

Feb. 6, 1968

A. OWENS  
EXPANSION VALVE AND REFRIGERATION SYSTEM  
RESPONSIVE TO SUBCOOLING TEMPERATURE

3,367,130

Filed Feb. 23, 1966

2 Sheets-Sheet 1

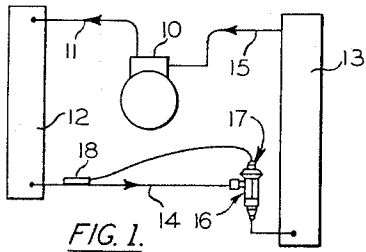


FIG. 1.

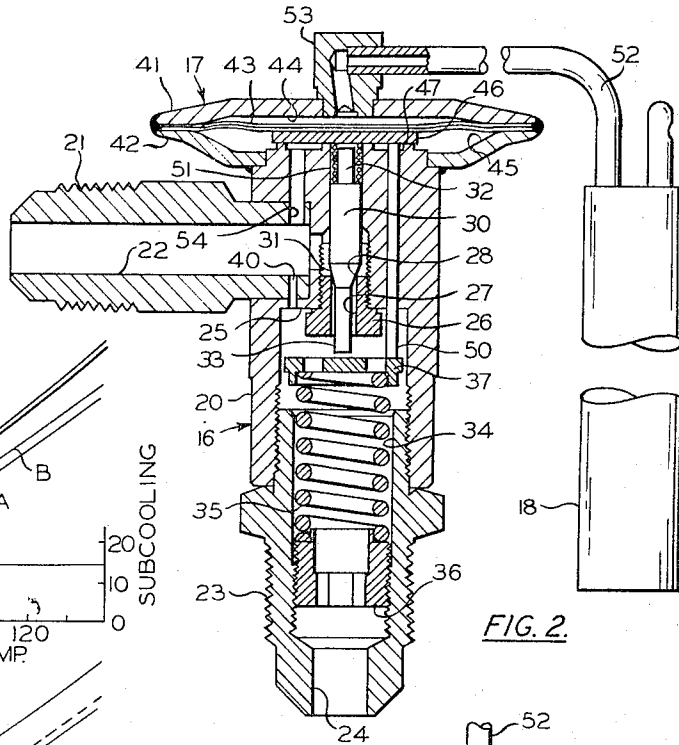


FIG. 2.

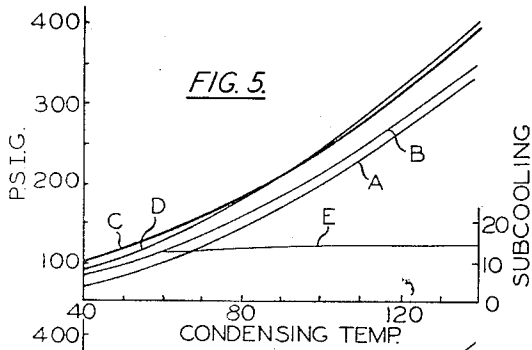


FIG. 5.

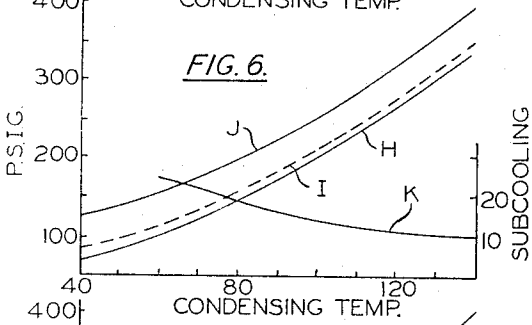


FIG. 6.

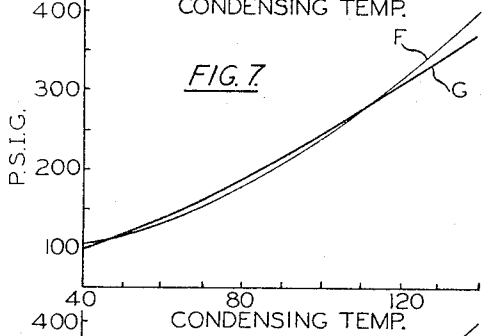


FIG. 7.

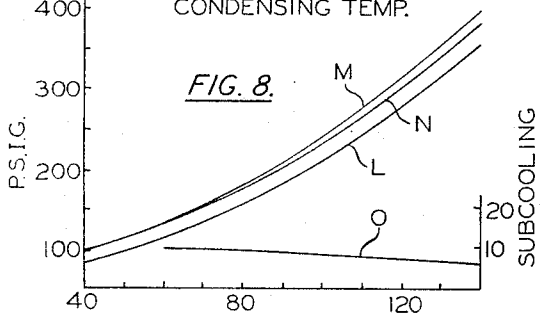


FIG. 8.

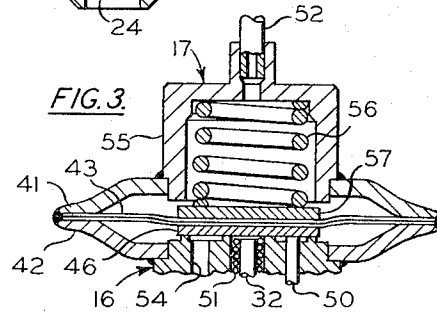


FIG. 3.

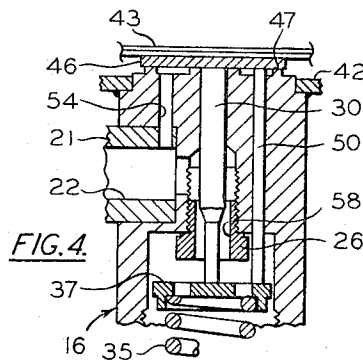


FIG. 4.

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2 Sheets-Sheet 2

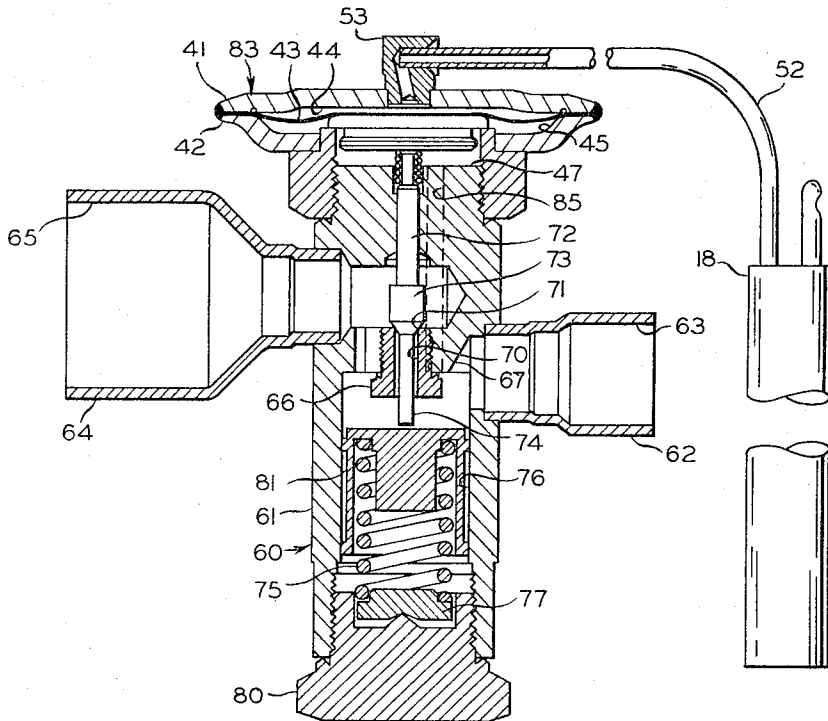


FIG. 9

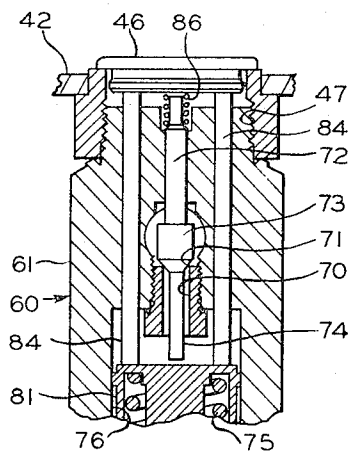


FIG. 10

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**EXPANSION VALVE AND REFRIGERATION SYSTEM RESPONSIVE TO SUBCOOLING TEMPERATURE**

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 Filed Feb. 23, 1966, Ser. No. 529,330  
 20 Claims. (Cl. 62—222)

This invention relates generally to improvements in an expansion valve and in a refrigeration system in which the expansion valve is utilized, and more particularly to an improved control valve of this type that is responsive to the number of degrees the liquid refrigerant is subcooled in the system at some point between the condenser and the control valve.

An important objective is achieved by the provision of an expansion valve in a refrigeration system in which a thermal-sensing means subjects one side of a flexible motor element to a pressure that is a function of the temperature of the refrigerant at a point in the line between the condenser and the expansion valve and which tends to urge the valve means toward a closed position, and by the provision of means subjecting the other side of the motor element to the liquid pressure of the refrigerant at substantially the same point, the liquid pressure cooperating with a spring to urge the valve means toward the open position. This expansion valve regulates the flow of liquid refrigerant to the evaporator in response to the amount of subcooling in the liquid refrigerant flowing past the point monitored by the thermal-sensing means.

Another important objective is realized in that the thermal-sensing means includes a charge having pressure-temperature characteristics that are related to the pressure-temperature characteristics of the refrigerant in the system to provide predetermined subcooling temperature values over a range of condensing temperatures.

Depending upon the application of the refrigeration system, it is at times advantageous to utilize a charge having pressure-temperature characteristics that are so related to the pressure-temperature characteristics of the refrigerant so as to maintain a substantially constant subcooling temperature value over a range of condensing temperatures. At other times it is advantageous to utilize a charge having pressure-temperature characteristics that are related to the pressure-temperature characteristics of the refrigerant to provide increasing subcooling temperature values under respective decreasing condenser temperatures in a range of condensing temperatures.

Yet another important object is attained by utilizing a charge that is any one of a class including (1) trifluoroethane and (2) carbon dioxide combined with carbon, in a refrigeration system in which the refrigerant is monochlorodifluoromethane, whereby to achieve the above advantageous functional results.

An important object is afforded by utilizing a charge consisting of carbon dioxide combined with carbon in a refrigeration system using any of the well known refrigerants so as to provide the above functional results.

Another important objective is achieved in a refrigeration system utilizing monochlorodifluoromethane as the refrigerant in which the charge is any one of a class including (1) monochlorodifluoromethane and (2) an isotropic mixture of monochlorodifluoromethane and monochloropentafluoroethane in a ratio of approximately 48.8% to 51.2%, respectively, the charge pressure being assisted by a cooperating spring on the same side of the

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flexible motor element which tends to urge the valve means toward the closed position, whereby the pressure-temperature characteristics of the combined charge and spring closely approximate the desired pressure-temperature characteristics needed to operate at a constant subcooling over a range of condensing temperatures.

Still another important object is realized by using a charge of cyclopropane in a refrigeration system in which the refrigerant is dichlorodifluoromethane, whereby to provide the above stated functional results.

An important object is attained by the provision of a bypass passage that interconnects the valve inlet and outlet to provide initial flow upon starting the system and to provide subcooling sufficient to open the expansion valve, the bypass passage being of a predetermined size so that its capacity is less than that required for the system at minimum load conditions.

Another important object is afforded by the structural arrangement in which a valve seat defines a valve port in the passage, and the valve means moves into the valve port in the closed position of the valve means, the valve means and valve seat having sufficient clearance to provide initial flow upon starting the system and to provide subcooling sufficient to open the expansion valve, the clearance being predetermined so that its capacity is less than that required for the system at minimum load conditions. In addition, this construction provides an override that prevents the valve means from being forced against the valve seat by the flexible motor element under excessive pressure exerted by the charge.

It is another important objective to provide an expansion valve in which the full force of the diaphragm under charge pressure is prevented from being exerted on the valve pin and/or on the valve seat when the valve pin is in the closed position. This advantageous result is achieved by the provision of a resilient means operatively between the flexible motor element and the valve means tending to urge the valve means toward or against the valve seat and causing the valve means to follow the flexing action of the motor element before the valve means engages the valve seat, and by the provision of limit means engaging and stopping movement of the flexible motor element against the charge pressure after the valve means has engaged the valve seat. Again, it will be understood that this construction enables the motor element and associated parts to override a short distance downward after the valve means is seated thereby preventing the valve means from being forced against the valve seat by the flexible motor element under excessive pressure exerted by the charge, yet assuring that the valve means is closed, and allowing for manufacturing tolerances.

Another important objective is to provide an expansion valve in which the pressure forces exerted on the valve means are balanced so as to prevent the pressure difference between the valve inlet and outlet from becoming a significant factor in the control characteristic and operation of the valve.

An important object is to provide an expansion valve that is simple and durable in construction, economical to manufacture and assemble, highly efficient in operation, and which will operate automatically to control the amount of liquid subcooling obtained at the exit of the refrigerant condenser in a refrigeration system.

The foregoing and numerous other objects and advantages of the invention will more clearly appear from the following detailed description of several embodiments, particularly when considered in connection with the ac-

companying drawing, in which:

FIG. 1 is a diagrammatic view illustrating the refrigeration system;

FIG. 2 is a cross-sectional view of the expansion valve;

FIG. 3 is a fragmentary cross-sectional view showing a modification of the expansion valve;

FIG. 4 is a fragmentary cross-sectional view illustrating another modification of the expansion valve;

FIG. 5 is a chart illustrating the pressure-temperature characteristics in a system utilizing monochlorodifluoromethane as the refrigerant and utilizing trifluoroethane as the charge;

FIG. 6 is a chart illustrating the pressure-temperature characteristics of a system utilizing monochlorodifluoromethane as the refrigerant and utilizing monochlorodifluoromethane combined with a spring pressure as the charge;

FIG. 7 is a chart illustrating the pressure-temperature characteristics of carbon dioxide and carbon when utilized as a charge in a refrigeration system in which monochlorodifluoromethane is used as the refrigerant;

FIG. 8 is a chart illustrating the pressure-temperature characteristics in a refrigeration system utilizing monochlorodifluoromethane as the refrigerant and utilizing a spring in combination with an isotropic mixture of monochlorodifluoromethane and monochloropentafluoroethane in a ratio of approximately 48.8% to 51.2%, respectively, as the charge;

FIG. 9 is a cross-sectional view of a modified expansion valve, and

FIG. 10 is a fragmentary cross-sectional view of the modified expansion valve of FIG. 9, taken in a longitudinal, right angle plane.

Referring now by characters of reference to the drawing, and first to FIG. 1, it will be understood that the refrigeration system includes a motor-compressor unit 10 connected by discharge line 11 to the inlet of condenser 12, the outlet of condenser 12 being operatively connected to the inlet of evaporator 13 by a liquid line 14, and the outlet of evaporator 13 being connected to the motor-compressor unit 10 by a suction line 15. An expansion valve 16 is connected in the liquid line 14, the expansion valve 16 including a power unit 17 having a control bulb 18 located in thermo-responsive relation to the liquid line 14 at a point closely adjacent the outlet of condenser 12. This expansion valve 16 regulates the flow of liquid refrigerant to the evaporator 13 in response to the amount of subcooling in the liquid refrigerant flowing past the control bulb 18.

The expansion valve 16 includes a valve body 20 having an inlet fitting 21 that defines an inlet 22 communicating with that portion of the liquid line 14 leading from the outlet of condenser 12, and having an outlet fitting 23 that defines an outlet 24 communicating with that portion of the liquid line 14 leading to the inlet of evaporator 13. Extending across an internal passage formed in valve body 20 and interconnecting the valve inlet 22 and valve outlet 24 is a partition 25. A valve plug 26 is threadedly attached to the body partition 25, the plug 26 including a valve port 27 defined by a valve seat 28.

Reciprocatively mounted in the valve body 20 is a valve pin 30 constituting a valve means, the valve pin 30 including a portion 31 on the inlet side of the valve port 27 that is movable toward or engages the valve seat 28 to regulate flow through the valve port 27 and hence through the passage interconnecting the valve inlet 22 and valve outlet 24. For reasons which will later appear, the valve pin 30 includes a reduced upper stem 32, and a reduced lower stem 33 extending through the valve port 27 and into the passage on the outlet side of such valve port 27.

Located in the chamber 34 constituting a part of the flow passage, on the outlet side of the valve port 27 is a compression spring 35. One end of spring 35 seats on

nut 36 threadedly connected internally of the outlet fitting 23. The opposite end of spring 35 seats on a spring guide 37, the spring guide 37 engaging the stem 33 of valve pin 30. The spring 35 tends to urge the valve pin 30 toward an open position. Adjustment of the axial position of nut 36 determines the force exerted by spring 35 on the push rods 50 and thus on the underside of diaphragm 43.

Formed in the body partition 25 is a small bypass passage 40 that directly interconnects the valve inlet 22 and the valve outlet 24. This bypass passage 40 is provided to enable initial flow through the expansion valve 16 upon starting the system and to enable subcooling of the liquid refrigerant sufficient to open the expansion valve 16. The bypass passage 40 is of a predetermined size so that its capacity is less than that required for the system at minimum load conditions.

The power unit 17 includes an upper housing 41 peripherally secured to a cooperating lower housing 42, and a diaphragm 43 constituting a flexible motor element, clamped between the housings 41 and 42. The diaphragm 43 forms an upper compartment 44 at one side and a lower compartment 45 at the opposite side. Secured to the underside of and movable with the diaphragm 43 is a buffer plate 46. The valve body 20 includes a top abutment 47 constituting a limit means, engaging and stopping the movement of the buffer plate 46, and hence limiting the flexing movement of diaphragm 43 in one direction.

Interconnecting the buffer plate 46 and the spring guide 37 are a plurality of reciprocatively mounted push rods 50 (one of which is shown in FIG. 2) so that the force of spring 35 is exerted through buffer plate 46 on the underside of diaphragm 43.

The valve pin 30 is urged against the spring guide 37 by the spring 51. Consequently the valve pin 30 follows the motion of spring guide 37 until the valve pin 30 contacts the valve seat 28. Because the spring guide 37 is urged against the push rods 50, which are in contact with the buffer plate 46 in contact with the diaphragm 43, the valve pin 30 follows the motion of the diaphragm 43 throughout the stroke of the valve pin 30 until the valve pin 30 contacts the valve seat 28. After the valve pin 30 contacts the valve seat 28, the diaphragm 43, buffer plate 46, push rods 50 and spring guide 37 are allowed to override and move downward an additional distance before the buffer plate 46 strikes the limit abutment 47. This override construction assures that the valve pin 30 is closed and allows for manufacturing tolerances. It is important to note that after the buffer plate 46 strikes the limit abutment 47, and the valve pin 30 is urged against the valve seat 28 by spring 51, there is still a clearance between the buffer plate 46 and the top of valve pin 30. This clearance prevents the force of the diaphragm 43 from being exerted on the valve pin 30 and against the valve seat 28.

A thermal-sensing means includes the control bulb 18 connected by a capillary tube 52 to the upper housing 41 by fitting 53. The control bulb 18 is in communication with the upper compartment 44 at one side of the diaphragm 43 through the capillary tube 52. The control bulb 18 is charged with a volatile refrigerant or another type of charge as will be described later. The charge pressure in the control bulb 18 is exerted on the top of diaphragm 43.

The valve body 20 is provided with an internal equalizer passage 54 that places the lower compartment 45 on the bottom side of diaphragm 43 in communication with the inlet 22, and hence subjects the underside of the diaphragm 43 to the liquid pressure that corresponds to the saturated condensing pressure in condenser 12 (less any pressure drop because of friction losses which are assumed to be negligible). The internal equalizer passage 54 is used on systems where the pressure drop between the control bulb 18 and the valve inlet 22 is low. Where ex-

cessive pressure drop exists, an external equalizer (not shown) can be used whereby the underside of the diaphragm 43 is vented to a separate connection on the valve body, which in turn is connected by means of tubing to the liquid line 14 at the location of control bulb 18.

The operation of the expansion valve 16 is thought to be apparent from the foregoing detailed description of parts, but for completeness of disclosure the usage of the valve 16 in the refrigeration system and its operation will be briefly described. The valve 16 responds to and controls the amount of liquid subcooling at the control bulb 18 in the refrigeration system. The liquid pressure at the valve inlet 22 is exerted through internal equalizer passage 54 to the underside of diaphragm 43. The spring force from spring 35 is also exerted on the underside of diaphragm 43 through the push rods 50 and the buffer plate 46. The control bulb 18 is provided with a charge, the pressure of which is exerted on the top side of diaphragm 43.

When the expansion valve 16 is controlling, the charge pressure equals the sum of the liquid pressure in the system plus the pressure equivalent of the spring force from spring 35. When the charge pressure exceeds the liquid pressure plus the pressure force of spring 35, the diaphragm 43 is forced downwardly, thereby moving the push rods 50 downwardly and increasing the compression on spring 35 until equilibrium of forces is established. A downward motion of diaphragm 43 is transmitted by push rods 50 to the spring guide 37. The valve pin 30 is held in contact with the spring guide 37 by compression spring 51. The motion of the valve pin 30 follows the motion of diaphragm 43 until the valve pin 30 contacts the valve seat 28, at which time the expansion valve 16 is closed. If the charge pressure is less than the sum of the liquid refrigerant pressure plus the pressure force of spring 36, the diaphragm 43 and valve pin 30 move upwardly to open the expansion valve 16. The compression spring 51 is used only to maintain contact between valve pin 30 and spring guide 37 except when the valve pin 30 is in contact with the valve seat 28, at which time there is a net upward force on buffer plate 46 and consequently on diaphragm 43. This small spring force is of no importance in the operation of the valve 16 at this point.

For purposes of illustration, it will be assumed that the effective area of diaphragm 43 is one square inch so that the pressure can be converted to force without calculation. It will also be assumed that the refrigeration system shown in FIG. 1 utilizes monochlorodifluoromethane, also known as Refrigerant-22, as the refrigerant.

The graph of FIG. 5 illustrates the pressure-temperature characteristics of monochlorodifluoromethane used as the refrigerant and trifluoroethane, also known as Refrigerant-143(a), as the bulb charge. The curve A represents the pressure-temperature characteristic of the system refrigerant monochlorodifluoromethane. The next adjacent and parallel curve B shows the pressure-temperature characteristic of the system refrigerant monochlorodifluoromethane augmented by a spring force which is the total opening force. The curve C represents the desired pressure-temperature characteristic of the bulb charge so as to realize a substantially constant subcooling rate over the entire range of condensing temperatures from approximately 140° F. to 60° F. The curve D represents the pressure-temperature characteristic of trifluoroethane used as the bulb charge. It will be noted that the curve D closely approximates the desired characteristic of curve C. The horizontal distance in degrees F, at any particular condensing pressure, between the curve B (total opening force) and the charge pressure curve D (total closing force) represents the number of degrees of subcooling at which the valve will control.

In FIG. 5, the substantially horizontal line E illustrates the subcooling temperature versus the condensing temperature, and illustrates that the subcooling tempera-

ture values are essentially constant when trifluoroethane is utilized as the bulb charge.

EXAMPLE 1.—R-22 (SYSTEM) AND R-143(A) (BULB), ASSUME DIAPHRAGM=1 SQ. INCH

	R-143(a) (Bulb)	R-22 (System)
Saturation pressure, p.s.i.g. ....	273.8	260.0
Temperature (condensing), ° F. ....	110	120
Temperature (bulb), ° F. ....	110	-----
Force on diaphragm top, lbs. ....	273.8	-----
Spring force, lbs. ....	-----	13.8
Force on diaphragm bottom, lbs. ....	-----	273.8
Subcooling, ° F. ....	10	-----

From the above table of Example 1 and the graph of FIG. 5, it will be noted that at a condensing temperature of 120° F. the saturated condensing pressure is 260.0 p.s.i.g. and that this pressure is exerted on the underside of diaphragm 43. The spring force is 13.8 lbs. and is also exerted on the underside of diaphragm 43. With 10° F. of subcooling at the control bulb 18, the charge temperature is equivalent to the saturated condensing temperature minus 10° F. or 110° F. The charge trifluoroethane has a saturation pressure of 273.8 p.s.i.g. at 110° F. Thus, the forces on the two sides of the diaphragm 43 are equal (260.0 p.s.i.g. plus 13.8 lbs. on the underside and 273.8 p.s.i.g. on the top), with 10° F. subcooling at the control bulb 18.

If the subcooling increased because of an increased pumping rate of the motor-compressor unit 10 (thus raising the level of the liquid in condenser 12 and allowing it to be further subcooled) the expansion valve 16 responds to a lower bulb temperature and/or a higher liquid pressure in the system, causing the valve 16 to open wider until a new equilibrium is established.

EXAMPLE 2.—SUBCOOLING INCREASED (INCREASED COMPRESSOR RATE)

	R-143(a) (Bulb)	R-22 (System)
Saturation pressure, p.s.i.g. ....	276.5	263.4
Temperature (condensing), ° F. ....	110.7	121
Temperature (bulb), ° F. ....	110.7	-----
Force on diaphragm top, lbs. ....	276.5	-----
Spring force, lbs. ....	-----	13.1
Force on diaphragm bottom, lbs. ....	-----	276.5
Subcooling, ° F. ....	10.3	-----

Thus, a new balance point might be 121° F. condensing temperature at which the liquid pressure is 263.4 p.s.i.g., and the spring force is slightly less at approximately 13.1 lbs. as the valve 16 is opened wider, thus giving a total opening force on the underside of diaphragm 43 of 276.5 p.s.i.g. At 110.7° F., the charge pressure would be 276.5 p.s.i.g., thus controlling at 10.3° F. subcooling at bulb 18.

Conversely, if the subcooling temperature value decreased because of a lower pumping rate of the motor compressor unit 10 (this happens if the evaporator temperature is lowered) the expansion valve 16 responds by moving to a more nearly closed position.

EXAMPLE 3.—SUBCOOLING DECREASED (DECREASED COMPRESSOR RATE)

	R-143(a) (Bulb)	R-22 (System)
Saturation pressure, p.s.i.g. ....	266.9	252.9
Temperature (condensing), ° F. ....	108.15	118
Temperature (bulb), ° F. ....	108.15	-----
Force on diaphragm top, lbs. ....	266.9	-----
Spring force, lbs. ....	-----	14.0
Force on diaphragm bottom, lbs. ....	-----	266.9
Subcooling, ° F. ....	9.85	-----

A new balance point might now be 118° F. condensing temperature at 252.9 p.s.i.g., and a spring force of 14.0 lbs. giving a total opening force of 266.9 p.s.i.g. A

bulb temperature of 108.15° F. would produce a charge pressure of 266.9 p.s.i.g., thus the expansion valve 16 would control at 9.85° F. subcooling.

It can be seen that the expansion valve 16 can be made to control over a wide range of condensing temperatures if the refrigerant charge used in the control bulb 18 has the appropriate pressure-temperature characteristic.

EXAMPLE 4.—SUBCOOLING DECREASED (LOW CONDENSING TEMPERATURE)

	R-143(a) (Bulb)	R-22 (System)
Saturation pressure, p.s.i.g.-----	157.4	143.6
Temperature (condensing), ° F-----	72.7	80
Temperature (bulb), ° F-----	157.4	-----
Force on diaphragm top, lbs.-----	-----	13.8
Spring force, lbs.-----	-----	157.4
Force on diaphragm bottom, lbs.-----	-----	7.3
Subcooling, ° F-----	7.3	-----

For instance, at a condensing temperature of 80° F. the saturated liquid pressure of the system refrigerant monochlorodifluoromethane is 143.6 p.s.i.g. Assuming a valve opening as before, the spring force is 13.8 lbs. and the total opening force is 157.4 lbs. At 72.7° F., the charge refrigerant trifluoroethane in the bulb 18 has a pressure of 157.4 p.s.i.g. Thus, the subcooling at bulb 18 is 80° F. minus 72.7° F. or 7.3° F.

The amount of subcooling that the expansion valve 16 will maintain can be adjusted by changing the compressive force of spring 35 upon manipulation of internal nut 36. Turning the nut 36 inwardly increases the force of spring 35 and results in a decrease of the amount of subcooling. Turning the nut 36 outwardly decreases the force of spring 35 and results in more subcooling.

There are two other methods of obtaining the appropriate pressure-temperature characteristics for the power unit 17 of an expansion valve 16 for a system utilizing monochlorodifluoromethane as the refrigerant. One of these is to use a bulb 18 filled with activated carbon and charged with carbon dioxide to a specific pressure. As an example, 1.5 cubic inches of coconut shell charcoal was placed in a bulb 18 having a volume of 1.5 cubic inches. The charging pressure of the carbon dioxide was 273 p.s.i.g. at 110° F. This charge works on the principle of adsorption and desorption of a gas by a solid as a function of temperature.

By varying the volume of the bulb 18 and/or the amount and type of carbon used, as well as the charging pressure of the carbon dioxide, a suitable pressure-temperature characteristic can be obtained. Reference is made to the graph of FIG. 7 in which the curve F illustrates the desired pressure-temperature characteristic of the bulb charge adapted to provide an essentially constant subcooling rate over the operating range of condensing temperatures in a system utilizing monochlorodifluoromethane as the refrigerant. The curve G in FIG. 7 represents the pressure-temperature characteristic of the carbon dioxide and carbon charge. It will be noted that these two curves F and G are very close.

By varying the amount of charcoal, the volume of bulb 18 and the charging pressure, the carbon dioxide and carbon charge can be designed to give the appropriate pressure-temperature characteristics for systems employing various refrigerants.

Another method is to utilize either monochlorodifluoromethane or an isotropic mixture of monochlorodifluoromethane and monochloropentafluoroethane (also known as Refrigerant-115) in a ratio of approximately 48.8% to 51.2%, respectively, also known as Refrigerant-502 as a bulb charge. Both of these charges alone have too low a pressure to be suitable. Therefore, it is necessary to modify the construction of the power unit 17 slightly as illustrated in FIG. 3.

This modified power unit includes an inverted cup 55

attached to the upper housing 41, the cup 55 receiving a compression spring 56. Another buffer plate 57 is secured to the top side of diaphragm 43 and is movable therewith. One end of spring 56 engages the cup 55, while the opposite end engages the buffer plate 57, the compressive force of spring 56 supplementing the charge pressure tending to urge the diaphragm 43 downwardly and tending to urge the valve pin 30 toward its closed position.

The graph of FIG. 6 illustrates the pressure-temperature characteristic of a bulb charge consisting of monochlorodifluoromethane supplemented by a spring pressure as used in a system in which the refrigerant is monochlorodifluoromethane. The curve H illustrates the pressure-temperature characteristic of the system refrigerant monochlorodifluoromethane. The broken-line curve I illustrates the pressure-temperature characteristic of the total opening force provided by the system refrigerant monochlorodifluoromethane supplemented by a 15-lb. spring. The curve J illustrates the pressure-temperature characteristic of the bulb charge consisting of monochlorodifluoromethane supplemented by a 55-lb. spring force, representing the total closing force.

The slightly curved line K illustrates a slight increasing amount of subcooling temperature values at the lower condensing temperatures in a refrigeration system of this type. In some applications, it is desirable to have this type of pressure-temperature characteristic because it serves to back liquid up into the condenser 12 at low condensing temperatures and maintain a higher than normal condensing pressure. This is useful at low ambient temperatures at the condenser 12 to maintain sufficient pressure difference across the expansion valve 16. With systems utilizing thermostatic expansion valves, special means are employed in the industry to achieve this result. With the present liquid subcooling valve 16 and an appropriate bulb charge of the type described above, no special means would be necessary.

The graph of FIG. 8 illustrates the pressure-temperature characteristics of a bulb charge consisting of an isotropic mixture of monochlorodifluoromethane and monochloropentafluoroethane in a ratio of approximately 48.8% to 51.2%, respectively, in a refrigeration system utilizing monochlorodifluoromethane as the refrigerant. The curve L illustrates the pressure-temperature characteristic of the system refrigerant monochlorodifluoromethane supplemented by 15-lb. spring force, which represents the total opening force. The curve M illustrates the desired pressure-temperature characteristic of the bulb charge to provide a substantially constant subcooling. The curve N illustrates the pressure-temperature characteristic of the specific bulb charge supplemented by a 20-lb. spring force, representing the total closing force. The substantially horizontal line O illustrates the subcooling characteristic that is obtained in this system.

Corresponding results are obtained in a refrigeration system in which dichlorodifluoromethane also known as Refrigerant-12 is used as the system refrigerant and cyclopropane is utilized as the bulb charge.

An important feature is the inclusion of the bypass passage 40 in the expansion valve 16 (FIG. 2). This permanent bleed around the valve port 27 is necessary to the operation of the expansion valve 16. Without the permanent bleed, this expansion valve 16 would remain closed when the system is started because the liquid refrigerant in the condenser 12 at start-up is at saturation temperature and thus the valve 16 is in the closed position. It is possible for the motor-compressor unit 10 to pump out the evaporator 13 without raising the condensing pressure sufficiently to provide subcooling to open the expansion valve 16. With the evaporator 13 empty and no flow, the condensing pressure would remain low and the valve 16 would remain closed. The bypass passage 40 around the valve port 27 provides the initial flow to provide subcooling to open the valve 16. This bypass pas-

sage 40 is sized so that its capacity is less than that required for the system at minimum load conditions.

The structural arrangement and mounting of the valve pin 30 provides a necessary override feature for the valve pin 30. The motion of diaphragm 43 is stopped when the buffer plate 46 contacts the body abutment 47. If the valve pin 30 were directly in contact with the buffer plate 46 without the provision of the intermediate spring 51, the full force of diaphragm 43 would be able to force the valve pin 30 against the valve seat 28. During shipment of the expansion valve 16 before installation on the system, the pressure under the diaphragm 43 would be atmospheric and the pressure on the top side could go as high as 300 p.s.i.g. at 120° F., as for example when an isotropic mixture of monochlorodifluoromethane and monochloropentafluoroethane in a ratio of approximately, 48.8% to 51.2%, respectively, is used as the bulb charge supplemented by the force of a spring 56, on top of diaphragm 43. This total force would be exerted on valve pin 30 and valve seat 28 as well as on the buffer plate 46. The present construction prevents this undesirable situation.

Another override construction is shown in FIG. 4. In this modification, there is no provision as such of a separate bypass passage 40 of the type shown in FIG. 2. In lieu thereof, the valve plug 26 is provided with an oversized valve port 58 larger than the external diameter of valve pin 30. Moreover, it will be noted that the upper end of the valve pin 30 directly engages the buffer plate 46 and is moved therewith. Because of the larger size of the valve port 58, the valve pin 30 can move into the valve port 58 in the closed position of the valve pin 30, the valve pin and the plug margin defining the valve port 58 having sufficient clearance to provide initial flow upon starting the system and to provide subcooling sufficient to open the expansion valve 16. This clearance is predetermined so that its capacity is less than that required for the system at minimum load conditions.

A modified subcooling control valve 60, shown in FIGS. 9 and 10, is designed for large capacity systems. As the port size of a subcooling control valve is increased to handle increased refrigerant flow, a point is reached where the port area is of sufficient size so that the pressure difference across the port exerts a significant force on the valve pin. This pressure force will tend to open or close the valve depending on the direction of flow through the port.

For example, considering the valve 16 in FIG. 2, the pressure force acts in a closing direction on the valve pin 30. With a condensing pressure of 250 p.s.i. and an evaporator pressure of 50 p.s.i., the pressure difference across the port is 200 p.s.i. With a port diameter of 0.125, the area is 0.012272 square inch, and the force exerted on the valve pin 30 in a closing direction is 2.5 lbs. With a port diameter of 0.250, the area is 0.049 square inch, and the force on the valve pin 30 would be 9.8 lbs. To keep this force from becoming a significant factor in the operation of the valve 16, a larger diaphragm area would be needed. The valve 60 in FIGS. 9 and 10 eliminates the need for a larger diaphragm area by balancing the pressure forces on the valve pin.

The expansion valve 60 includes a valve body 61 having an inlet fitting 62 that defines an inlet 63 communicating with that portion of the liquid line 14 leading from the outlet of condenser 12, and having an outlet fitting 64 that defines an outlet 65 communicating with that portion of the liquid line 14 leading to the inlet of evaporator 13. A valve plug 66 is threadedly attached to body partition 67, the plug 66 including a valve port 70 defined by a valve seat 71.

Reciprocally mounted in the valve body 61 is a valve pin 72 constituting a valve means, the valve pin 72 including an enlarged portion 73 on the outlet side of the valve port 70 that is movable toward or away from the valve seat 71 to regulate flow through the valve port

70. A reduced lower stem 74 of the valve pin 72 extends through the valve port 70 and into the passage on the inlet side of the valve port 70.

A compression spring 75 is located in chamber 76. One end of the spring 75 seats on a holder 77 abutting against a bottom cap 80 threadedly attached to the valve body 61. The opposite end of spring 75 bears against a spring guide 81 reciprocally mounted in the chamber 76. The spring 75 tends to urge the valve pin 72 toward an open position.

The body partition 67 is provided with a small bypass passage 82 that directly interconnects the valve inlet 63 and the valve outlet 65. This bypass passage 82 enables initial flow through the expansion valve 60 upon starting the system, and enables subcooling of the refrigerant sufficient to open the valve 60. The bypass passage 82 is of a predetermined size so that its capacity is less than that required for the system at minimum load conditions.

The power unit generally indicated by 83 conforms exactly to the power unit 17 previously described with respect to valve 16 in FIG. 2. The component parts of the power unit 83 are given corresponding reference numbers.

Interconnecting the buffer plate 46 and the spring guide 81 are a pair of push rods 84 (FIG. 10) so that the force of spring 75 is exerted through buffer plate 46 and on the underside of diaphragm 43.

This expansion valve 60 utilizes a thermal-sensing means identical to that described above with respect to valve 16 in FIG. 2, and corresponding reference numbers will be utilized wherever possible to indicate the component parts. The control bulb 18 is charged with a volatile refrigerant or other type of charge previously described with respect to the structure and operation of expansion valve 16.

From FIG. 9, it is seen that the valve body 61 is provided with an internal equalizer passage 85 that places the lower compartment 45 on the bottom side of diaphragm 43 in communication with the inlet 63, and hence subjects the underside of the diaphragm 43 to the liquid pressure that corresponds to the saturated condensing pressure in condenser 12. A relatively weak compression spring 86 constituting a resilient means, is located about the valve pin 72, one end of the spring 86 engaging the valve pin 72 and the opposite end engaging the buffer plate 46. The spring 86 corresponds to the spring 51 previously described with respect to the valve 16 in FIG. 2, and performs the same override function.

The refrigerant flow through this valve 60 is in the opposite direction of the flow through the smaller valve 16 shown in FIG. 2. The refrigerant enters through the inlet 63, flows through the annular space between the valve port 70 and the valve pin 72, expands through the opening formed by valve port 70 and the valve pin 72, and flows out through the outlet 65 to the evaporator 13 of the system. In addition, a portion of the refrigerant flows through the bypass passage 82.

The liquid pressure at the valve inlet 63 is exerted on an area of the valve pin 72 equal to the area of the valve port 70, and this pressure tends to push the valve pin 72 in an opening direction. The liquid pressure at the valve inlet 63 is also transmitted to the lower compartment 45 on the bottom side of diaphragm 43 through equalizer passage 85, where such pressure acts on the top of valve pin 72 in a closing direction.

The diameter of valve pin 72 subjected to the liquid pressure in lower compartment 45 under the diaphragm 43 is made equal to the diameter of the valve port 70 so that the opening and closing forces on the valve pin 72 are equal and opposite, and thus cancel one another. This construction provides a balanced port such that the pressure difference between valve inlet 63 and outlet 65 has no effect on the control characteristic of the valve 60. The valve pin 72 is made to fit closely in its compatible opening through the valve body 61 to prevent ex-



cessive leakage from inlet 63 to outlet 65 of the valve 60, because it is desirable to control such leakage by the diameter of the bypass passage 82.

In addition to having the additional unique feature of a balanced port, the valve 60 incorporates the override construction described in connection with the smaller valve 16 in FIG. 2, and incorporates the auxiliary bypass passage 82 similar to the passage 40 in the smaller valve of FIG. 2 required for desired operation.

Although the invention has been described as making detailed reference to several embodiments, such detail is to be understood in an instructive, rather than in any restrictive sense, many variants being possible within the scope of the claims hereunto appended.

I claim as my invention:

1. In a refrigeration system having a compressor, condenser, and evaporator operatively interconnected:

- (a) an expansion valve connected in the line between the condenser and evaporator, the expansion valve comprising a valve body having an inlet connected to the line leading from the condenser, and an outlet connected to the line leading to the evaporator, and the valve body having a passage connecting the valve inlet and outlet,
- (b) a valve means movably mounted in the body for controlling flow through the passage,
- (c) spring means tending to urge the valve means toward an open position,
- (d) a flexible motor element carried by the body,
- (e) means operatively connecting the flexible motor element and the valve means,
- (f) thermal-sensing means subjecting one side of the motor element to a pressure that is a function of the temperature of the refrigerant at a point in the line between the condenser and the expansion valve, and which tends to urge the valve means toward a closed position, and
- (g) means subjecting the other side of the motor element to the liquid pressure of the refrigerant at substantially said point, the liquid pressure cooperating with the spring means and tending to urge the valve means toward the open position.

2. A refrigeration system as defined in claim 1, in which:

- (h) the thermal-sensing means includes a charge having pressure-temperature characteristics that are related to the pressure-temperature characteristics of the refrigerant in the system to provide predetermined subcooling temperature values over a range of condensing temperatures.

3. A refrigeration system as defined in claim 1, in which:

- (h) the thermal-sensing means includes a charge having pressure-temperature characteristics that are related to the pressure-temperature characteristics of the refrigerant in the system to maintain a substantially constant subcooling temperature value over a range of condensing temperatures.

4. A refrigeration system as defined in claim 1, in which:

- (h) the thermal-sensing means includes a charge having pressure-temperature characteristics that are related to the pressure-temperature characteristics of the refrigerant in the system to provide increasing subcooling temperature values under respective decreasing condenser temperatures in a range of condensing temperatures.

5. A refrigeration system as defined in claim 2, in which:

- (i) the refrigerant is monochlorodifluoromethane, and
- (j) the charge is any one of a group including (1) trifluoroethane and (2) carbon dioxide combined with carbon.

6. A refrigeration system as defined in claim 2, in which:

- (i) the charge is carbon dioxide combined with carbon.

7. A refrigeration system as defined in claim 2, in which:

- (i) a second spring is operatively connected to the flexible motor element and cooperates with the pressure exerted by the charge from the said one side of the motor element tending to urge the valve means toward the closed position,
- (j) the refrigerant is monochlorodifluoromethane, and
- (k) the charge is any one of a group including (1) monochlorodifluoromethane and (2) an isotropic mixture of monochlorodifluoromethane and monochloropentafluoroethane in a ratio of 48.8% to 51.2%, respectively.

8. A refrigeration system as defined in claim 2, in which:

- (i) the refrigerant is dichlorodifluoromethane, and
- (j) the charge is cyclopropane.

9. A refrigeration system as defined in claim 2, in which:

- (i) a bypass passage interconnects the valve inlet and outlet to provide initial flow on starting the system and to provide subcooling sufficient to open the expansion valve, the bypass passage being of a predetermined size so that its capacity is less than that required for the system at minimum load conditions.

10. A refrigeration system as defined in claim 2, in which:

- (i) a valve seat defines a valve port in the passage,
- (j) the valve means moves into the valve port in the closed position of the valve means, and
- (k) the valve means and valve seat have sufficient clearance to provide initial flow upon starting the system and to provide subcooling sufficient to open the expansion valve, the clearance being predetermined so that its capacity is less than that required for the system at minimum load conditions.

11. A refrigeration system as defined in claim 2, in which:

- (i) a valve seat defines a valve port in the passage,
- (j) resilient means is connected operatively between the flexible motor element and valve means and tends to urge the valve means toward or against the valve seat,
- (k) means interconnects the valve means and flexible motor element so that the valve means follows the movement of the motor element until the valve means engages the valve seat, and
- (l) limit means engages and stops movement of the flexible motor element against the pressure of the charge in the thermal-sensing means after the valve means engages the valve seat,
- (m) the resilient means enabling movement of the motor element a distance after the valve means is seated until halted by the limit means.

12. A refrigeration system as defined in claim 2, in which:

- (i) a valve seat defines a valve port in the passage,
- (j) the flexible motor element includes a buffer plate carried by and movable therewith,
- (k) the valve means includes a pin,
- (l) means including the spring means operatively interconnecting the valve pin with the flexible motor element so that the pin follows the motion of the motor element until the pin engages the valve seat,
- (m) another spring operatively between the valve pin and buffer plate tending to urge the pin toward or against the valve seat, and
- (n) limit means engages and stops movement of the buffer plate against the charge pressure in the thermal-sensing means after the valve pin engages the valve seat,
- (o) the said other spring holding the valve pin against the valve seat and permitting the buffer plate to override without subjecting the valve pin or valve seat to excessive pressure exerted by said charge.



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13. A refrigeration system as defined in claim 12, in which:

- (p) the valve pin is reciprocally mounted in the valve body in sealing relation, and is engageable with the valve seat on the outlet side of the valve port, 5
- (q) the pin has one end adjacent to the flexible motor element provided with a pressure surface subjected to the liquid pressure of the refrigerant at the said other side of the motor element and tending to urge the pin toward the valve seat, and 10
- (r) the area of the said pressure surface is equal to the area of the valve port so as to balance the liquid pressure forces exerted on the valve pin so that the pressure difference between valve inlet and outlet has no effect on the control characteristic of the valve. 15

14. A refrigeration system as defined in claim 2, in which:

- (i) a valve seat defines a valve port in the passage, 20
- (j) the valve means includes a pin engageable with the valve seat on the outlet side of the valve port,
- (k) the pin includes a pressure surface subjected to the liquid pressure of the refrigerant existing at the said other side of the motor element, and 25
- (l) the area of the pressure surface is equal to the area of the valve port to balance the pressure forces exerted on the valve pin so that the pressure difference between the valve inlet and outlet has no effect on the control characteristic of the valve. 30

15. An expansion valve for a refrigeration system, comprising:

- (a) a valve body having an inlet and an outlet interconnected by a passage for flow therethrough,
- (b) a valve seat defining a valve port in the passage, 35
- (c) a valve pin reciprocally mounted in the valve body and engageable with the valve seat, the pin extending through the valve port,
- (d) a spring mounted in the passage,
- (e) a spring guide engaging and urged by the spring against the valve pin and tending to urge the valve pin toward an open position, 40
- (f) a flexible motor element carried by the body including a buffer plate attached to and movable with the motor element,
- (g) means operatively interconnecting the valve pin and buffer plate, 45
- (h) push rods interconnecting the buffer plate and spring guide,
- (i) thermal-sensing means subjecting one side of the motor element to a charge pressure that is a function of the temperature of the refrigerant in the system and which tends to urge the valve pin toward the closed position, 50
- (j) means subjecting the other side of the motor element to the liquid pressure of the refrigerant in the system, the liquid pressure cooperating with the spring to urge the valve pin to the open position, and 55
- (k) means associated with the valve pin precluding the valve pin from being forced against the valve seat by the buffer plate under excessive pressure exerted by the charge. 60

16. An expansion valve as described in claim 15, in which:

- (l) the last said means includes a second spring located about the valve pin, one end of which engages the buffer plate and the other end of which engages the valve pin tending to urge the pin toward or against the valve seat, 65
- (m) the first said spring causes the valve pin to follow the action of the buffer plate until the valve pin engages the valve seat, and 70
- (n) limit means engages and stops movement of the buffer plate against the charge pressure in the thermal-sensing means after the valve pin engages the valve seat, 75

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- (o) the second spring prevents the valve pin from being forced against the valve seat by the buffer plate under excessive charge pressure and enables the buffer plate to override a distance before the buffer plate engages the limit means.

17. An expansion valve as defined in claim 16, in which:

- (p) a bypass passage interconnects the valve inlet and outlet to provide initial flow upon starting the system and to provide subcooling sufficient to open the expansion valve, the bypass passage being of a predetermined size so that its capacity is less than that required for the system at minimum load conditions.

18. An expansion valve as defined in claim 17, in which:

- (q) the valve pin is sealingly mounted in the valve body and is engageable with the valve seat on the outlet side of the valve port,
- (r) the liquid pressure of the refrigerant at the valve inlet is exerted on the valve pin through the valve port tending to urge the valve pin away from the valve seat,
- (s) the valve pin includes a pressure surface adjacent to the motor element and subjected to the liquid pressure of the refrigerant exerted on the said other side of the motor element, tending to urge the valve pin toward the valve seat, and
- (t) the area of the valve port is equal to the area of the pressure surface to balance the liquid pressure forces exerted on the valve pin so that the pressure difference between valve inlet and outlet has no effect on the control characteristic of the valve.

19. An expansion valve as defined in claim 15, in which:

- (l) the valve pin moves into the valve port in the closed position of the valve pin, and
- (m) the valve pin and valve seat have sufficient clearance to provide initial flow upon starting the system and to provide subcooling sufficient to open the expansion valve, the clearance being predetermined so that its capacity is less than that required for the system at minimum load conditions,
- (n) the movement of the valve pin into the valve port providing an override that prevents the valve pin from being forced against the valve seat by the buffer plate under excessive charge pressure.

20. An expansion valve as defined in claim 15, in which:

- (l) the valve pin is sealingly mounted in the valve body and engageable with the valve seat on the outlet side of the valve port,
- (m) the liquid pressure of the refrigerant in the system exerted through the valve port on the valve pin tends to urge the valve pin away from the valve seat,
- (n) the valve pin includes a pressure surface adjacent to the motor element subjected to the liquid pressure of the refrigerant exerted on the said other side of the motor element and tending to urge the valve pin toward the valve seat, and
- (o) the area of the valve port is equal to the area of the pressure surface so as to balance the liquid pressure forces exerted on the valve pin so that the pressure difference between the valve inlet and outlet have no effect on the control characteristic of the valve.

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