

Miller

[15] **3,693,370**

[45] **Sept. 26, 1972**

[54] **THERMODYNAMIC CYCLES**

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[22] Filed: **Sept. 25, 1970**

[21] Appl. No.: 75,337

[52] U.S. Cl.62/175, 60/26, 62/332,
62/384

[51] Int. Cl. F25b 25/00

[58] **Field of Search**.....62/87, 116, 117, 118, 175,
62/332, 333, 334, 498, 501, 484; 60/26

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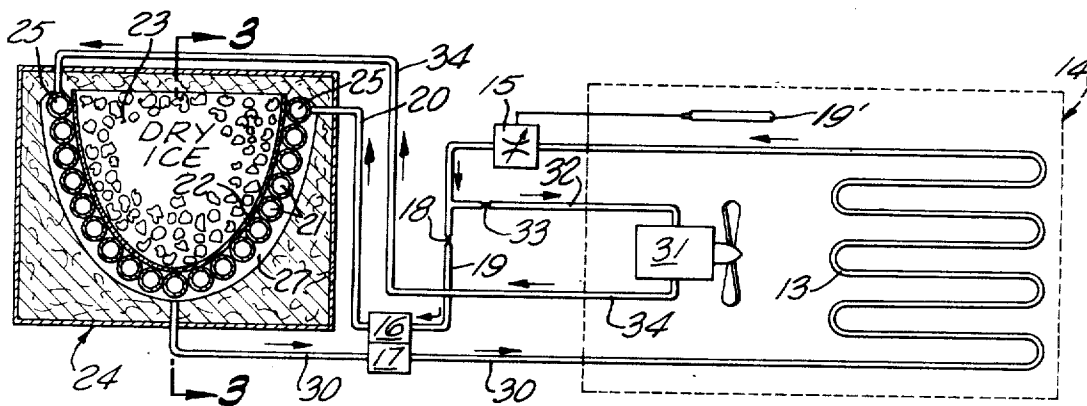
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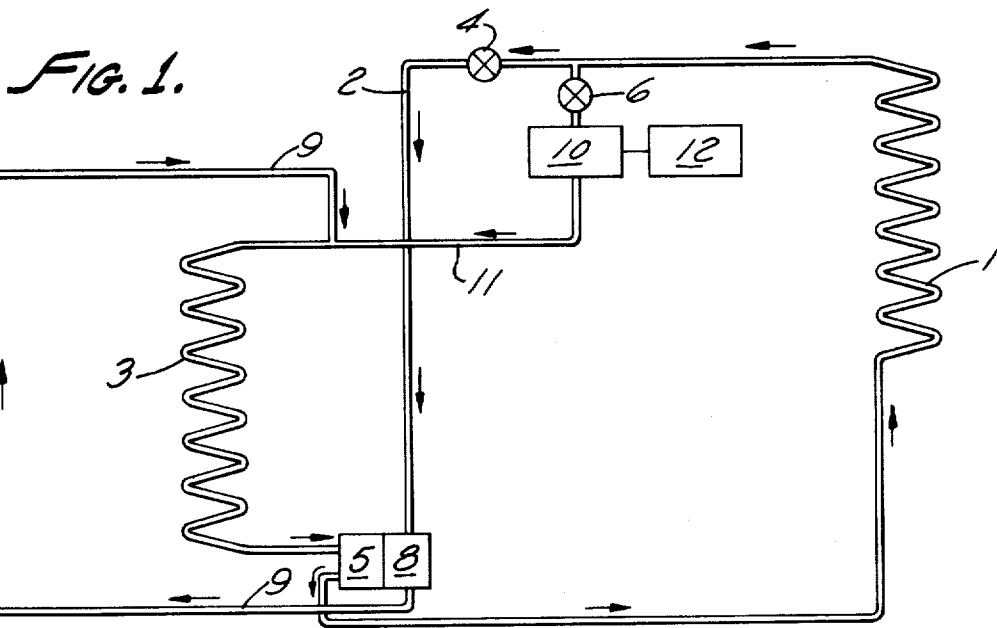
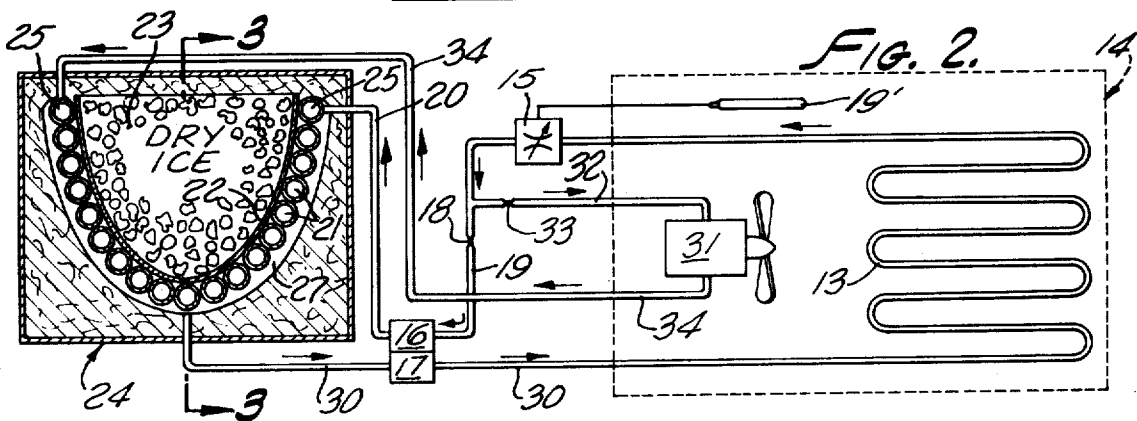
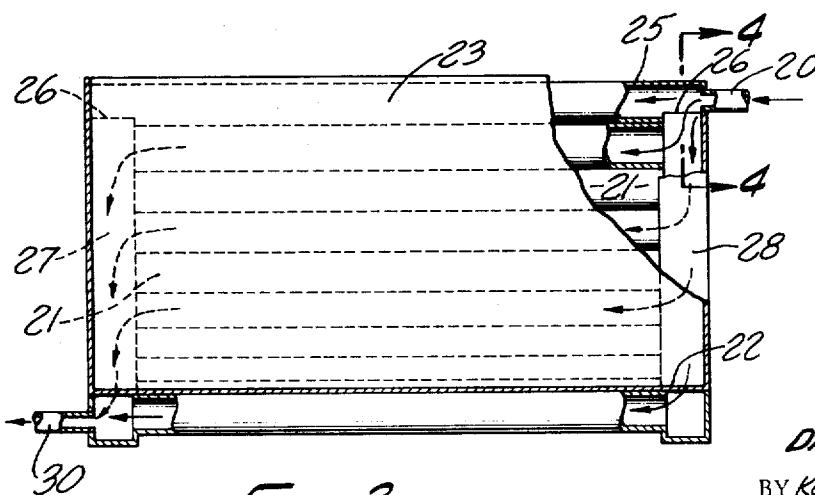
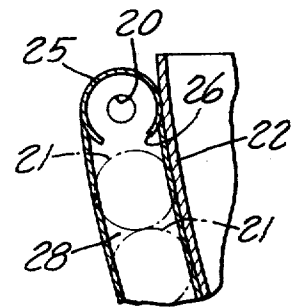
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[57] **ABSTRACT**

This invention relates to thermodynamic cycles operating between two levels of subatmospheric temperature, whereby power is generated and/or refrigeration obtained, in which a heat transfer liquid is employed having a relatively high vapor pressure at atmospheric temperature.

3 Claims, 6 Drawing Figures





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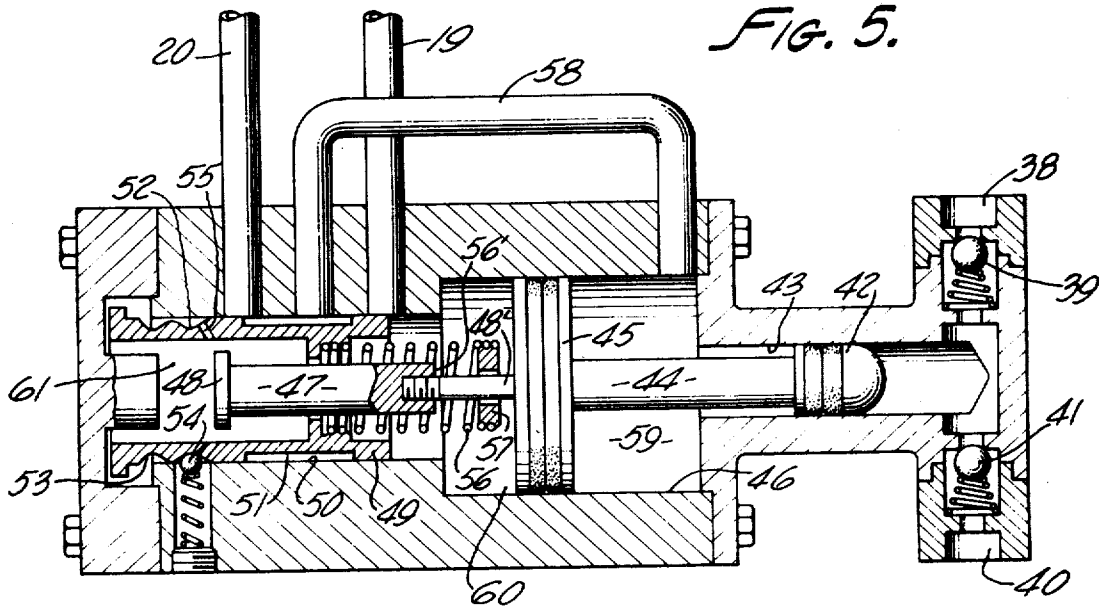
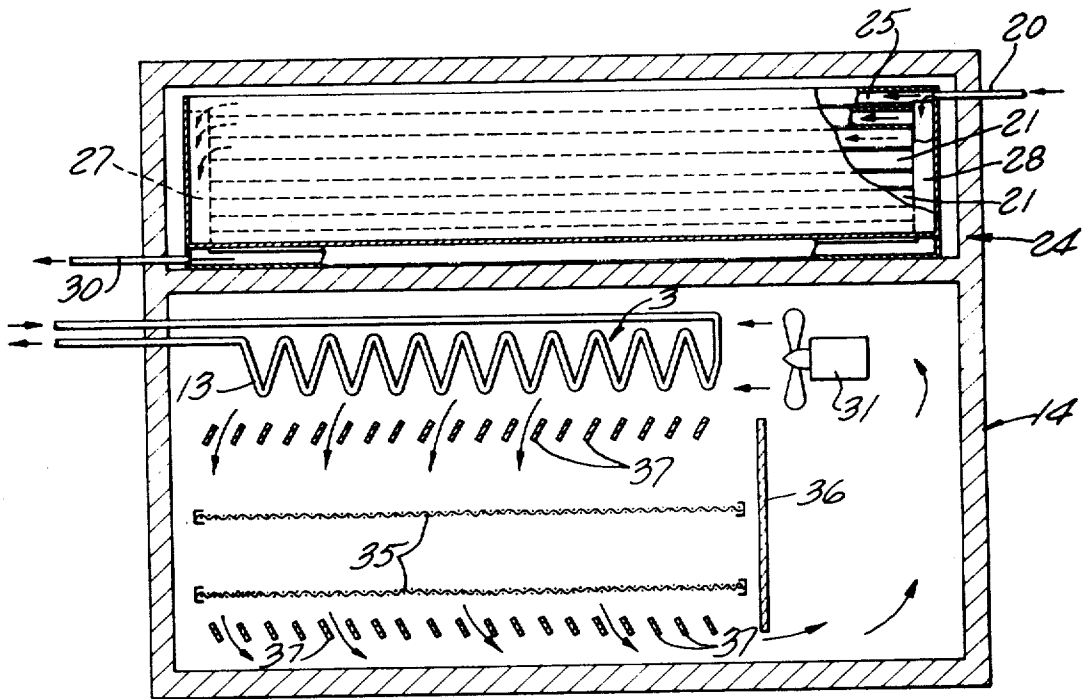


FIG. 6.

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THERMODYNAMIC CYCLES

BACKGROUND OF THE INVENTION

In the conventional Rankine cycle, the system operates at superatmospheric temperatures. In such cycles, the high temperature stage, sometimes referred to as the "boiler" is maintained by the expenditure of energy either chemical or nuclear in form. This energy source forms the consumable energy portion of the cycle. The necessary rejection of heat to the atmosphere involved in the very nature of the conventional Rankine cycle contributes to its relatively low thermal and economic efficiency. Additionally, the relatively high temperature base of the cycle in addition to its low efficiency requires a large capital expenditure per horsepower of usable energy.

SUMMARY OF MY INVENTION

In the preferred embodiment of my invention, I employ, as a means of vaporizing the operating liquid, a heat transfer system employing the ambient atmosphere as the heat source to cause vaporization of a liquid whose boiling point is substantially below ambient temperature. The vapor passes to a condenser operating at substantially low subatmospheric temperature due to presence of a heat sink wherein the vapor is condensed. The condensate is pumped from the relatively low pressure condensation zone into the vaporization zone which is at substantially higher pressure than the condensation zone. The power employed in transferring the condensate to the vaporization zone is derived from the high pressure vapor source of the evaporator. Such means may be an injector which passes with the condensate into the evaporator. I prefer, however, to use a pump in which the motive power is derived from the relatively high pressure vapor generated in the evaporator, the exhaust vapor from the pump is introduced into the condenser in a closed cycle without any exhaust to the atmosphere.

The system of my invention may be a source of refrigeration. The evaporator abstracts heat from the ambient atmosphere and thus causes refrigeration in the surrounding atmospheric space.

Alternatively, or in addition, I may employ a portion of the high pressure vapor source from the evaporator to operate an engine for the generation of power and the exhaust from the engine may also be introduced into the condenser in a closed cycle.

It will be seen that the energy expended in this cycle, which has economic value is only that consumed in the condensation stage of the cycle since the energy absorbed in the evaporation stage is derived from the ambient atmosphere. The economic cost of the power depends on the economic cost of the cooling medium and where the cooling medium is of negligible cost, the power generator has a cost based on operating cost which may be substantially zero.

The heat abstracted from the atmosphere in the evaporator stage is returned to the heat sink medium in the condensation stage, both of which operate at temperatures below the temperature of the heat source which is the atmosphere. This is distinguished from the conventional Rankine cycle in which low efficiency is largely dependent upon the loss of heat to the atmosphere in the high temperature stage and the prime movers and also in the low temperature state, all of

which are at a temperature above the atmospheric heat sink. For this reason, I have denominated my unique cycle an "Inverted Rankine Cycle" to distinguish it from the conventional Rankine cycle.

In my preferred embodiment, I employ the cycle as a refrigerating means and employ dry ice as the cooling medium that is heat sink in the condenser.

The liquid medium employed as the heat transfer medium should, therefore, have a boiling point substantially above the temperature of the solid carbon dioxide, which at 760 m.m. has a sublimation temperature of about -78°C . The liquid medium should have a boiling point substantially below the ambient temperature. In the case of refrigeration, this is preferably below about 5°C in the case of food storage. For example, I may use Freon-12, which is a material sold by Du Pont - De Nemours Co., and said to be dichlorodifluoro-methane having a boiling point at 760 m.m. pressure of -29°C .

DETAILED DESCRIPTION OF THE INVENTION

FIG. 1 is a generalized schematic diagram of my invention.

FIG. 2 is a schematic diagram of the preferred embodiment of my invention.

FIG. 3 is a section on line 3—3 of FIG. 2.

FIG. 4 is a fragmentary section taken on line 4—4 of FIG. 3.

FIG. 5 is a section through the pump shown on FIG. 1 or 2.

FIG. 6 is a modification of the system of FIG. 2.

FIG. 1 illustrates, schematically, the principles of my invention. The evaporator 1 is exposed to ambient temperature in the space surrounding the coil of the evaporator. The liquid contained in the cycle has a boiling point to be vaporized at the ambient temperature at the pressure contained in the coil 1. The condenser 3 is maintained at a temperature below that in the ambient space surrounding 1 by means of a cooling medium which will result in the desired low temperature in the space surrounding the condenser coil 3. The vapor condenses at the lower pressure contained in the condensation zone and the condensate liquid is pumped by feed pump 5 into the evaporator 1 against a higher pressure. The pump 5 is operated by a motor 8 which takes its operating vapor from the high pressure vapors derived from 1 through valve 4. The motor 8 may be in the form of a reciprocating engine and from which the exhaust vapors discharge into the condenser 3 through line 9. The evaporator 1 is connected through valve 4 in line 2, pump motor 8 and line 9 to condenser 3.

The high pressure vapors may be also used to generate additional power beyond that absorbed by the feed pump motor in keeping the cycle continuously operating. A vapor take-off can be at any suitable position in the high pressure vapor stage, for example, through valve 6 into the motor 10 which may be of a positive displacement or turbine kind and the exhaust from 11 is introduced into the condenser 3. The motor 10 may be used to drive any power absorbing unit or a generator such as an electrical generator 12.

In my preferred embodiment, I apply this system to refrigeration in a closed system where the pressure necessary for the efficient operation of the refrigera-

tion system is obtained entirely from the change in enthalpy between the high and low temperature stages including the energy losses in the motors.

A schematic representation of a preferred embodiment is shown at FIG. 2 in which the evaporator 13 encased in an insulated chamber 14 is connected through a variable orifice 15 to the motor 16 of the feed pump 17 through the fixed orifice 18 in line 19. The variable orifice is actuated thermostatically by a temperature sensitive thermostatic device 19 which decreases the orifice opening when temperatures in the ambient space in 14 drops below a predetermined temperature in the ambient space of 14 and opens the orifice when the temperature in the ambient space 14 rises above a predetermined temperature. The thermostatic element 19' may be a mercury or other thermally sensitive unit. Such temperature responsive orifices are well known and need not be described further.

The exhaust from the motor 16 passes through line 20 to tubes 21 positioned in the jacket 21 of the dry ice chamber 23 contained in the insulated box 24. The line 20 connects to top tubes 25, one on each side of the dry ice chamber. The tubes 25 extend across the length of the condenser and are provided end slots 26 to permit entry of vapor into the plenums 27 and 28. The tubes 21 are stacked one on top of another and may be brazed for rigidity. They are mounted in the condenser jacket 22 and are open in both ends at the plenums 28 and 27 to discharge into the plenums 28 and 27. The liquid condensate collects in the bottom of the plenums and is pumped through line 30 by pump 17 and introduced into the evaporator 13.

Circulation in the ambient space of 14 may be obtained by a fan whose motor 31 is operated by the high pressure gas derived from evaporator 13 passing through 32 and through fixed orifice 33 and the exhaust is introduced to line 34 into the tube 25.

The orifices 18 and 33 act to regulate the portion of the vapor, passing the fixed orifice, which is employed in operating the fan.

FIG. 6 shows a preferred embodiment of my invention shown in FIG. 2 employed in refrigeration. Like elements bear the same numbers as in FIG. 2.

The condenser 3 is mounted in insulated chamber 14 positioned in a unitary arrangement with the insulated chamber 24. The evaporator unit is mounted adjacent the top of the chamber 14, and the fan and motor 31 is mounted to circulate air over the coils 13 and over and through the trays 35 aided by suitable baffles 36 and louvers 37.

While any suitable pump may be employed, I prefer to use a pump operated by a positive displacement motor which is capable of reciprocating at low frequencies and which will avoid being hung up at dead center. Such a pump is shown in cross section in FIG. 5.

The inlet 38 is provided with a suitable check valve 39 and the outlet 40 is provided with a suitable check valve 41. The pump piston 42 in pump cylinder 43 is connected to a piston rod 44, which is connected to the power piston 45 in the motor cylinder 46. The pistons are each provided with a suitable seal. The power piston is screw connected to a valve stem 47 by pump piston extension 48. The valve stem 47 carries a flanged head 48. The valve barrel 49, slidably positioned in the valve cylinder 50, carries an elongated

cylindrical groove 51 and circular detent grooves 52 and 53 co-operating with a spring loaded detent 54. The valve barrel is pierced with bores 55. Push-pull spring 56 is connected at one end to the valve barrel 49 and at the other end to spring retainer 57.

The high pressure inlet line 19, downstream from the variable orifice 15 and the fixed orifice 18 is connected to the input at the valve cylinder 50 and the exhaust 20 from the pump is connected to the valve cylinder as shown. A by-pass is provided at 58 connecting the grooved space 51 with the space 59 of the power cylinder.

In the position as shown in FIG. 5, showing the position at the completion of the stroke to the left of the drawing as shown, the piston 45 is in position for the initiation of the outbound stroke to the right. The space 59 is connected to the exhaust 20 and the space 60 ahead of the piston 45 is connected directly to the inlet line 19. Cavity 61 is at the same pressure as 60 due to the clearances between the valve stem, spring retainer, spring, valve barrel and the orifice 40.

The piston 45 thus starts moving to the right, vapor exhausting through 58, 51 and 20 until the end 56' of the stem 47 engages the spring retainer 57 and the continuing motion of the piston causes a stretching of the spring 41 which continues until the spring tension creates a tractive effort sufficient to overcome the spring pressure of the spring loaded detent ball 54. The valve barrel snaps to the right until the detent ball enters the groove 53. In that position the valve barrel has moved to uncover the inlet 19 to register it with the groove 51 closing off 20 and therefor connecting space 59 through the by-pass 58 to the inlet 19. The exhaust outlet 20 is connected to the space 60 through port 55, grooves 52, 53, cavity 61 and the clearances described above.

The piston 45 and the valve stem 47 moving to the left releases the spring tension in the spring 56. The continued motion of the piston 45 engages the retainer 57 and the resultant compression of the spring in normal operation is sufficient to overcome the spring loaded detent whereupon the barrel snaps to the position shown, where the spring loaded detent ball 54 engages the groove 52 to complete the cycle of reciprocation of the piston 45 and piston rod 44 causes a pumping action of the liquid into 38 and out of 40.

The permissible stroke of piston without inducing a movement of the valve barrel may be controlled by adjusting the length of the rod 48', thus controlling the stroke of the piston.

Because of the presence of foreign substances, the valve barrel may stick so that the spring 56 when it extends to the right develops insufficient force to move the valve barrel and to pull it away from the detent ball 54. In such case, piston 45 will continue to move to the right until the flanged head 48 engages the valve barrel shoulder and the continuing motion 45 will exert sufficient force to free the valve member barrel and pull it away from the detent ball 54.

Should the valve barrel stick during the inboard motion of the piston to the left, and the compressive force on the spring be insufficient to dislodge the valve barrel, the piston 45 will continue its motion to the left until the spring is compressed establishing a solid connection between the piston 45 and the valve barrel to

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dislodge the barrel, whereupon the expansion of the spring will cause the valve barrel to snap to the position shown in FIG. 5.

Driving the valve barrel by means of the spring force imparts to the system a very large frequency response range so that the valve reciprocates properly at both high and low reciprocation rates.

The driving spring force co-acts with the spring loaded detent to cause the valve stem to snap from one position to the other and the valve is not subjected to inertia forces inherent in the piston portion of the system.

The valve therefor cannot be hung up at dead center at the end of either the inbound or outbound stroke.

I claim:

1. A refrigerant cycle operating between two levels of temperature, including an evaporation zone exposed to the temperature to be controlled by the cycle, a condensation zone operating at a lower temperature than the evaporation zone, a vapor conduit connecting the evaporation zone with the condensation zone, a liquid collecting zone connected to said condensation zone, a

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liquid conduit connecting said liquid collecting zone with said evaporation zone and a pump in said conduit, a motor in said vapor conduit, a fan positioned adjacent to said evaporation zone, a fan motor connected to said fan, a vapor conduit connecting said first mentioned vapor conduit to said last-named motor, an exhaust vapor connection connected to said last-named motor and said condensation zone.

2. In the cycle of claim 1, said evaporation zone comprising a coil connected to said pump and to said first mentioned vapor conduit, an insulated enclosure for said coil, said fan mounted for circulation of air over said coil and in said enclosure.

3. In the cycle of claim 2, a thermostatic element mounted in said enclosure responsive to the temperature in said enclosure, a variable orifice mounted in said first mentioned vapor line between said condenser and said first mentioned motor, an operative connection between said thermostatic element and said variable orifice to vary the opening of the orifice responsive to the variation in temperature in said enclosure.

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UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,693,370 Dated September 26, 1972

Inventor(s) David T. Miller

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Claim 1, column 6, line 3, after "conduit" should read -- connected to said pump --.

This certificate supersedes the Certificate of Correction issued December 4, 1973.

Signed and sealed this 16th day of April 1974.

(SEAL)
Attest:

EDWARD M. FLETCHER, JR.
Attesting Officer

C. MARSHALL DANN
Commissioner of Patents