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COMPOUND PRESSURE REGULATING SYSTEM FOR REFRIGERATION

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Fig.1.

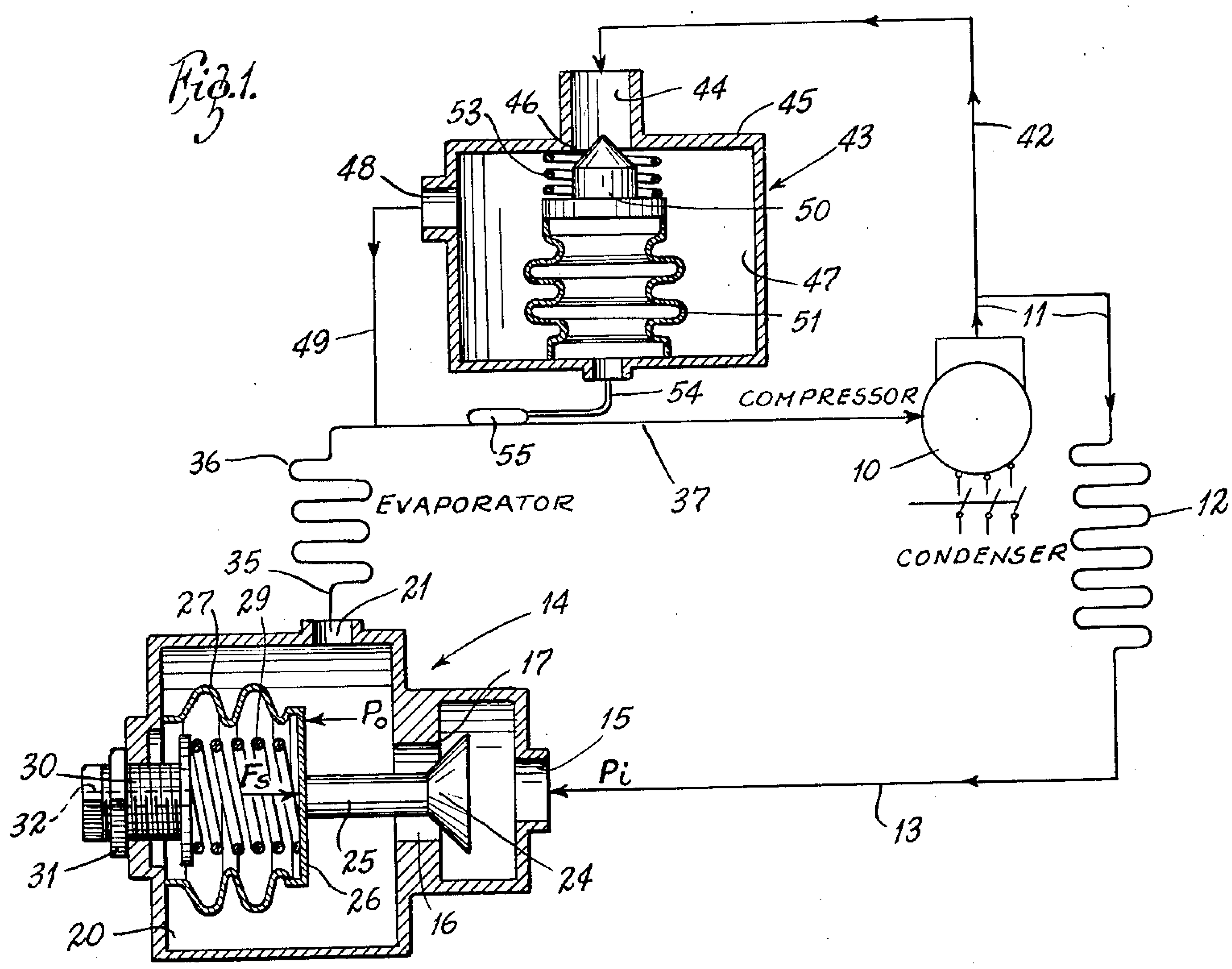
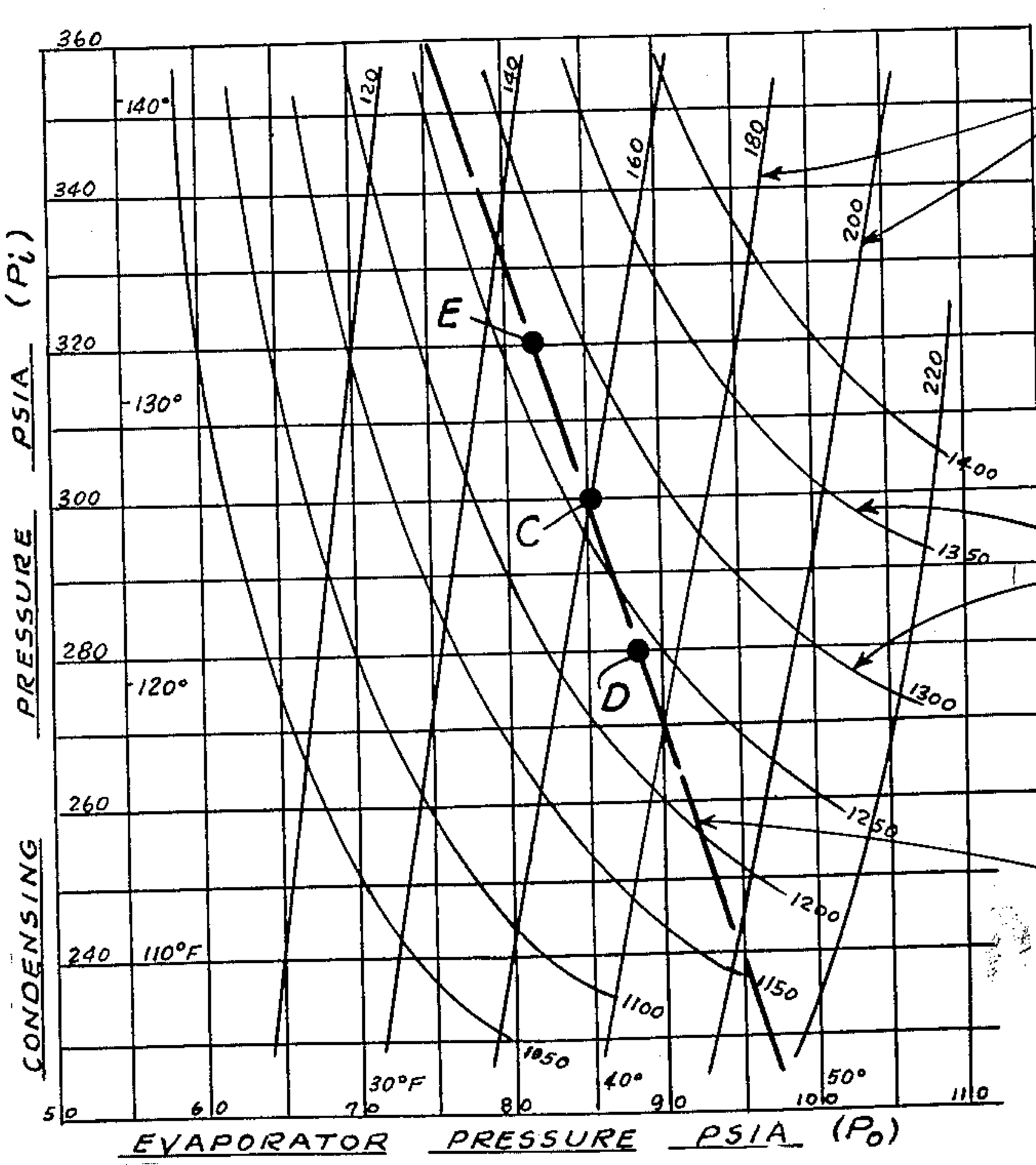


Fig.2.



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COMPOUND PRESSURE REGULATING SYSTEM FOR REFRIGERATION

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The present invention relates to a novel refrigeration system and, particularly, to a control for such a system. It is particularly applicable to the type of system having a compressor, a condenser, a refrigerant liquid expansion device, and an evaporator all connected in a closed circuit.

In the illustrated embodiment of the invention, believed to be the preferred but not the only embodiment, the refrigeration system is one of the type aforesaid, but wherein a refrigerant liquid expansion valve is positioned by a combination of condenser as well as evaporator pressures so that it can not only open further with decrease in evaporator pressures and vice versa, but also will tend to reduce expansion valve outlet or evaporator pressures with increase in condenser pressures, and vice versa. Hence the operating pressures provided for the system can be made to approach those wherein the power consumption of the compressor is constant, or below a predetermined maximum. Further, there is a constant superheat control included in a compressor bypass, to prevent flooding of the compressor, as well as to equalize the evaporator and condenser pressure rapidly despite existence of a closed condition of the expansion valve.

Heretofore the conventional automatic control of such refrigeration system has involved the use of either a constant pressure regulator valve sensitive to evaporator pressures, or a thermostatic expansion valve usually of the constant superheat type, sensitive to a combination of temperature at the evaporator outlet and evaporator pressure. In such systems the valve is designed to react as little as possible to the effects of the inlet, or condenser, pressures. As a result the power consumption of the compressor varies widely with variations in condenser pressures.

The prime objective of the present invention is to provide a refrigerating system in which the power consumed by the compressor is substantially constant, or varies below a predetermined maximum. To attain this principal object, the invention consists of controlling the flow of refrigerant from the condenser to the evaporator as a function of the differences between pressures of the condenser and of the evaporator or, as it may be, the difference between pressures at the inlet and those at the outlet of the expansion valve.

It is a specific object of the invention to provide a system wherein an expansion valve, or control valve, between the condenser and evaporator of a refrigeration system, produces reduction in expansion valve outlet or evaporator pressures as expansion valve inlet or condensing pressures rise. It is especially an object to provide a system wherein inlet pressures vary inversely with expansion valve outlet pressures in predetermined ratio. A particular object is to make that ratio substantially parallel the constant power curve of the compressor.

A further object of the invention is to provide an expansion valve that is made responsive to both its inlet and outlet pressures. It is a specific object to provide such a valve differing from the conventional expansion valve in that the valve itself is made sufficiently large as to be significantly affected by its inlet pressure.

It is an especial object of this invention to provide a control valve in a system of the kind described in which the valve is positioned by the combination of valve clos-

ing forces that are functions of the valve's inlet and outlet pressures and valve opening forces in the form of a resilient means, all in such wise that the valve can respond to pressures that vary directly with increase in the net effective pressure area responsive to outlet pressure and directly as the net effective area subjected to inlet pressure.

Another object of the invention is to provide a refrigeration control of the foregoing type that has means to prevent flooding of the compressor with liquid refrigerant, as well as a system wherein equalizing occurs rapidly after an "off" cycle starts, and wherein the operation is stable without rapid changes in evaporator pressure.

Other objects will appear from the description to follow wherein:

FIGURE 1 is a schematic view of a system embodying a system of the kind described with the valve shown in medial section; and

FIGURE 2 is a graph of the operation of the present system.

The system illustrated includes a compressor **10** having a principal outlet pipe **11** connecting into a condenser **12**. From the condenser a pipe **13** leads, either with or without the interposition of one of the conventional receivers, into the valve **14** of the present invention.

The valve **14** includes a housing having an inlet **15** leading to a valve port **16** having a valve seat **17**. The valve port **16** opens into a pressure chamber **20** having an outlet **21**.

Control of the flow of refrigerant through valve **14** is provided by a valve **24** that cooperates with the valve seat **17**. The valve **24** is mounted on a valve stem **25** that in turn is attached to the head **26** of a bellows or equivalent pressure-responsive movable wall device **27**. This bellows is here illustrated as attached to the housing **14**.

A spring **29** acts within the bellows to urge the valve **24** toward open position. A threaded adjustment screw **30**, with a lock nut **31**, represents a typical adjustment for the force of the spring **29**. A port **32** admits atmosphere to the interior of the bellows. A sealed, pressurized bellows, with or without a spring, may be used, if desired, in place of the spring arrangement.

The valve **24** is of substantial area as will be described; and the effective area of the bellows **27** is considerably larger than that of the valve, as will be described. The face area of the valve **24** is designed to be acted upon and to be responsive to inlet pressures, as will appear. It constitutes a pressure-responsive actuating means for the valve. This arrangement is preferable to a second bellows or the like because of its greater simplicity.

The outlet **21** of the housing is connected by a pipe **35** into the evaporator **36**. From the evaporator a suction line **37** leads back to the compressor **10**.

The system includes also a superheat valve interposed in a bypass around the compressor. A bypass line **42** leads from the compressor outlet **11** into the superheat valve **43**, by way of inlet **44** opening into the superheat valve housing **45**. The housing **45** has a valve seat **46** opening into a pressure chamber **47** that is connected to compressor suction via valve outlet **48**, a line **49**, and a portion of the suction line **37**. The flow of refrigerant through the valve **43** is regulated by a valve **50** mounted upon a bellows **51**, conventionally charged with the same fluid as is used for the refrigerant in the system. The pressure in the bellows urges the valve **50** toward the valve seat **46**, against action of a spring **53** that urges the valve toward an open position.

The interior of the bellows **51** is connected by a capillary tube **54** or the like to a bulb **55**, located on the suction line 37 between the evaporator outlet and the compressor inlet to sense temperature of the refrigerant at the compressor inlet. The charging fluid in the bellows system produces pressure increases with increase in temperature at the evaporator outlet.

In FIGURE 2, certain characteristics of a typical compressor are plotted with respect to the use of a given refrigerant. The ordinates are condensing pressures in pounds per square inch absolute. The abscissae are evaporator pressures in pounds per square inch absolute for a typical compressor, for which there are corresponding saturation temperatures. Plotted on the graph are lines of constant refrigerant flow rates in pounds per hour, from 120 to 200 pounds per hour, and plotted also on this graph are lines of constant compressor input power in watts. These are illustrated with curves from 1050 to 1400 watts. These curves show that constant compressor input power uniformly requires that, as the condensing pressure increases, evaporator pressure must decrease, or conversely, as condensing pressure decreases the evaporator pressure increases.

Operation

In the primary system, when the main switch to operate the compressor is closed, the compressor will start. Thereafter its operation is conventionally controlled by a space thermostat. The compressor delivers refrigerant such as Freon 12 to the line **11** and into the condenser **12**, where it is condensed. This liquid may then flow from the condenser **12** through the pipe **13** to the control valve **14**. Assuming the valve element **24** to be off of the seat **17**, the refrigerant will flow across the seat **17**, being expanded in the process, and will then flow through the outlet **21** of the valve to the pipe **35** and enter the evaporator **36**. In the evaporator the liquid vaporizes and cools the ambient fluid, such as air. The vapor then flows through the pipe **37** back into the compressor.

The valve **24** is designed to regulate the amount of flow into the evaporator. The valve is acted upon by several forces. In the first place, there is the opening force of the spring **29**, or its equivalent, which is of predetermined and presettable value. There is the closing force of the vapor pressure within the chamber **20** which corresponds to evaporator pressure **36**, acting upon the head **26** of the bellows **27**. Since this pressure acts also in an opening direction against the left hand side of the valve head **24**, it can be seen that the net valve closing force exerted by outlet pressure is the vapor pressure in the evaporator, multiplied by the area of an annular portion of the head **26**, the inner diameter of which equals the diameter of the valve seat **17**, and the outer diameter of which is the effective diameter of the bellows. The valve is also acted on in a closing direction by a force equal to inlet or condenser pressure times the area of the valve seat **17**, this force comprising the inlet pressure acting upon the pressure-responsive right side or face of the valve **24**, from which the balanced area is subtracted. The valve seat is made of sufficient size to make this force of sensible value as will hereafter appear.

The following formula therefore represents equilibrium conditions on this valve:

$$F_s = P_o A + P_i a$$

Where F_s is the force of the spring **29**, P_o is outlet pressure or evaporator pressure, the one being a function of the other, A is the excess of the effective area of the bellows head **26** over the effective area of the valve **24**, and a is the effective area of the valve **24** acted upon by the inlet or evaporator pressure P_i. The ratio of these areas may be represented as follows:

$$R = \frac{A}{a}$$

Solving the foregoing for outlet or evaporator pressure, produces the formula:

$$P_0 = \frac{1}{R}\left(\frac{F_s}{a} - P_i\right)$$

From the foregoing formulation, it can be seen that outlet pressure is reduced as inlet pressure is increased; and that the rate of reduction of outlet pressure with increase of inlet pressure is determined by the ratio of the areas A and a. An increase of a with respect to the size of A increases the slope of the curve in the sense of increasing its abscissae with respect to its ordinates.

A characteristic curve is shown by the dashed line in FIGURE 2 to illustrate the variation of evaporator pressures with changes of condenser pressures, when the valve **14** is used in a typical installation. Considering a point such as the point C, where the dashed line crosses 300 pounds per square inch condensing pressure, the corresponding evaporator pressure will be 85 p.s.i. The point C may be taken as the starting point for a description of the operation.

The particular usefulness of the present invention can be demonstrated by considering that the machine is in operation under constant load with the valve **24** in a predetermined relationship with respect to the valve seat **17**. The outside temperature to which the condenser **12** is subjected is assumed to be at some predetermined value.

Under such circumstances, if there be a drop in outside temperature, there will be a corresponding drop in P_i or condenser pressure acting upon the face of the valve **24**. A reduction of P_i is followed by a movement of the valve **24** to a more open position because of a reduced opposition to the force of the spring F_s. Opening of the valve **24** tends to increase the rate of flow through the valve and to increase the outlet pressure P_o. Such a change could be represented by a shift in the operating characteristics from the point C to the point D. It is to be noted that a shift from the point C to the point D is not accompanied by a rise in power consumption. To the contrary, there is a small reduction in power consumption which, for this particular compressor, is from about 1260 to 1240 watts.

By the same token, if the outside temperature around the condenser **12** rises, P_i will increase and force the valve **24** toward a more closed position. This will tend to reduce the rate of evaporator flow through the valve and reduce the outlet pressure P_o and the evaporator pressure. This could be represented by a shift of the operating conditions from the point C to the point E. A shift from the point C to the point E does not change the power requirements to any substantial degree, since, in this area the value P_i—P_o curve is substantially parallel to the constant power drive.

If it be realized that the normal P_i—P_o curves for conventional refrigerant controls slope in the other direction, it can also be observed that if the outside temperature should rise, as it might when there is an increase in load, such rise would be accompanied by a very substantial rise in power requirement. Therefore this present control automatically operates to maintain throughout the major part of the operation range a substantially constant power consumption and throughout all of the operating range provides an operation that is below the desired maximum power consumption.

If the outside temperature around the condenser falls by a large amount, P_i will be reduced by a corresponding amount and the valve **24** may open to a rather wide position allowing a larger rate of refrigerant flow. With such larger rate of flow, particularly if it occurs at a period of reduced evaporator load, there could be a "flood-back" to the compressor that is made somewhat more acute by the presence of the control **14**, of this invention. However, where the superheat valve **43** is employed, the reduction or loss of superheat of the refrigerant in suction line **37** caused by the rise in pressure in the evaporator or by decrease in the evaporator load, will cause a collapse of the bellows 51 and a movement of the valve 50 in an opening direction under influence of the spring 53. This valve will therefore introduce a certain amount of hot refrigerant vapor from the compressor discharge into the suction line 37, and thereby tend to increase the pressure in the suction line 37, in the evaporator 36, and in the outlet P_o, of valve 14. However, any increase in the outlet pressure P_i of the expansion valve 14, will exert a force on the bellows 26 tending to move the valve 24 toward the closed position, and thus to reduce the flow of liquid refrigerant through the valve to evaporator 36. This inter-related action of valves 43 and 14 will automatically proportion the rates of flow of hot refrigerant vapor through valve 43 and liquid refrigerant through valve 14, 50. A preset value of superheat is maintained in the suction line 37, and the danger of "flood-back" to the compressor is eliminated.

Another gain from the use of the superheating valve 43 is that it permits a rapid equalization of pressures on the opposite sides of the compressor after an off-cycle is started. If an off-cycle is started with a high evaporator pressure or a very high condenser pressure, or both, the valve 24 may be closed tightly and its opening would be considerably delayed because, as soon as the system stops, the refrigerant in the evaporator tends to evaporate and create an even higher vapor pressure condition. Hence, if a new off-cycle starts when the valve 24 is thus closed, the compressor may be starting under very undesirable conditions. Where the valve 43 is present, the superheat in the suction line 37 will tend to drop to zero and the valve 50 will open and permit rapid equalization of pressure in the suction and compression lines of the system.

The foregoing description shows that the present invention accomplishes the objectives set forth. The elements of the system may be varied in detail and still embody certain aspects of the invention. Therefore, it is intended that such variations as may be reasonably encompassed within the scope of the following claims, shall be subjected to the protection of this grant.

What is claimed is:

1. The combination of a compressor, a condenser, an expansion device and an evaporator piped in series wherein the expansion device includes means for responding to changes in condenser pressure to produce corresponding changes in evaporator pressure, and means for maintaining the ratio between condenser pressure and evaporator pressure such that compressor power consumption is substantially constant with variations from constant being in the direction of slightly reduced compressor power consumption with increase in evaporator pressure wherein the evaporator pressure varies inversely with condenser pressure, and the substantially constant power consumption is generally above 270 p.s.i.a. of condenser pressure.

2. In a control for a refrigeration system having a compressor, a condenser, an expansion device, and an evaporator piped in series; the combination wherein there is means to cause changes in evaporator pressures to accompany and vary inversely with changes in condenser pressures along substantially constant levels of compressor power consumption.

3. The combination of claim 1, and means responsive to conditions producing flooding of the compressor to reduce the refrigeration capacity of the system to prevent such flooding.

4. The combination of claim 1, and a bypass for the compressor, with a valve regulating the opening of the bypass, and means responsive to temperatures corresponding to evaporator outlet temperatures, to operate the valve to move the same in an opening direction in response to existence of conditions in the suction side of the system that cause compressor flooding.

5. A method of regulating a refrigeration system of the kind having a compressor, a condenser, and an evaporator connected in series, comprising: varying evaporator pressures inversely as condenser pressures vary in such wise as to cause the variations substantially to parallel the constant wattage consumption of the compressor above a predetermined value of condenser pressure.

6. A method of regulating a refrigeration system of the kind having a compressor, a condenser, and an evaporator connected in series, comprising: producing a force that varies with condenser pressure, producing a force that varies with evaporator pressure, applying both forces to a valve in the evaporator inlet to cause the valve to maintain constant evaporator pressure except that when condenser pressure lowers, the valve is operated in a direction to raise evaporator pressure, and when condenser pressure rises, the valve is operated to lower evaporator pressure, and maintaining substantially constant compressor power consumption for a predetermined range of condenser and evaporator pressures.

7. The method of claim 6, and the step of bypassing refrigerant around the compressor in response to lowering of suction line temperature, and thereby preventing flooding of the compressor.

8. In a control for a refrigeration system having a compressor, a condenser, an expansion device, and an evaporator piped in series, the combination wherein the expansion device comprises a valve; means responsive to increase and decrease in refrigerant pressures at the evaporator inlet to urge the valve in a throttling direction and opening direction, respectively; and means responsive to increase and decrease in pressures at the condenser outlet to urge the valve in throttling and opening directions, respectively; the means being proportioned such that increase in condenser pressures is accompanied by such reduction in evaporator pressures that the compressor power requirement remains substantially constant.

9. The control of claim 8, wherein the valve has a head of substantial size and a valve port, the valve port being connected to the condenser outlet so as to subject the valve head to condenser pressure changes; the means responsive to evaporator pressures comprising a pressure-responsive movable device on the downstream side of the valve, and connected to the valve, and the valve head having such substantial size relative to the movable device that the evaporator and condenser pressures vary oppositely as aforesaid.

10. A refrigeration system having a compressor, a condenser, an expansion device and an evaporator piped in series, the expansion device comprising a valve having an inlet port connected to the pipe leading from the condenser outlet and an outlet connected to the pipe leading to the evaporator inlet, means defining an orifice between the inlet and outlet ports, a valve member movable between orifice throttling and orifice opening positions, the valve member having means defining a surface exposed to refrigerant pressure at the inlet port and means defining a surface exposed to pressure at the outlet port, each of the surfaces being so disposed as to respond to an increase in the pressure to which it is exposed to move the valve member in an orifice throttling direction, and vice versa, the area of the surface exposed to inlet port pressure being at least about one-tenth as large as the area of the other surface.

References Cited in the file of this patent

UNITED STATES PATENTS

2,191,102 Zerk ------------------ Feb. 20, 1940

OTHER REFERENCES

Article in "Air Conditioning and Refrigeration News," February 4, 1946, entitled "Refrigeration Problems and Their Solution," by P. B. Reed, page 13, one sheet of drawing and one page of printed matter.