. The top surface of the holder 122 functions as a stopper defining the maximum valve lift of the expansion valve 101.

[0036] In the expansion valve 101 constructed as described above, before the air conditioner is started, the center disk 117 is in abutment with the top surface of the holder 122, and the expansion valve 101 is fully open. Therefore, when the air conditioner is started, the expansion valve 101 starts its operation from the fully open state thereof.

[0037] More specifically, in the conventional expansion valve, as shown in Fig. 5(A), the stroke S of the shaft 119 is from a position at which the center disk 117 is in contact with the top surface of the holder 122 to the illustrated position assumed when the valve is fully closed. Further, P designates an amount of protrusion of the shaft 119 from the top surface of the body block 102 when the valve is fully closed. Further, A designates a height from the stepped portion of the body block 102, where the holder 122 is received, to the top surface of the body block 102, B a height from the bottom surface of the holder 122 lying on the stepped portion to a surface of the holder 122 with which the center disk 117 is brought into contact when the valve is fully opened, and C a height from the surface of the center disk 117 with which the shaft 119 is in abutment to a surface of the center disk 117 which is brought into contact with the holder 122 when the valve is fully opened.

[0038] By using the stepped portion receiving the holder 122 of the body block 102 as the reference, an expression of $(A + P) + C = B + S$ holds, and from this, the stroke S can be expressed as $S = A + P + C - B$. More specifically, the number of parameters determining the stroke S is four.

[0039] On the other hand, in the expansion valve 1 according to the present invention, as shown in Fig. 5 (B), the stroke S of the shaft 19 is from a position in which the center disk 17 is in abutment with the inner surface of the lower housing 15 to the illustrated position assumed when the valve is fully closed. Now, if the top surface of the body block 2 on which the power element 13 is mounted is used as the reference surface, and the thickness of the lower housing 15 is represented by t, the amount P of protrusion of the shaft 19 from the reference surface is expressed as $P = t + S$, so that the stroke S can be expressed by $S = P - t$. Therefore, the number of parameters determining the stroke S becomes two, which means the number of dispersion-causing factors is reduced to half. This makes it possible to make the tolerance dispersion smaller than the conventional expansion valve.

[0040] Particularly, when the holders 22, 122 are made of resin, since the resin is thermally expanded, the conventional expansion valve suffers from dispersion of the parameter B caused by the refrigerant temperature, which makes the value of the stroke S a function of temperature. In contrast, in the present expansion valve, the parameters determining the stroke S do not contain the parameter B, which makes it possible to further decrease the tolerance dispersion.

[0041] An expansion valve 1a according to Fig. 6 is different from the expansion valve 1 shown in Fig. 1 in which the center disk 17 is guided by the inner wall surface of a vertical portion of the lower housing 15, in that the same is guided by the holder 22 of the shaft 19.

[0042] More specifically, the center disk 17 in Fig. 6 has a lower central portion protruding downward, and this protruding portion is inserted in a hole formed in the top of a holder 22, whereby it is guided by the holder 22 in a manner movable forward and backward along the axis of the shaft 19. This causes the center disk 17 to be positioned by the holder 22 on the same axis as that of the shaft 19, which enables the center disk 17 to move smoothly without being caught by the lower housing 15 when the center disk 17 is moved forward and backward by displacement of the diaphragm 16, providing stable flow rate characteristics. The holder top surface 22a does not contact the center disk 17 when the center disk 17 is in contact with the inner wall of the housing part 15a of the lower housing 15.

[0043] The center disk 17 is configured such that a surface of the protruding portion for abutment with the shaft 19 and a surface thereof for abutment with the stopper of the lower housing 15 are both formed to be flat, and a plurality of ventilation grooves 17a is formed on the surface for abutment with the stopper in a radially extending manner, thereby allowing the refrigerant returned from the evaporator to enter the chamber below the diaphragm 16 via the ventilation grooves even when the valve is in the state of the maximum valve lift in which the center disk 17 is brought into contact with the lower housing 15.

[0044] Further, the expansion valve 1a according to Fig. 6 is configured such that an adjustment screw 12a also plays the role of the spring receiver to thereby reduce the number of component parts.

Claims

1. An expansion valve (1, 1a) including a power element (13) that senses pressure and temperature of refrigerant at an outlet of an evaporator and controls a valve lift of a valve portion, to thereby control a flow rate of refrigerant supplied to the evaporator,
   characterized in that a maximum value of the valve lift is set such that the flow rate is equal to 1.0 to 1.4 times a flow rate corresponding to a set tonnage.

2. Expansion valve as in claim 1, **characterized in that** the power element (13) causes a center disk (17) for transmitting displacement of a diaphragm (16) sensing the pressure and temperature of the refrigerant to a valve element (8) of the valve portion via a shaft (19) to be brought into abutment in a displacement direction towards the valve portion with an inner wall of a housing (15), thereby defining the maximum valve lift of the valve element (8) of the valve portion.

3. Expansion valve as in claim 2, **characterized in that** the center disk (17) is guided in the direction of displacement of the diaphragm (16), by a holder (22) holding an end of the shaft (19) on a side opposite to the valve portion.

4. Expansion valve as in claim 1, **characterized in that** the valve portion comprises a valve seat (7), a valve element (8) having a shape of a ball and disposed in a manner opposed to the valve seat (7) from an upstream side, and a spring (10) for urging the valve element (8) in a valve-closing direction, and that the valve seat (7) has a tapered portion such that an amount or axial length of the tapered portion is equal to or larger than the maximum amount of axial motion of the valve element (8) out of a seated state on the valve seat (7).

5. An expansion valve (1, 1a), particularly for an automotive air conditioning system, the expansion valve comprising a stationary valve seat (7), an axially moveable valve element (8) associated to the valve seat (7) within a valve body (2), a power element (13) for moving the valve element (8) by displacement of a diaphragm (16), a diaphragm center disk (17) and a driving shaft (19), the power element (13) comprising a housing (14, 15) with a lower housing part (15a) extending below the center disk (17), and a holder (22) mounted in the valve body (2) with a holder top surface (22a) facing the center disk (17), the expansion valve (1,1a) having a tonnage depending capacity setting determining the flow rate through the expansion valve to guarantee at least a flow rate corresponding to the set tonnage, **characterized in that** the center disk (17) has a lower abutment surface axially aligned to a counter-abutment surface of the housing part (15a) of the lower housing (15) of the power element (13) for direct contact between the axially maximally displaced center disk (17) and the inner wall of the housing part (15a),
   that the axial distance between the abutment surface and the counter-abutment surface is set such that in seated condition of the valve element (8) the maximum valve element lift stroke is limited to an amount corresponding to a flow rate between the fully lifted valve element (8) and the valve seat (7) 1.0 to 1.4 times the flow rate corresponding to the set tonnage, and that in the maximally axially displaced center disk position a clearance is formed between the center disk and the holder top surface (22a).

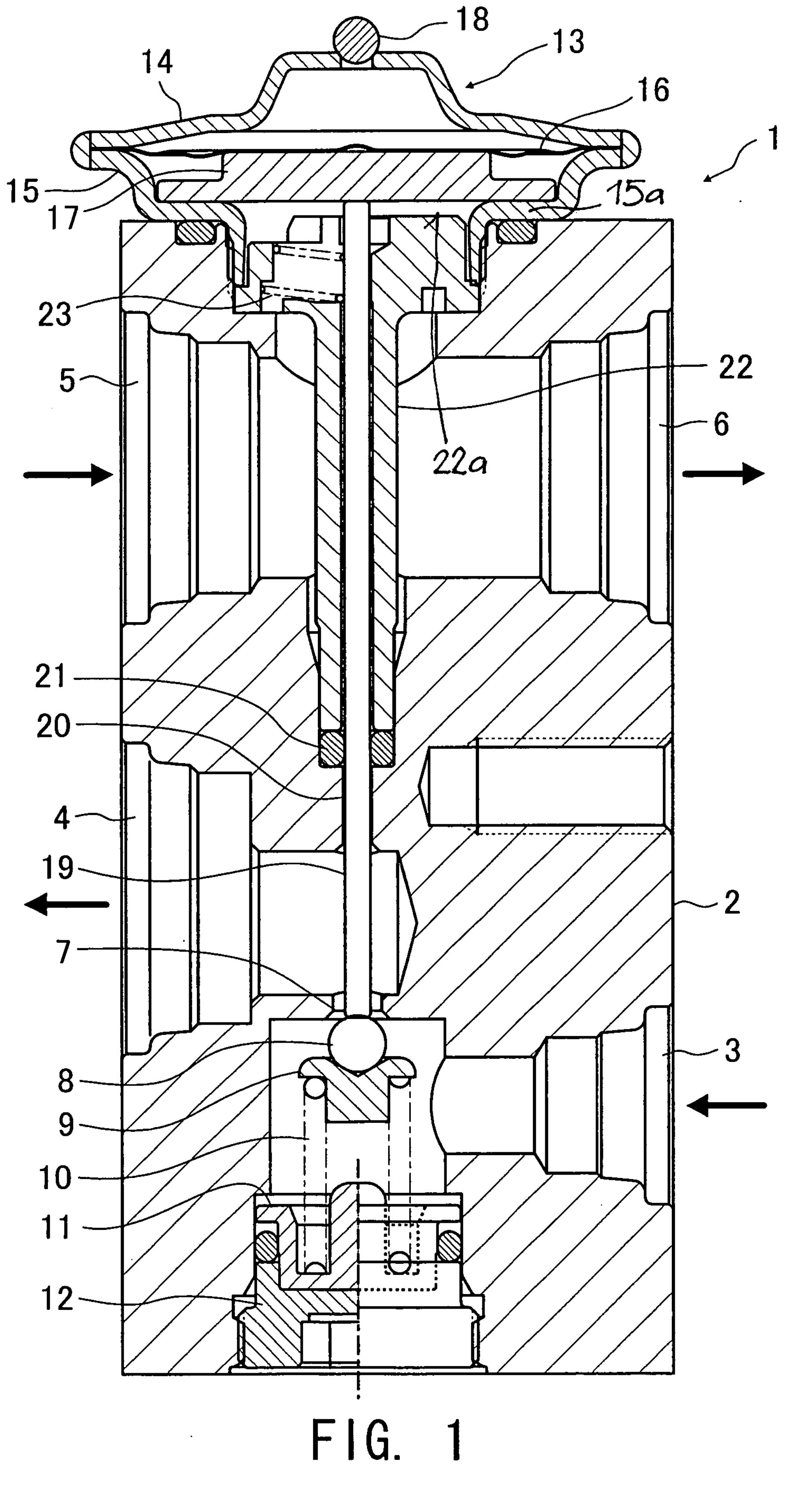
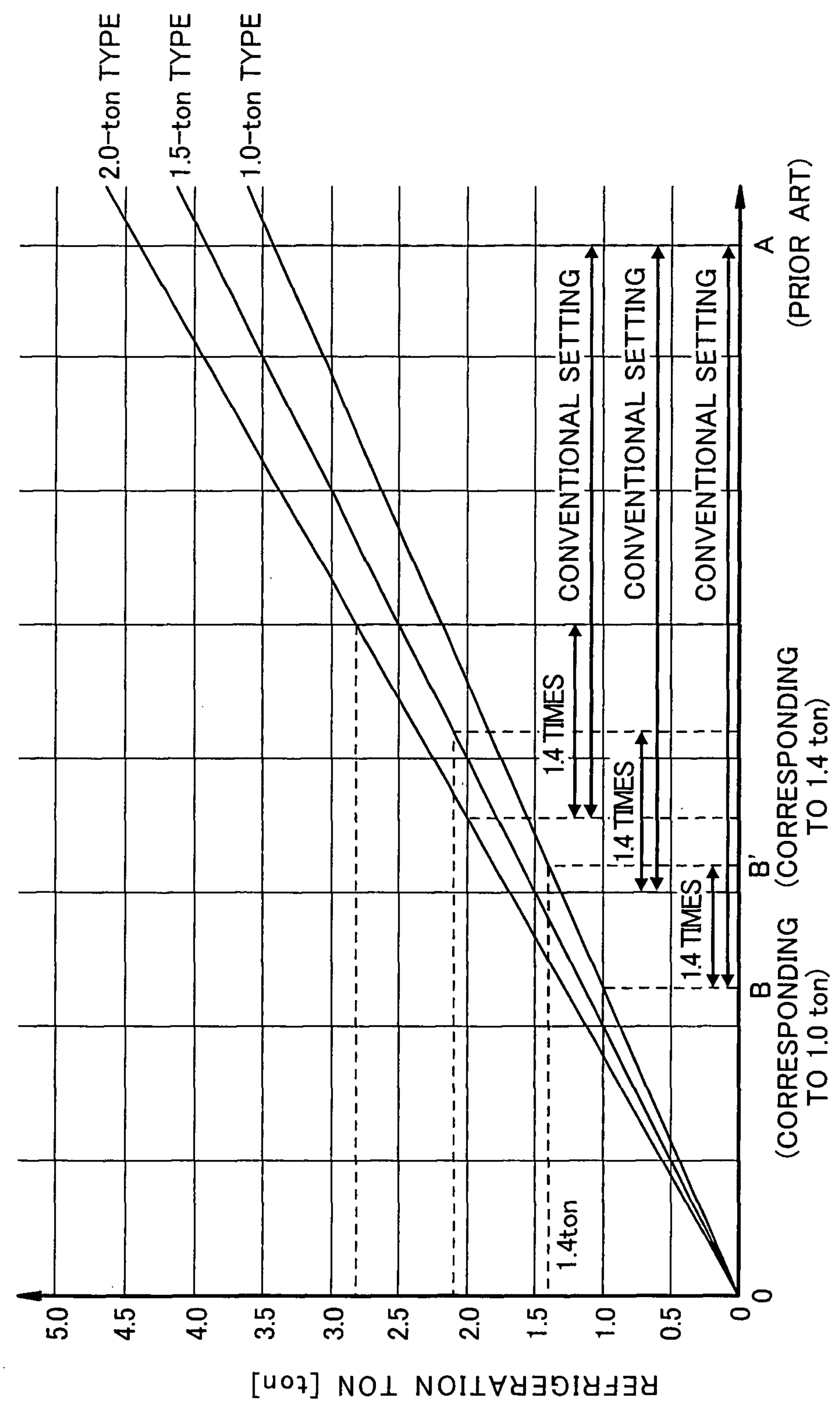


FIG. 1



2.0-ton TYPE
1.5-ton TYPE
1.0-ton TYPE
REFRIGERATION TON [ton]
5.0
4.5
4.0
3.5
3.0
2.5
2.0
1.5
1.0
0.5
0
0
1.4ton
CONVENTIONAL SETTING
CONVENTIONAL SETTING
CONVENTIONAL SETTING
1.4 TIMES
1.4 TIMES
1.4 TIMES
B
(CORRESPONDING
TO 1.0 ton)
B'
(CORRESPONDING
TO 1.4 ton)
A
(PRIOR ART)
STROKE

FIG. 2

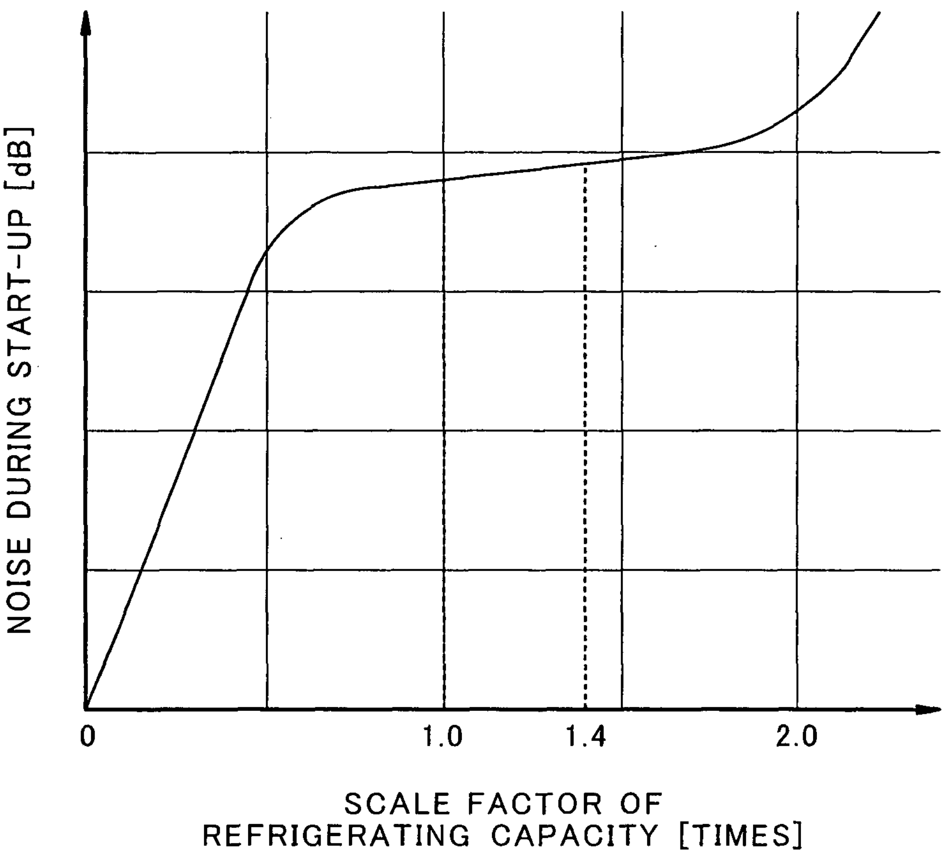

NOISE DURING START-UP [dB]
0
1.0
1.4
2.0
SCALE FACTOR OF
REFRIGERATING CAPACITY [TIMES]

FIG. 3

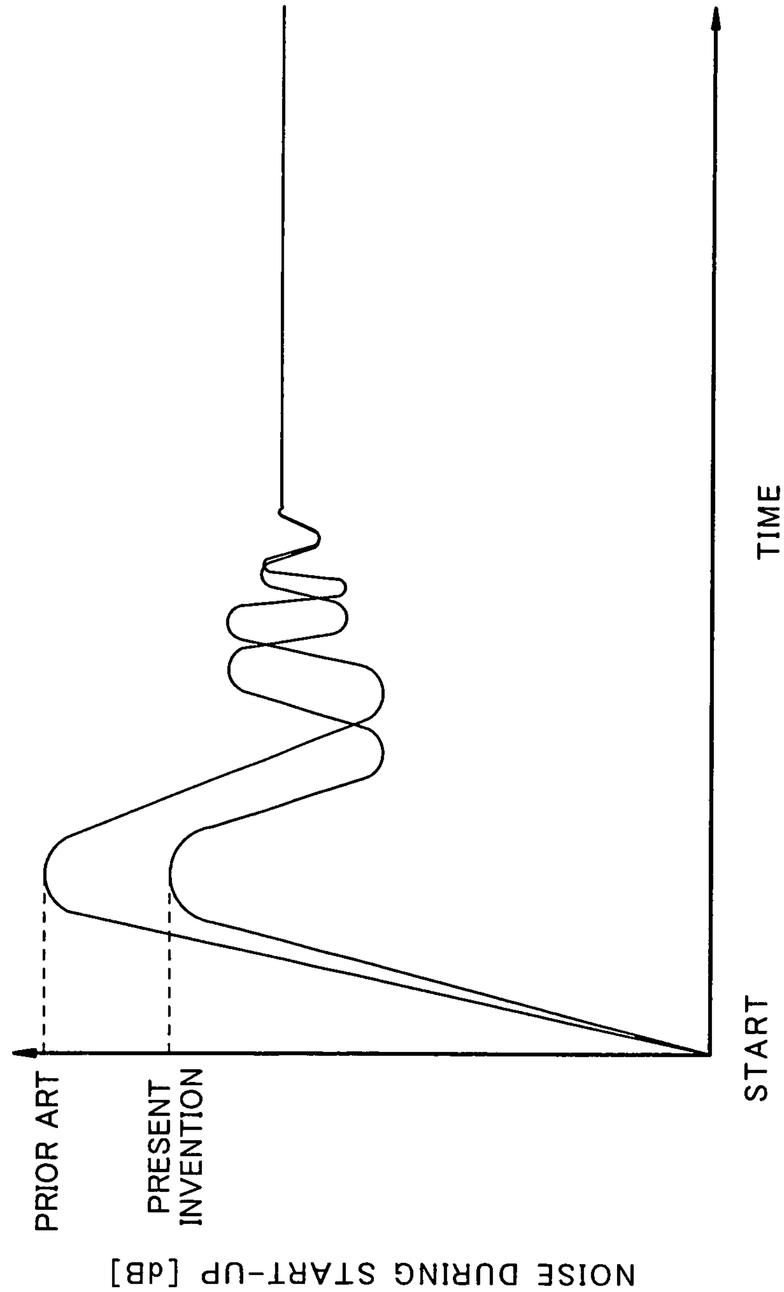

PRIOR ART
PRESENT INVENTION
NOISE DURING START-UP [dB]
START
TIME

FIG. 4

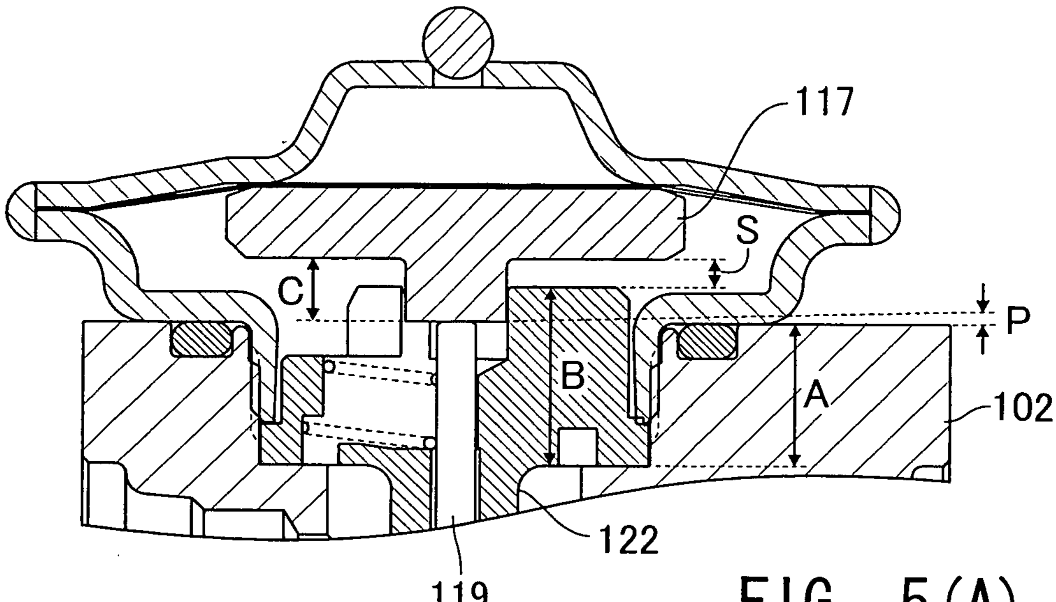


FIG. 5(A)
PRIOR ART

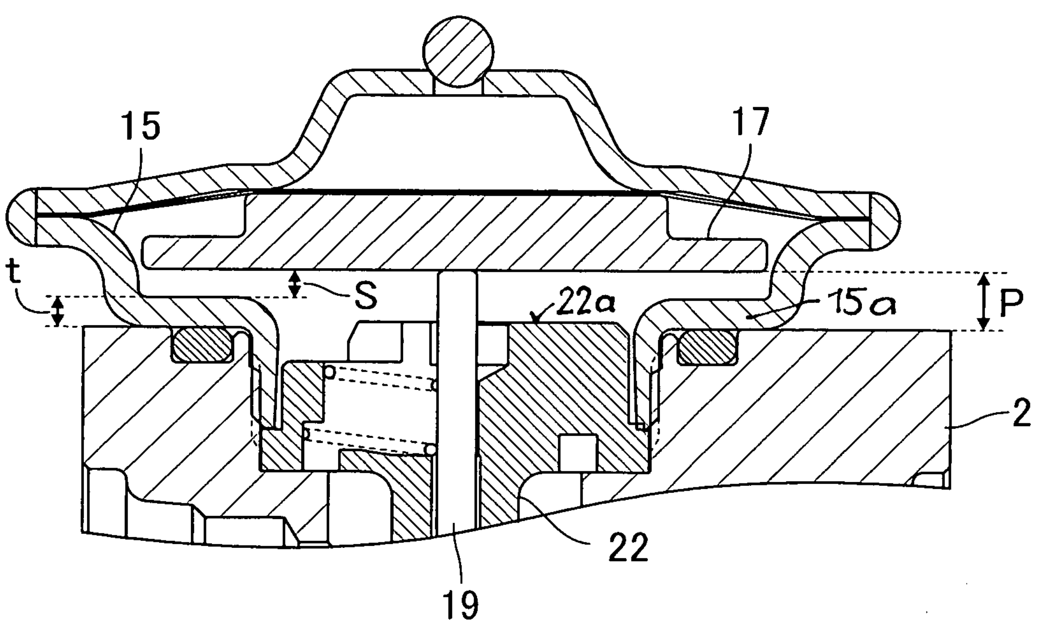


FIG. 5(B)

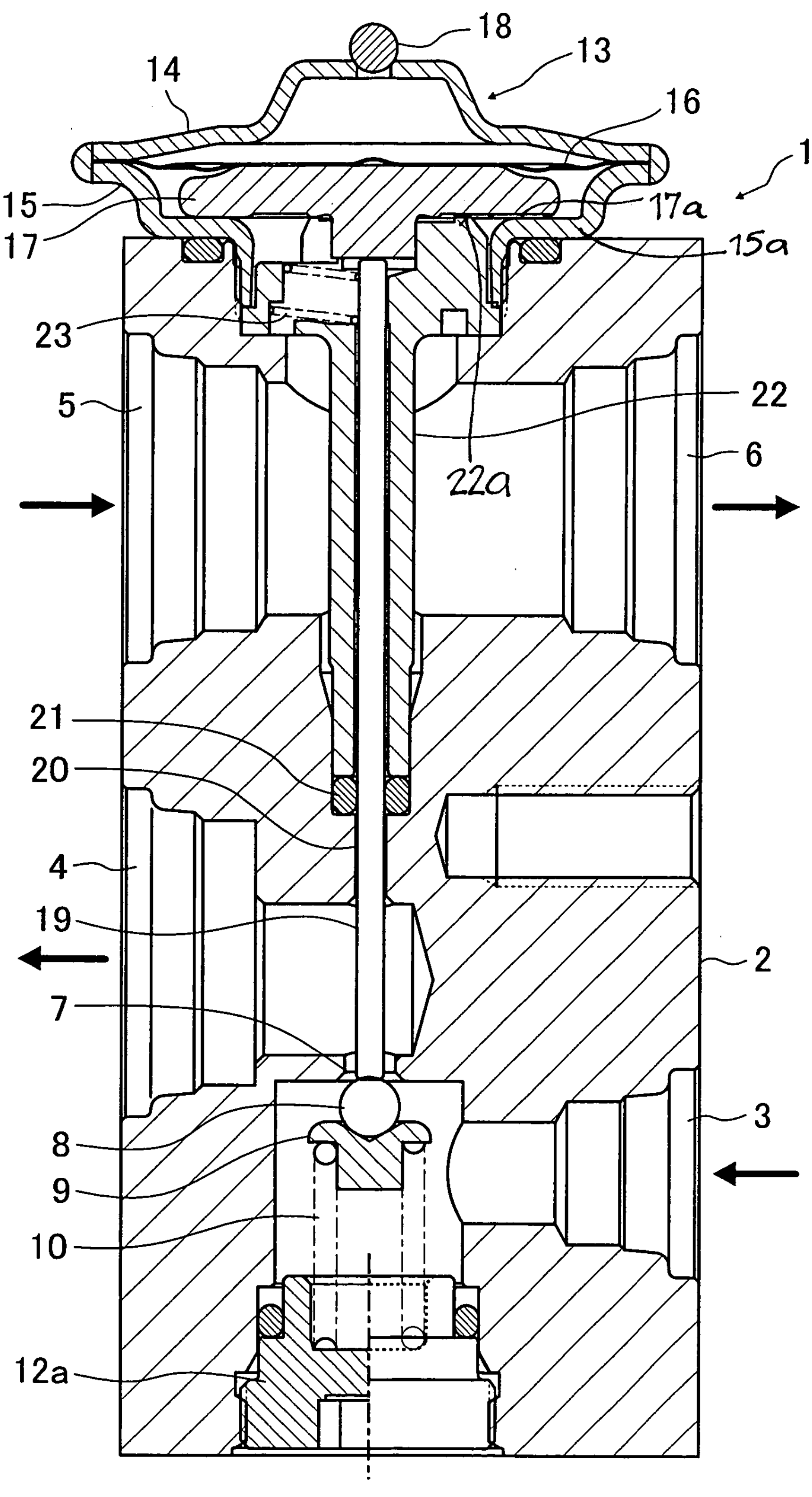


FIG. 6

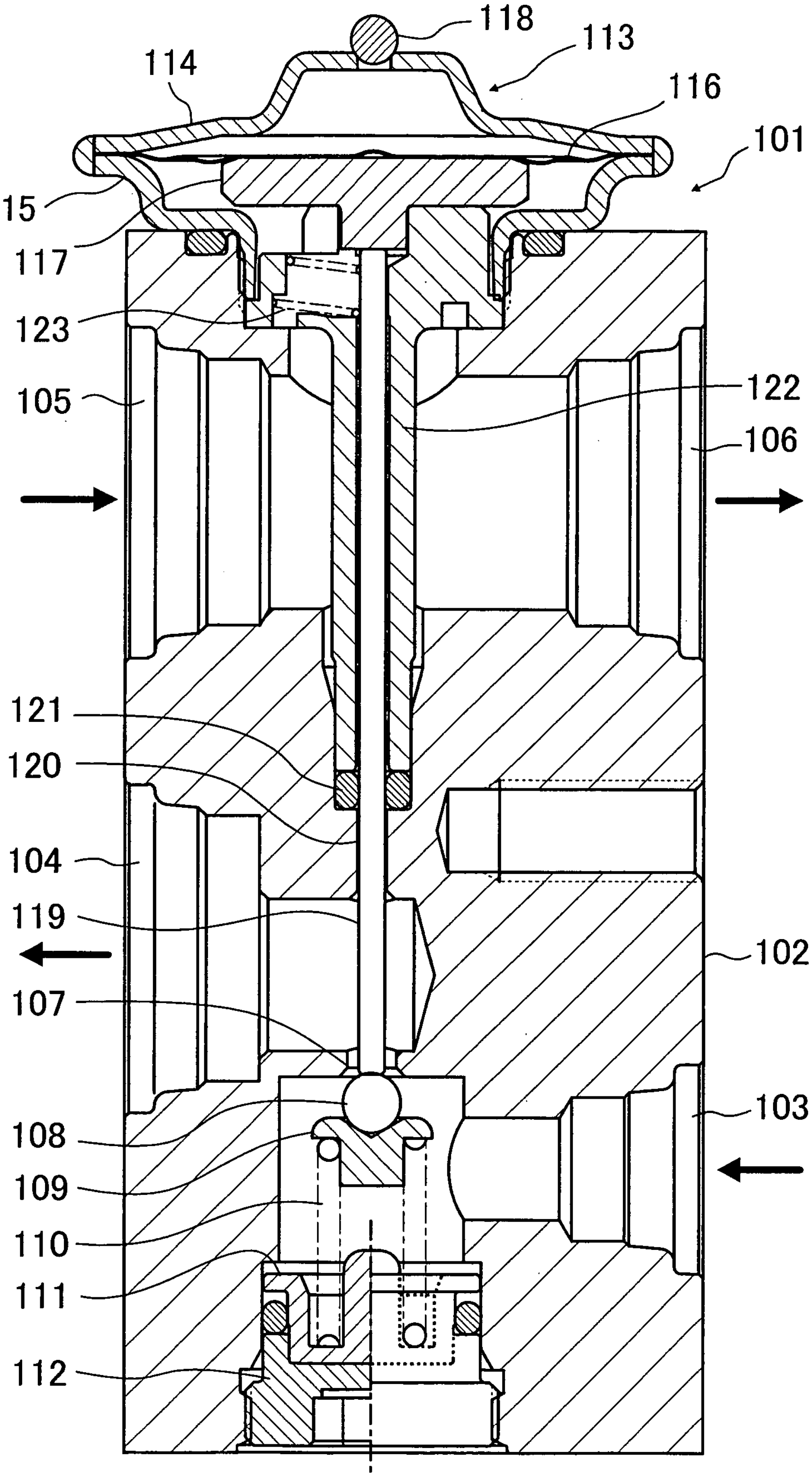


FIG. 7 (PRIOR ART)