July 27, 1965 W. J. BARBIER 3,196,630

CONSTANT HORSEPOWER CONTROL VALVE

Filed July 31, 1961

Fig. 1.

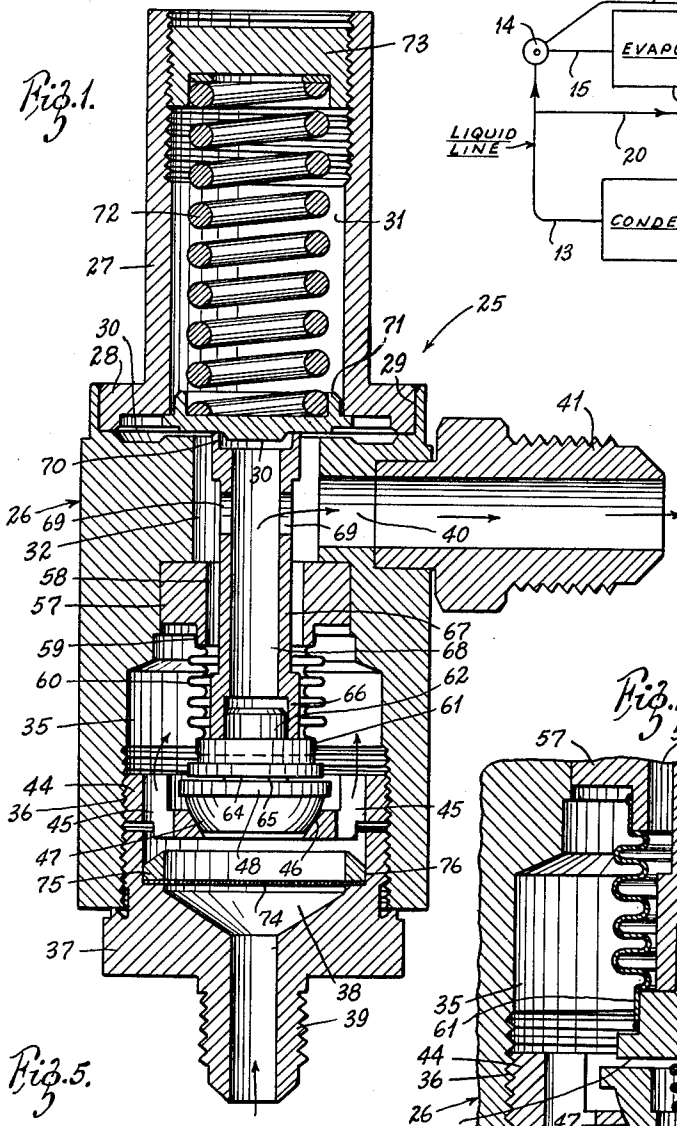


Fig. 4.

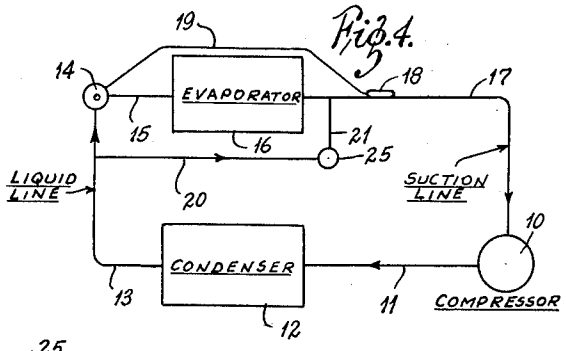


Fig. 3.

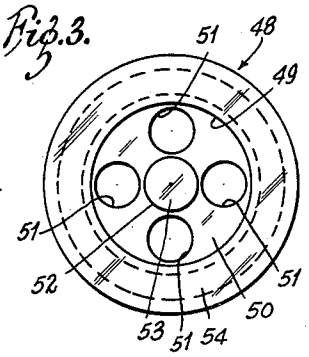


Fig. 2.

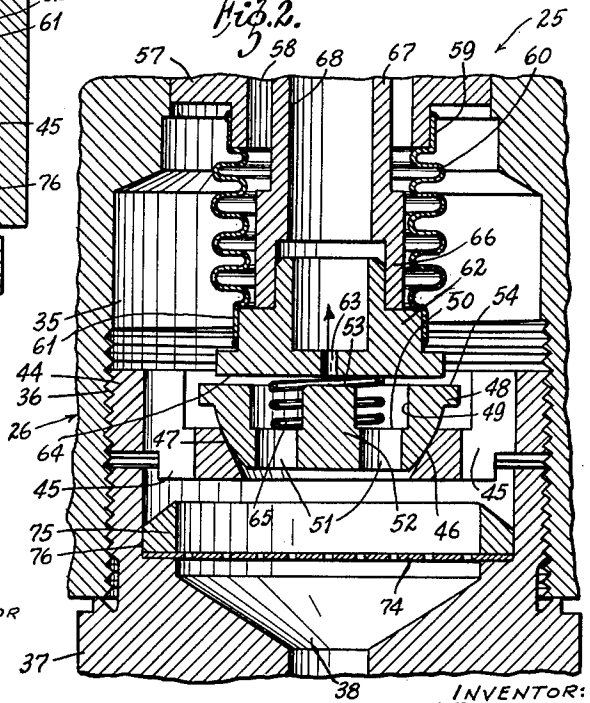
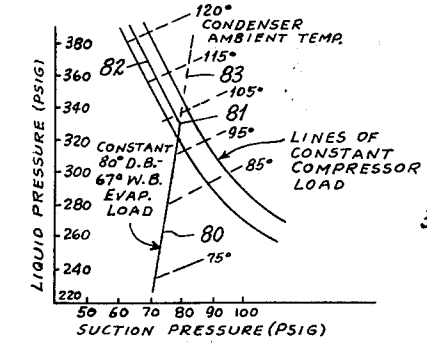


Fig. 5.



INVENTOR:
WILLIAM J. BARBIER,
BY Kingsland, Rogers, Ezell & Robbins
ATTORNEYS

United States Patent Office 3,196,630 Patented July 27, 1965

3,196,630

CONSTANT HORSEPOWER CONTROL VALVE

William J. Barbier, St. Louis, Mo., assignor to Alco Valve Company, St. Louis, Mo., a corporation of Missouri

Filed July 31, 1961, Ser. No. 128,191
5 Claims. (Cl. 62—197)

This invention relates to a control for a refrigeration system, and more particularly to a control for maintaining substantially constant compressor load in spite of variations in either condenser or evaporator outlet pressures, or high and low side pressures in the system.

In essence, the invention includes a conventional refrigeration system having a compressor, a condenser, an expansion device and an evaporator with pipe connections forming a closed circuit; the system is charged with a suitable refrigerant fluid. The refrigerant gas is driven by the compressor to the condenser where it is condensed to a liquid having a relatively high pressure. The pressure of this liquid is reduced when it passes through an expansion device, such as a valve. The low pressure liquid is delivered from the expansion device to the evaporator where it is evaporated to a sufficient extent to create a superheat condition at the evaporator outlet and in the suction line leading to the compressor inlet. This condition of superheat in the suction line is necessary so that the compressor will not be flooded with liquid. In such a conventional refrigeration system, compressor power consumption is increased with an increase in either condenser pressure or evaporator pressure.

This invention includes a pilot valve connected in a bypass conduit that bypasses the evaporator and feeds liquid refrigerant into the suction line downstream of the evaporator but upstream of the sensing bulb that controls the expansion valve. The pilot valve operates in an opening direction to feed a greater amount of liquid refrigerant with increases in either evaporator or condenser pressure, and this liquid refrigerant reduces the temperature of the refrigerant that the expansion valve bulb senses. Hence, the bulb effects a closing of the expansion valve to reduce the amount of refrigerant that is fed to the evaporator. This throttling of the expansion valve reduces the pressure within the suction line, thereby compensating for the higher pressure that had existed and caused the opening of the pilot valve. In other words, when condenser and/or suction pressures tend to increase, the resultant increase in compressor power normally expected is eliminated by the compensating effect of the pilot valve upon the expansion device.

It is therefore an object of this invention to provide a control for a refrigeration system to maintain constant compressor power consumption regardless of variations in condenser and evaporator pressures.

Another object of this invention is to reduce the temperature of the refrigerant that flows to the compressor inlet below that temperature at the evaporator outlet to eliminate potential compressor burnout caused by excessive superheat produced by the evaporator.

Still another object of this invention is to provide a bypass conduit for bypassing the evaporator of a refrigeration system, with valve means for regulating the flow of refrigerant through the bypass conduit to maintain substantially constant compressor power consumption over a predetermined range of operating conditions of the refrigeration system; in addition it is an object to provide such valve means wherein the starting point for the control may be varied according to specific operating requirements; and another object of the invention is to provide such a valve that can be adjusted to maintain substantially constant compressor power consumption regardless of the constant compressor load curves of the particular compressor involved.

Other objects and advantages will be apparent to those skilled in the art.

In the drawing:

FIGURE 1 is a longitudinal section view in side elevation of the pilot valve;

FIGURE 2 is a view in section of the lower part of the valve of FIGURE 1, but on an enlarged scale with the metering orifice shown in section;

FIGURE 3 is a plan view of the flow restrictor;

FIGURE 4 is a schematic diagram of a refrigeration system with an evaporator bypass conduit having the pilot valve connected into it; and

FIGURE 5 is a graph of liquid pressure vs. suction pressure showing the effect of the valve upon compressor load.

Referring now to the drawing, FIGURE 4 shows a conventional refrigeration system having a compressor 10, the outlet of which is connected by a pipe 11 to the inlet of a condenser 12. The outlet of the condenser 12 is connected by a pipe 13 to the inlet of an expansion device 14. The outlet of the expansion device 14 is connected by a pipe 15 to the inlet of an evaporator 16 and the outlet from the evaporator 16 is connected by a pipe 17 to the compressor inlet. The expansion device 14 has a temperature sensitive bulb 18 that is positioned alongside the suction pipe 17, downstream of the evaporator; the bulb is connected by a tube 19, usually a capillary tube, to the expansion device 14 for regulation of the expansion device.

This expansion device 14 is conventional and, as is known in the art, regulates the amount of superheat of the refrigerant leaving the evaporator. The bulb 18 is connected by the tube 19 to a chamber in the expansion device 14 and is charged with a fluid that responds to increasing temperatures in the pipe 17 to open the expansion device 14 and permit more refrigerant to flow into the evaporator. As the superheat at the evaporator outlet, sensed by the bulb 18, reduces, the lower pressure of the charge within the bulb 18 influences the expansion device 14 in a fluid flow throttling direction.

There is a bypass conduit 20 connected at one end of the pipe 13 downstream of the condenser; another pipe 21 is connected at one end to the suction pipe 17 at a point between the evaporator 16 and the bulb 18. The other end of the pipe 20 is connected to the inlet port of a pilot valve 25; the pipe 21 is connected to the outlet port of the valve 25.

The pilot valve 25 is shown in detail in FIGURES 1 and 2. It has a two part valve housing 26. The upper half 27 of the housing has a lower end 28 that is fitted within a recess 29 in the lower housing and is welded thereto. A flexible diaphragm 30 is clamped between the flange portion 28 and the lower side of the recess 29. The upper half 27 of the housing is hollow to define a spring chamber 31. The lower half of the housing has a chamber 32 in it that is separated from the chamber 31 by the diaphragm 30.

The lower end of the housing 26 has a recess within it that defines a chamber 35 with internal threads 36. An end plug 37 is threaded into the lower end of the housing 26. This end plug defines the lower side of the chamber 35 and has an inlet portion 38 through it. This inlet port 38 has a threaded fitting 39 for connection to the pipe 20 that bypasses the evaporator 16. There is an outlet 40 communicating with the chamber 32, and a pipe fitting 41 connects the outlet port 40 to the bypass pipe 21.

A flow restrictor insert 44 is threaded into the lower end of the housing 26 above the end plug 37. The insert 44 has holes 45 spaced about its periphery and a centrally disposed frusto-conical shaped seat 46. The seat 46 supports the spherical shaped lower side 47 of a flow restrictor 48. As show in FIGURE 2, the flow restrictor

48 has an annular recess 49 in its upper side, the bottom wall 50 of which acts as a spring seat. There are a plurality of holes 51 through the insert and in communication with the annular recess 49.

The central portion 52 of the insert has a flat upper surface 53 that is as high as the upper surface 54 that surrounds the outer periphery of the annular recess 49. These surfaces 53 and 54 are preferably machined together so that the annular surface 54 can bear against the fitting 62 when the valve is in its closed position to align the flow restrictor within the seat 46.

A bellows support 57 is fitted within the housing 26. The bellows support 57 is pressfitted or otherwise securely fastened to the housing. It has a hole 58 through its center for a purpose to appear. The upper end 59 of a collapsible bellows 60 is fastened in fluid tight relation to the lower end of the bellows support 57. The lower end 61 of the bellows has a fluid tight connection to and supports a fitting 62 that has a very small orifice 63 through it. The lower side 64 of the fitting 62 is flat. The orifice 63 is positioned opposite the face 53 of the flow restrictor 48.

There is a light spring 65 that bears against the face 64 of the fitting 62 and against the bottom 50 of the annular recess 49. The pressure of this spring 65 assures proper alignment of the flow restrictor within the annular seat 46 so that the face 53 of the flow restrictor will always be substantially parallel to the lower face 64 of the fitting 62.

It can be seen that the bellows support 57 and the bellows 60, together with the fitting 62 combine to separate the chambers 32 and 35 with the only communication between these chambers being the orifice 63 through the fitting 62. It can also be seen that this orifice 63 is throttled when the fitting 62 moves toward the flow restrictor 47, and the orifice is relatively opened when the fitting 62 moves away from the flow restrictor 48.

The lower end 66 of a tubular sleeve 67 is connected to the upper end of the fitting 62. The sleeve 67 has a hollow passage 68 through it that communicates with the orifice 63, and there are holes 69 through the side walls of the sleeve which are large enough to allow refrigerant fluid to flow freely through them. The upper end 70 of the sleeve 67 bears against the lower side of a spring seat 71 which is supported by the diaphragm 30. The lower end of a compression spring 72 bears against the top of the spring seat 71, and the upper end of the spring 72 bears against a stud 73 that is threaded into the top of the upper half 27 of the housing 26. It can be threaded more or less into the valve housing to adjust the pressure of the compression spring 71.

As shown near the lower end of the valve, there is a perforated filter 74 adjacent the inlet port 38. The filter 74 is held in place against the end cap 37 by a retainer ring 75 that is pressfitted or threaded within a recess 76 in the end cap.

Operation

From the foregoing description, it can be seen that the spring 72 acts downwardly upon the diaphragm 30 and this downward pressure is transmitted through the sleeve 67 to the fitting 62 which has the orifice 63 through it. Hence the spring 72 normally biases the fitting 62 in a direction to throttle fluid flow through the orifice 63. The force of the spring 72 is adjustable by changing the position of the plug 73.

The chamber 32 communicates by way of the outlet port 40 with the suction pressure that is in the pipe 17 and the pipe 21. This pressure acts in an upward direction upon the diaphragm 30 and in a downward direction upon the top of the fitting 62 within the area of the bellows 60. Hence the effective area upon which the suction pressure acts is in the area of the diaphragm 30 less the horizontal area of the bellows 60. The net effect of this suction pressure biases the diaphragm in an upward direction, and the

fitting 62 follows the diaphragm because of the force of the light spring 65, and because of the pressure within the chamber 35 to be described.

The liquid pressure within the pipe 20, which is the same as the pressure within the liquid line 13, also biases the fitting 62 in an upward direction. This liquid pressure fills the chamber 35 and operates upon the lower face 64 of the fitting 62 in an upward direction. The net upward pressure of the liquid pressure within the chamber 35 is in an upward direction upon an area of the fitting equal to the horizontal area of the bellows.

The balance of forces upon the movable fitting 62 may be set forth in equation form as follows:

$$\text{Spring force} = P_S(A-a)+P_{La}$$

$$\frac{\text{Spring force}}{P_{La}} = \frac{P_S}{P_L}\left(\frac{A}{a}-1\right)+1$$

Wherein P_S designates suction pressure, P_L designates liquid pressure, A is the area of the diaphragm 30, and a is the horizontal effective area of the bellows 60.

The graph of FIGURE 5 shows the effect of the valve 25 and the bypass pipes 20 and 21 upon compressor load. In the graph, suction pressure is plotted against liquid pressure for a refrigeration system incorporating this invention. The solid line 80 designates evaporator load for an 80° F. dry bulb and 67° F. wet bulb for the indicated condenser ambient temperatures. At the point 81, the pilot valve 25 begins to have an effect and the valve operates over the portion 82 of the evaporator load line. The dotted line 83 indicates what the evaporator load would be without the pilot valve 25. It can be seen from FIGURE 5 that the portion 82 of the evaporator load line parallels the lines of constant compressor load in that the pilot valve 25 causes suction pressure to vary inversely with liquid pressure and vice versa.

The point 81 at which the pilot valve 25 begins to have an effect upon the performance of the refrigeration system is controlled by the strength of the compression spring 72. The slope of the line 82 is regulated by the relationship between the area of the diaphragm 30 and the area of the bellows 60. Hence, by a variation of this relationship, the pilot valve 25 can be made to maintain a constant compressor load for any given compressor, regardless of its operating characteristics. The most convenient way to adjust the ratio between the areas is to change the bellows 60.

It is also possible to change the slope of the portion 82 of the evaporator load curve by varying the size of the orifice 63. However, if the orifice 63 is constant and the ratio of the area of the bellows 60 to the diaphragm 30 is constant, the portion 82 of the evaporator load line will assume a straight line of constant slope.

The effect of the bypass pipes 20 and 21 and the valve 25 is to introduce small amounts of liquid into the pipe 17 upstream of the bulb 18 to reduce the temperature at the bulb 18. Upon an increase in either condenser pressure or suction pressure, the pilot valve opens further and a greater volume of liquid is admitted into the suction line 17. This greater amount of liquid reduces the temperature sensed by the bulb 18 and, therefore, causes the expansion device 14 to move in a throttling direction.

The effective areas upon which the liquid pressure within the chamber 35 operates and the suction pressure within the chamber 32 operates are important in the proper operation of the valve. As stated, each of these pressures has an opening effect upon the orifice that eventually produces a reduction in refrigerant pressure within the suction line 17. The net area $(A-a)$ that responds to suction pressure to admit colder liquid to the suction line 17 must be such as to produce an ultimate compensation for increases in suction line pressure caused by refrigerant conditions at the evaporator outlet. In other words, assuming constant condenser pressure, the pressure in the chamber 32 causes the introduction of enough cooling liquid upstream of the bulb 18 to effect throttling of

the expansion device 14 by an amount which will reduce suction line pressure; and the reduction in suction line pressure thus effected is substantially equal to the pressure increase otherwise existing at the evaporator outlet.

Likewise the net area (*a*) that responds to liquid pressure within the chamber 35 must produce such an opening of the orifice 63 as will reduce suction pressure within the pipe 17 to compensate for the inward condenser pressure and maintain substantially constant compressor power consumption.

It is evident that, as the pressure within one of the chambers 32 or 35 causes opening of the orifice 63, the ultimate reduction in pressure within the suction line 17 will cause a reduction in pressure within the chamber 32. This latter pressure reduction causes throttling of the pilot valve as suction pressure reduces. In this manner, the pilot valve and expansion device compensate one another to maintain combinations of condenser and evaporator pressures that limit compressor power consumption.

This limiting of compressor power consumption as a function of condenser and evaporator outlet pressures can be maintained for any compressor operating characteristic. The compressor load curves of FIGURE 5 are examples of these characteristics. Variations for different compressors are accomplished by varying the ratio of bellows area to diaphragm area, or by changing the size of the orifice 63. In general, the ratio of diaphragm area to bellows area will range in the neghborhood of 4:1 with an orifice diameter of .039 inch. Variations of these dimensions for compressor load are easily made with the most convenient change being in the horizontal size of the bellows.

Various changes and modifications may be made within the process of this invention as will be readily apparent to those skilled in the art. Such changes and modifications are within the scope and teaching of this invention as defined by the claims appended hereto.

What is claimed is:

1. In a refrigeration system of the type having a compressor, a condenser, an expansion device and an evaporator piped in series, with a sensing bulb connected to the expansion device and positioned downstream of the evaporator, a pilot valve comprising a valve housing having an inlet port connected upstream of the evaporator and an outlet port connected downstream of the evaporator, a partition within the housing separating its interior into two chambers, an orifice through the partition through which refrigerant fluid can flow, the partition defining a planar face immediately adjacent the edge of the orifice, a member movable relative to the orifice for alternately throttling and opening the orifice, the member having a planar face on it opposite the planar face of the partition, and means for automatically maintaining the face of the member substantially parallel to the face of the partition, having a body portion defining a portion of the sphere, and means defining a seat within which the body portion protrudes with a spring for adjusting the position of the body portion within the seat to maintain the aforesaid parallel relationship.

2. The valve of claim 1 with a filter between the inlet port and the opening through the partition.

3. In a refrigeration system of the type having a compressor, a condenser, an expansion device, and an evaporator piped in series, with a sensing bulb connected to the expansion device and positioned downstream of the evaporator, a pilot valve comprising a valve housing having an inlet port for connection to the high pressure side of the refrigeration system and an outlet port for connection to the low pressure side of the refrigeration system, a partition within the housing separating its interior into two chambers, the first including the outlet port and the second including the inlet port, and orifice through the partition, a movable member for throttling and opening the orifice, the first chamber having first movable wall means and means establishing a connection to cause the movable member to follow the first movable wall means, the second chamber having second movable wall means and means establishing a connection to cause the movable member to follow the second movable wall means, the positions of the first and second movable wall means and the movable member being such as to cause opening of the orifice with an increase in pressure within either chamber, the net effective area of both movable wall means being variable by changing the size of one of the movable wall means.

4. The pilot valve of claim 3 wherein one of the movable wall means includes a bellows.

5. The pilot valve of claim 3 with means for biasing the movable member in an orifice closing direction and means for adjusting the strength of the biasing means.

References Cited by the Examiner

UNITED STATES PATENTS

399,548	3/89	Nageldinger	137—508
1,238,051	8/17	Peterson	62—47
2,472,049	5/49	Schneck	137—508
2,631,612	3/53	Buescher	251—360
2,839,078	6/58	Lornitzo	137—508
2,972,236	2/61	Nussbaum	62—176
3,014,352	12/61	Leimbach	62—197
3,099,140	7/63	Leimbach	62—197

ROBERT A. O'LEARY, *Primary Examiner.*

MEYER PERLIN, *Examiner.*