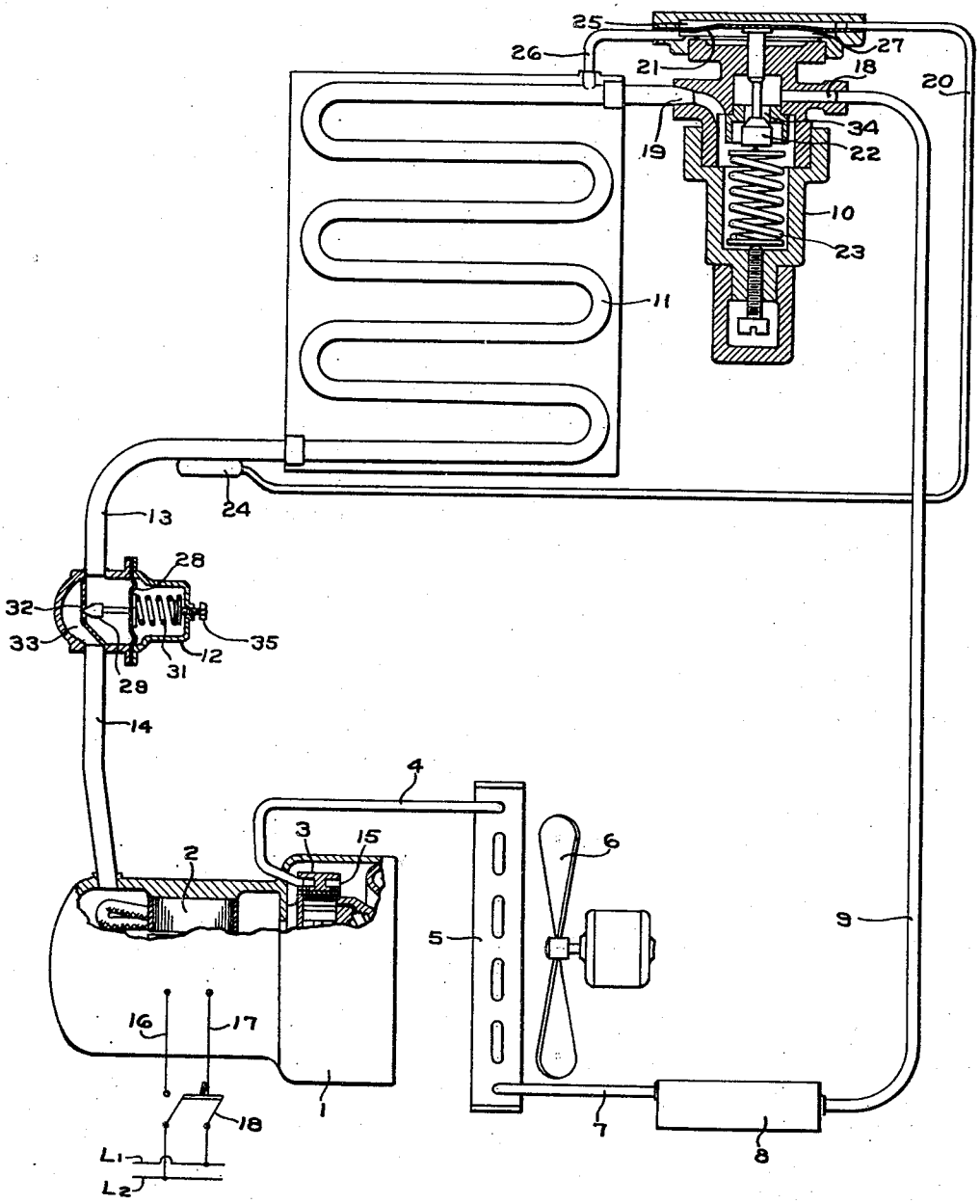
April 25, 1939. R. H. TULL ET AL 2,155,516

REFRIGERATION APPARATUS

Filed Sept. 3, 1936



WITNESSES:

W. M. Crane Seiver

E. Lutz.

INVENTOR

ROBERT H. TULL, AND

BARTEL J. HOMKES.

BY Wattenga

ATTORNEY

Patented Apr. 25, 1939

2,155,516

UNITED STATES PATENT OFFICE

2,155,516

REFRIGERATION APPARATUS

Robert H. Tull, Springfield, Mass., and Bartel J. Homkes, Detroit, Mich., assignors to Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa., a corporation of Pennsylvania

Application September 3, 1936, Serial No. 99,180

5 Claims. (Cl. 62—3)

Our invention relates to refrigeration apparatus and particularly to a temperature and pressure control for the evaporator of a refrigerating system.

In refrigeration apparatus for certain applications, for example, package sealing machines, wool shearing machines, and the like, it is necessary to maintain a substantially constant temperature in an evaporator which has very little thermal storage capacity, and which is subject to sudden and appreciable changes in heat load. It is also necessary in many applications to prevent excessive moisture from collecting on the suction line from the evaporator which might drip and spoil the material which is being operated upon. Heretofore, the usual method of maintaining a substantially constant temperature was to cycle the apparatus in response to heat load; in other words, the motor driving the refrigerator compressor was energized when the evaporator required refrigeration and was shut off when the evaporator no longer required refrigeration.

However, such an arrangement required cycling controls, caused the machine to cycle too much as the load varied, and at intervals reduced the suction pressure of the evaporator to such a low value that moisture condensed on the suction line and dripped on the material operated upon. Furthermore, the success of this type of temperature control was dependent on the evaporator having sufficient thermal capacity to store up refrigeration to take care of the heat load for an appreciable period of at least several minutes. The only way in which refrigeration can be obtained from this type of evaporator, during the off cycle, is to permit a rise in temperature, which in certain applications cannot be permitted.

It is an object of our invention, therefore, to operate a refrigerating machine in such a manner that, if the amount of heat absorbed by the evaporator is varied or the capacity of the compressor is changed slightly due to variations in condensing pressure, the temperature of the evaporator will be maintained substantially constant, and furthermore, to accomplish this result without the use of controls which stop and start the motor in response to evaporator temperature.

It is another object of our invention to maintain a constant pressure in the evaporator and in the suction conduit extending from the evaporator so that the temperature of the evaporator and a portion of the suction line remain substantially constant.

These and other objects are effected by our invention, as will be apparent from the following description and claims taken in connection with the accompanying drawing, forming a part of this application, in which:

The single figure of the drawing illustrates diagrammatically a refrigerating system of the compression type embodying our invention.

Referring specifically to the drawing for a detailed description of our invention, numeral 1 designates a hermetically sealed casing which encloses a motor 2 driving a compressor 3. Compressed refrigerant vapor is forced from the outlet of the compressor through a conduit 4 to a condenser 5, wherein the compressed refrigerant is liquefied due to the cooling action of air circulated over the condenser by a fan 6. Condensed refrigerant is conveyed through a conduit 7 to a liquid receiver 8, the liquid receiver 8 and condenser 5 having sufficient capacity to contain the entire refrigerant charge in the system.

Liquid refrigerant is conveyed from the receiver 8 to an evaporator 11 through a conduit 9. The conduit 9 has a thermally responsive expansion valve 10 interposed therein adjacent to the evaporator 11. Refrigerant is vaporized in the evaporator 11, and is conveyed, first through a constant back-pressure valve, generally indicated at 12 through a conduit 13, and from the back-pressure valve 12, to the interior of the sealed casing 1 through a conduit 14. Refrigerant vapor is withdrawn from the interior of the sealed casing 1 through an inlet 15 of the compressor 3, and the above-described cycle is then repeated. The motor 2 is energized from a line L_1, L_2 through conductors 16 and 17. A manually operated switch 18 is provided for energizing the motor 2 when refrigeration is desired.

The thermo-expansion valve 10 is provided with an inlet 18 and an outlet 19, which outlet connects with the evaporator 11. The expansion valve 10 is also provided with a diaphragm 21, a plunger-actuated valve 22 and an adjustable spring 23 which opposes the action of the diaphragm 21. A bulb 24, filled with a suitable gas, is connected by a conduit 20 to a chamber 25 on the upper side of the diaphragm 21. Another conduit 26 connects with a chamber 27 on the lower side of the diaphragm 21 and also connects with the interior of the evaporator 11.

The thermo-expansion valve 10 is, therefore, responsive to both the pressure of the evaporator 11 and the pressure of the gas in the bulb 24 and conduit 20. Before the compressor 3 is started by closing the switch 18, the pressure in the evapo-

rator 11 is higher than the normal operating pressure, due to the raising of the temperature of the refrigerant in the evaporator while the compressor has not been in operation. Under this condition of abnormal evaporator pressure, the expansion valve is closed because the high pressure refrigerant in the evaporator communicates with the underside of the diaphragm 21 through the conduit 26 and tends to raise the valve 22.

In accordance with our invention, refrigerant liquid is supplied to the evaporator 11 in varying quantities, depending on the heat load, as long as the motor and compressor are in operation, but the suction-pressure and consequently the temperature of the evaporator and the portion of the suction line between the evaporator and the back pressure valve are maintained substantially constant. In order to effect balance of heat load and condensing unit capacity without starting and stopping the motor 2 in response to the pressure or temperature of the evaporator, a constant pressure valve 12 is inserted in the conduit between the evaporator and the compressor. The constant pressure valve 12 is provided with a diaphragm 28 and a plunger valve 29. A spring 31 opposes the pressure acting on the diaphragm 28 through the conduit 13. The plunger valve 29 operates to adjust the flow of refrigerant through an orifice 32 which is provided between the conduit 13 and a chamber 33 communicating with the suction conduit 14.

As soon as the compressor 3 is started, there is an immediate reduction in pressure in the evaporator 11, which reduction of pressure is translated to the chamber 27 on the under side of the diaphragm 21 of the expansion valve 10, thus tending to force the valve plunger 22 downwardly and permit refrigerant to flow from the conduit 18 through an orifice 34 in the expansion valve 10 and out through conduit 19 to the evaporator 11, which refrigerant boils at a temperature corresponding to the pressure in the evaporator. The gas in the bulb 24 being warm due to the inactivity of the compressor, exerts pressure through the conduit 20 on the upper surface of the diaphragm 21 and also tends to hold the expansion valve open and flood the evaporator 11 with refrigerant. More refrigerant is thus admitted to the evaporator than can be vaporized and some of the refrigerant reaches the suction conduit 13 producing a cooling effect on the gas in the bulb 24, which is thermally attached thereto. The cooling of the bulb 24 produces a reduction in pressure of the gas, which is communicated to the upper side 25 of the diaphragm 21 through the conduit 20, the spring 23 thereafter tending to raise the plunger valve 22 and throttle the flow of refrigerant passing through the orifice 34, so that less refrigerant is admitted to the evaporator 11. As this throttling effect continues, the compressor, which is operating at constant speed and is assumed to have sufficient capacity to cool the evaporator, is enabled to reduce the pressure in the evaporator 11 until the desired operating pressure is reached, as determined by the setting of the back pressure valve 12. The constant back pressure valve 12 is provided with an adjusting screw 35 which changes the bias of the spring 31, and the spring 31 is so adjusted that the back pressure valve begins to throttle the flow of refrigerant through the orifice 32 when the desired suction pressure has been reached in the evaporator 11. As the unit continues to operate the compressor pulls down to a lower pressure and temperature in the conduit 14 and in the compressor casing 1, but the constant back pressure valve 12 permits only sufficient refrigerant to be withdrawn from the evaporator 11 to maintain the suction pressure and temperature of the evaporator and suction conduit 13 substantially constant.

So long as the evaporator load remains constant and the unit capacity is unchanged, the system will operate at a condition in which the desired pressure will be maintained in the evaporator 11 and will operate at a suction pressure in the conduit 14 and casing 1 of the compressor such that the unit capacity will be substantially equal to the evaporator load. Slight variations in the unit capacity due to differences in condensing temperature will only effect the pressure in the conduit 14 in the casing 1 of the compressor without effecting the temperature and pressure of the evaporator 11.

Assume now that the load on the evaporator 11 is increased; refrigerant which is being admitted to the expansion valve 10 will be evaporated before it reaches the bulb 24 on the suction conduit 13 and the temperature of the gas in the bulb will thereafter rise. The increase in the bulb temperature increases the temperature of the gas therein which increase in pressure is communicated to the chamber 25 in the upper side of the diaphragm 21, thus tending to force the plunger 22 downwardly and open the orifice 34 wider so that more refrigerant will pass through the expansion valve into the evaporator 11, which additional refrigerant absorbs the increased heat load. The addition of more refrigerant and a greater heat load would normally result in a higher operating pressure in the evaporator. However, the increase in the evaporating pressure of the refrigerant in the evaporator 11 acts upon the diaphragm 28 of the constant pressure valve 12 to open the orifice 32 and permit the flow of more refrigerant to the compressor 3. Since the constant pressure valve 12 is a throttling device it will adjust itself to give the proper opening and to pass whatever amount of refrigerant is admitted by the expansion valve 10. Thus, the increase in the evaporator load is accompanied by an increase in the amount of refrigerant admitted by the expansion valve 10 but the rise in the pressure which would normally accompany this increase in refrigerant and heat load is offset by further opening of the back pressure valve 12. The increased evaporator load effects an increase in the pressure in the suction conduit 14, and the interior of the casing 1 to such a point that the unit capacity at the new pressure in the compressor will give the compressor a capacity substantially equal to the evaporator load.

Assuming now that the load on the evaporator is decreased, some of the refrigerant in the evaporator 11 will not be vaporized and will flow into the suction conduit 13 causing a reduction in the pressure of the gas in the bulb 24. This reduction in pressure is communicated to the chamber 25 above the diaphragm 21 of the expansion valve 10 so that the spring 23 will force the plunger 22 up to close the orifice 34, thus throttling the flow of refrigerant through the expansion valve to the evaporator 11. A smaller amount of refrigerant is, therefore, admitted to the evaporator and the normal tendency would be for the compressor 3 to lower the pressure of the refrigerant in the evaporator 11. However, any decrease in the pressure of the refrigerant in the evaporator 11 acts upon the diaphragm 28 of the constant pres-

sure valve 12 and the plunger 29 is therefore moved in a direction tending to close the orifice 32, thus decreasing the amount of refrigerant drawn from the evaporator 11 by the compressor 3. Again, since the back pressure valve 12 is a throttling valve, it will adjust itself to the new load and maintain substantially the desired constant pressure in the evaporator 11. Under this new condition, the pressure in the conduit 14 and the compressor casing 1 will be lowered to such a point that the capacity of the compressor at the lower suction pressure will equal the evaporator load.

When the compressor 3 is stopped by operating the manual switch 18 to deenergize the motor 2, the pressure of the refrigerant in the evaporator 11 gradually builds up. The increase in pressure during the inactive period tends to open the constant back pressure valve 12, and the increased pressure is also communicated to the chamber 27 on the underside of the diaphragm 21 of the expansion valve 10, eventually closing the valve 10, and isolating the evaporator 11 from the high side of the system. It will be seen, therefore, that, if the motor and compressor are again started after a short period of time, it will not be necessary to perform as much work to pull the evaporator 11 down to the desired suction pressure, as if the pressures in the system had completely equalized.

It will be noted that the thermo-expansion valve 10 is directly effected by the pressure of the refrigerant in the evaporator 11 only during the time that the evaporator is inactive, in order first to effect a rapid flow of refrigerant to the evaporator as soon as the compressor is started, which in turn reduces the temperature of the evaporator to the desired value very quickly, and secondly to prevent rapid pressure equalization after the compressor is stopped. After the compressor has been started and is in normal operation, the control of the amount of refrigerant admitted to the evaporator is dependent upon the temperature of the refrigerant vapor leaving the evaporator as translated to the expansion valve through the medium of the gas in the bulb 24 and the conduit 20.

It will be seen that the capacity of the evaporator may be controlled by adjusting the spring 31 of the constant back pressure valve 12. In this system the evaporator capacity will be increased by lowering the evaporator pressure because the lowering of the pressure will decrease the temperature of refrigerant in the evaporator and increase the temperature differential between the refrigerant and the load, permitting more heat to flow into the refrigerant. Obviously, if the constant back pressure valve is adjusted to raise the evaporator pressure, the capacity of the evaporator will, in like manner, be decreased.

From the foregoing, it will be apparent that we have devised a system for maintaining the suction pressure and temperature of the refrigerant in an evaporator substantially constant regardless of changes in heat load on the evaporator, and have accomplished this result in a system where the motor operates continuously without being responsive in any way to pressure or temperature of the system. Our invention, therefore, removes the necessity for controls to cycle the apparatus or control the speed of the compressor, and maintains a constant predetermined temperature in an evaporator which has small storage capacity, but is subjected to fluctuating loads. By maintaining the temperature of the evaporator and suction line at a sufficiently high value, excessive condensation of moisture with resulting dripping thereof on the material operated upon is prevented.

While we have shown our invention in but one form, it will be obvious to those skilled in the art that it is not so limited, but is susceptible of various changes and modifications without departing from the spirit thereof, and we desire, therefore, that only such limitations shall be placed thereupon as are imposed by the prior art or as are specifically set forth in the appended claims.

What we claim is:

1. In a refrigerating system wherein a refrigerant is compressed, condensed, and evaporated in a continuous cycle, the combination of an evaporator in which the evaporation of refrigerant takes place, means for effecting continuous operation of the system and, therefore, continuous evaporation of refrigerant in said evaporator, and means for maintaining the temperature of the evaporator substantially constant during continuous operation of the system and regardless of the heat load on said evaporator.

2. In a refrigerating system wherein a refrigerant is compressed, condensed, and evaporated in a continuous cycle, the combination of a compressor for compressing refrigerant vapor, an evaporator in which evaporation of the refrigerant takes place, means for effecting continuous operation of said compressor, flow control means for supplying varying quantities of refrigerant to said evaporator in response to the heat load thereon and valve means for maintaining a constant pressure in said evaporator during said continuous operation of the compressor, regardless of the heat load on said evaporator.

3. In a refrigerating system wherein a refrigerant is compressed, condensed and evaporated in a continuous cycle, the combination of a compressor for compressing refrigerant vapor, an evaporator in which vaporization of the refrigerant is effected, a connection for conveying refrigerant vapor from the evaporator to the compressor, means for supplying varying quantities of refrigerant to said evaporator in response to the load thereon, and a constant pressure valve in the connection between the evaporator and the compressor for maintaining the pressure and the temperature of the refrigerant in said evaporator substantially constant while continuing constant speed operation of the compressor and regardless of the heat load on the evaporator.

4. In a refrigerating system in which a refrigerant is compressed, condensed and evaporated in a continuous cycle, the combination of a compressor for compressing refrigerant, a condenser for liquefying the compressed refrigerant, an evaporator in which liquefied refrigerant is vaporized, connections for conveying refrigerant from said condenser to said evaporator and from said evaporator to said compressor, a valve in the connection between the condenser and the evaporator responsive to the load on the evaporator for supplying varying quantities of refrigerant thereto, and a constant pressure valve in the connection between the evaporator and the compressor for maintaining a substantially constant temperature and pressure of the refrigerant in the evaporator while continuing operation of the compressor and regardless of the heat load on the evaporator.

5. The method of operating a refrigerating system embodying an evaporator and a compressor

which comprises operating the compressor continuously and at constant speed, varying the amount of refrigerant supplied to the evaporator to correspond with varying loads thereon, and maintaining the pressure of the refrigerant in the evaporator substantially constant to maintain a substantially constant evaporator temperature while varying the amount of heat abstracted therefrom in accordance with the load.

ROBERT H. TULL.
BARTEL J. HOMKES.