March 28, 1961 G. MUFFLY **2,976,698**

REVERSIBLE REFRIGERATING SYSTEMS

Original Filed Sept. 19, 1951 3 Sheets-Sheet 1

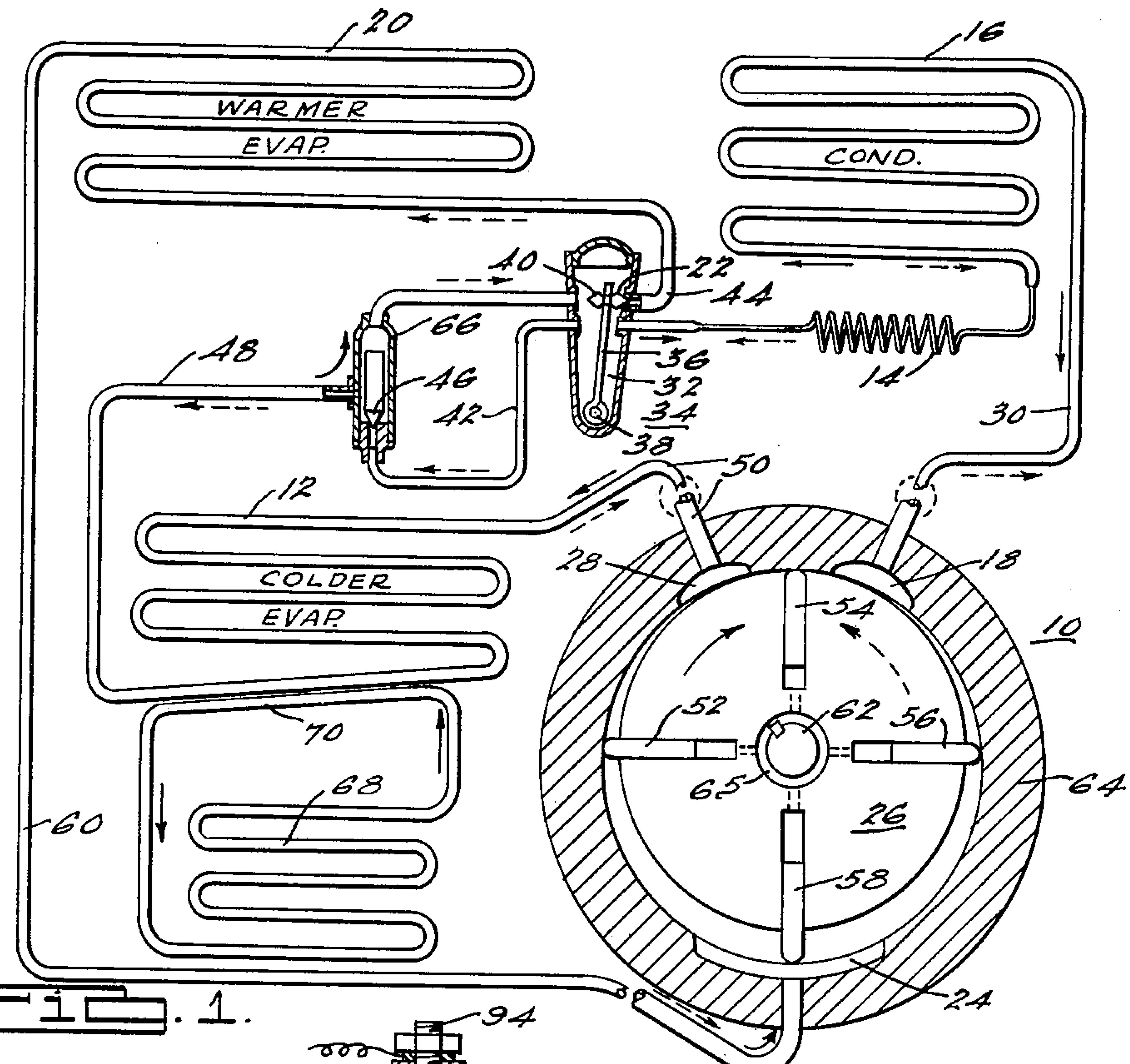

FIG. 1.

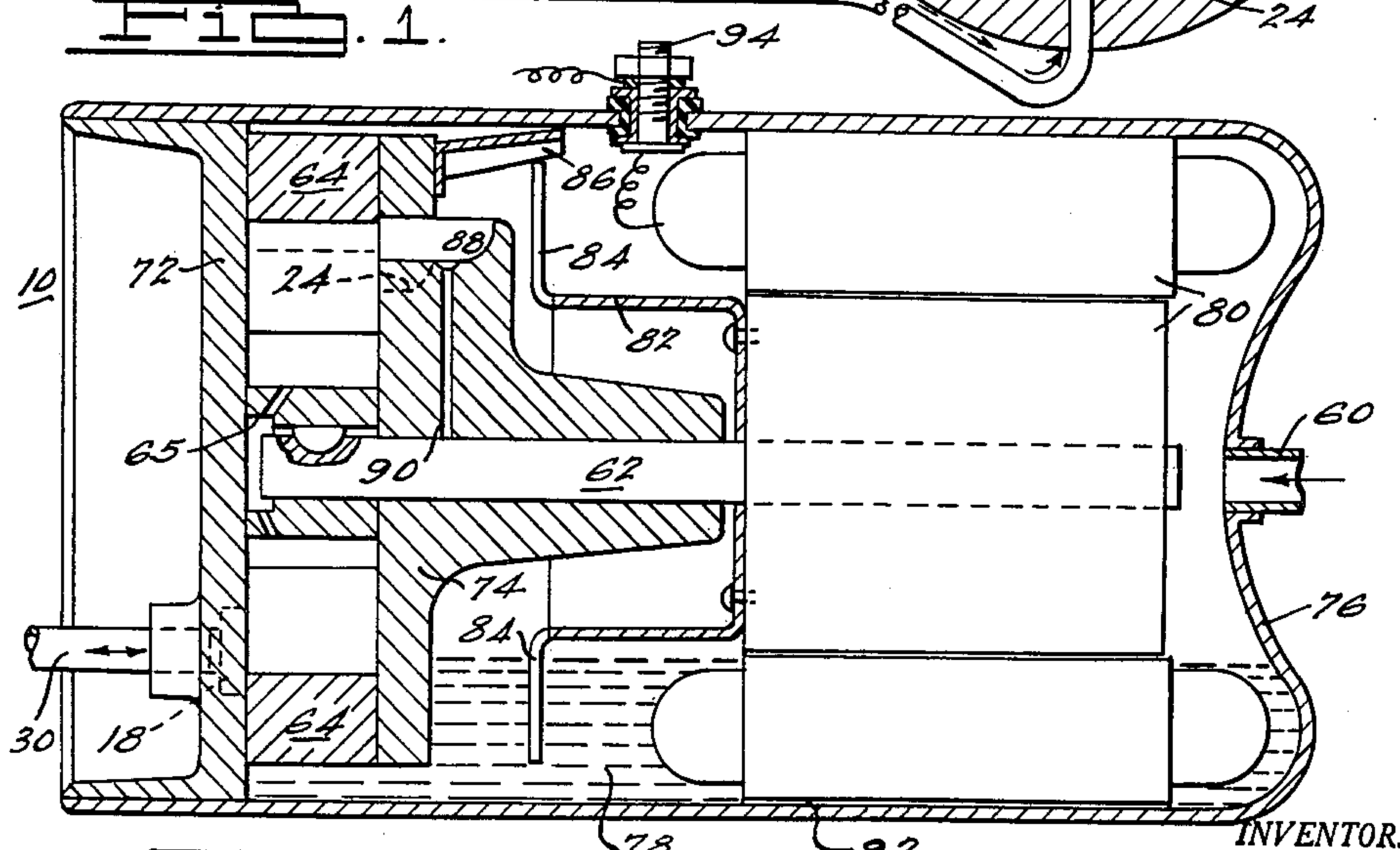

FIG. 2.

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REVERSIBLE REFRIGERATING SYSTEMS

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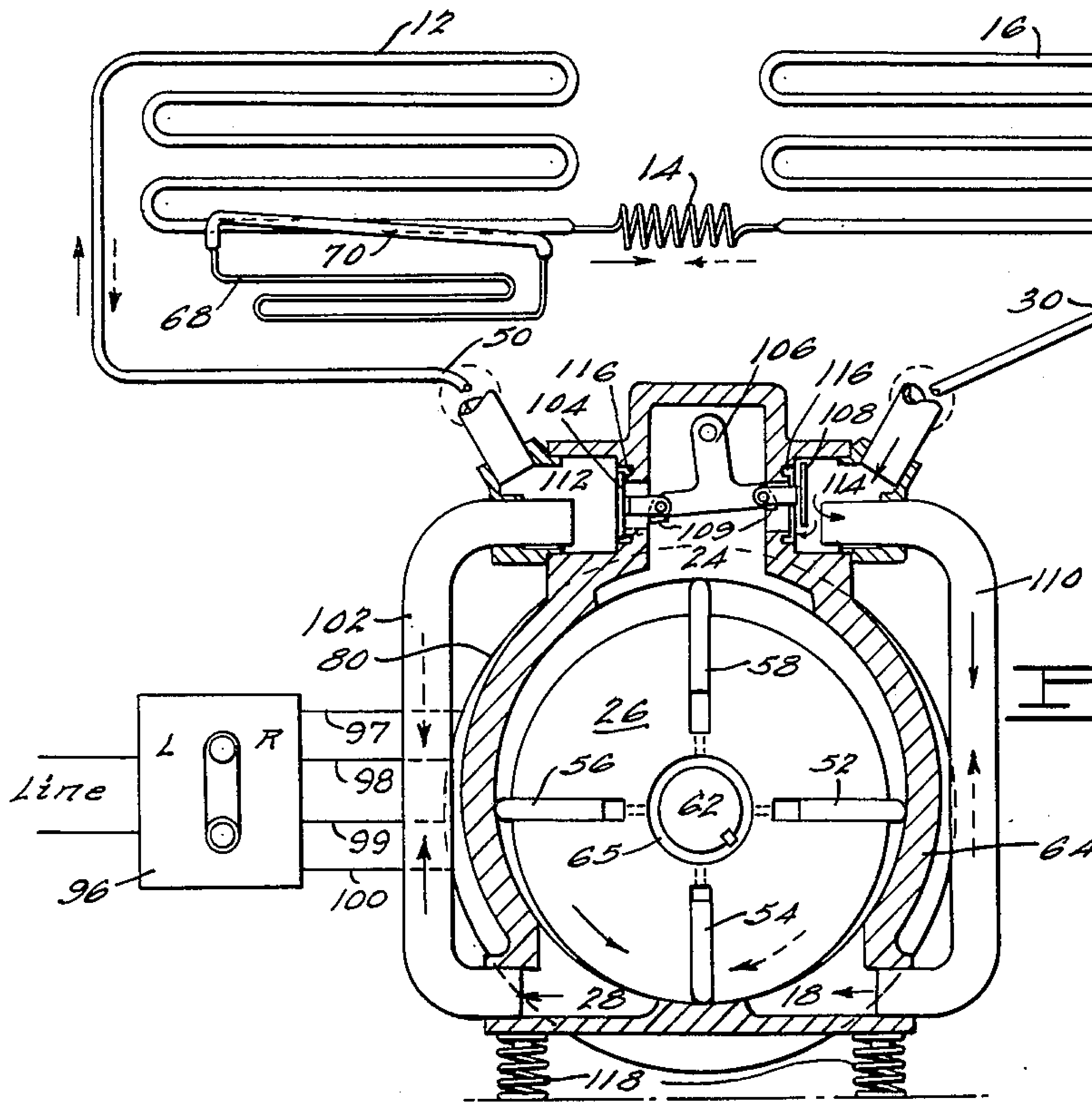


FIG. 3.

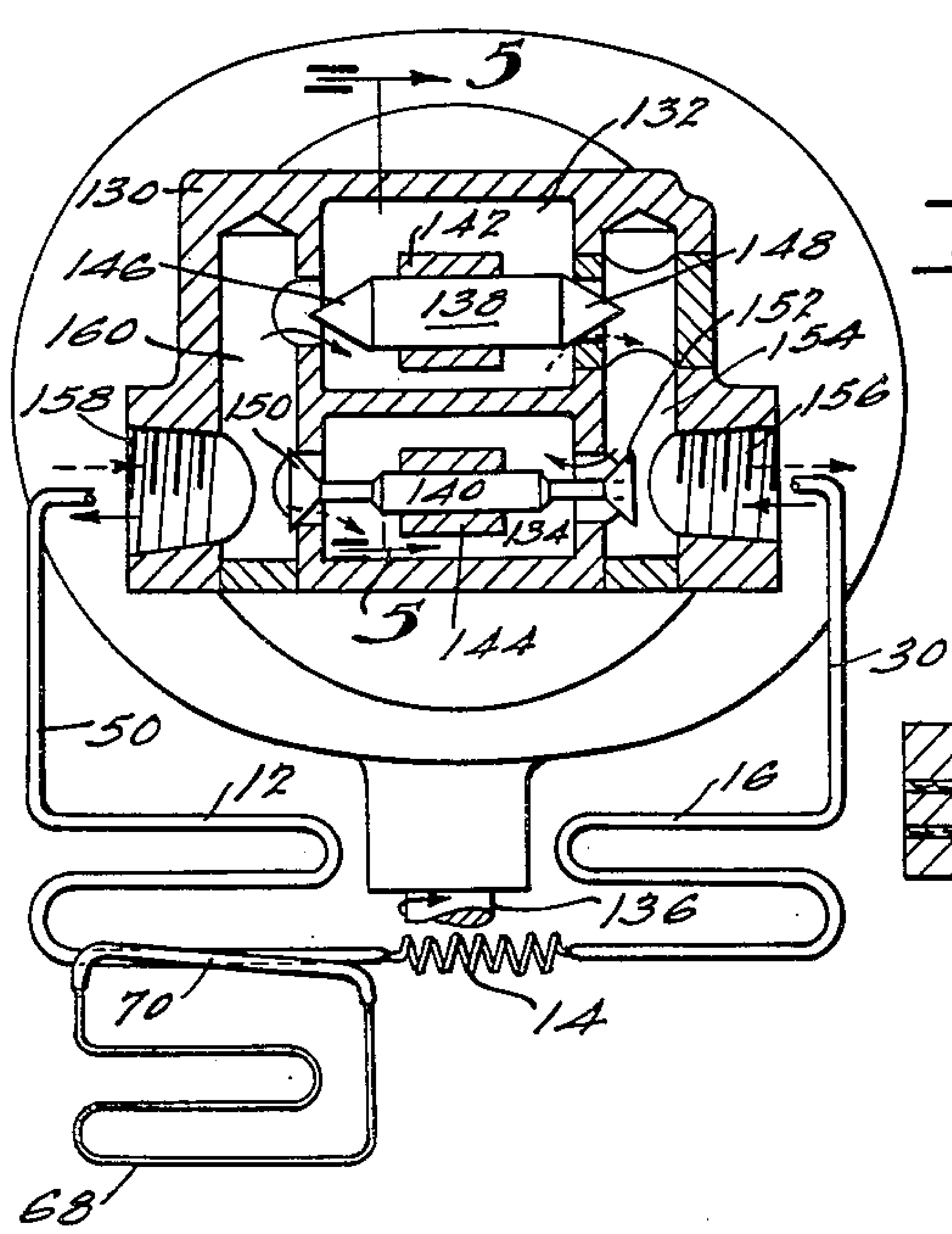
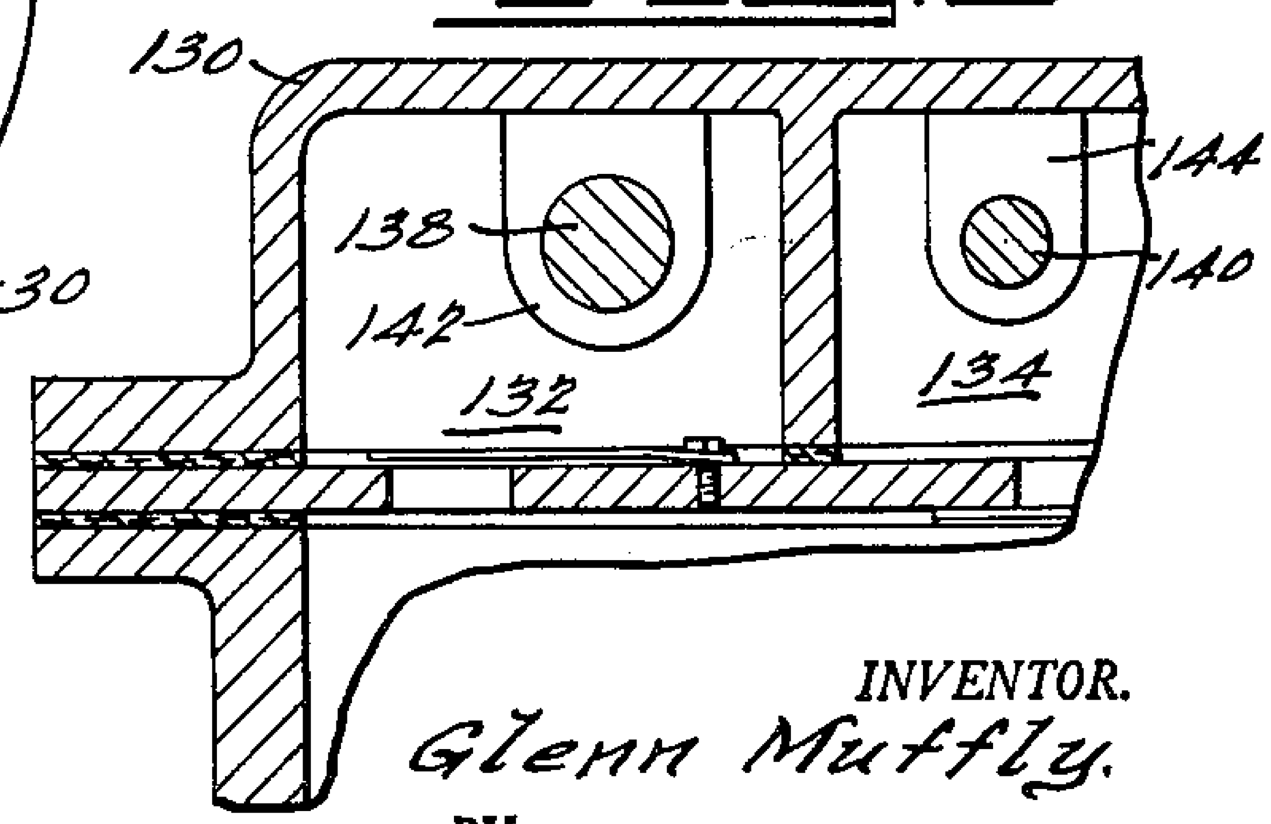


FIG. 4.

FIG. 5.



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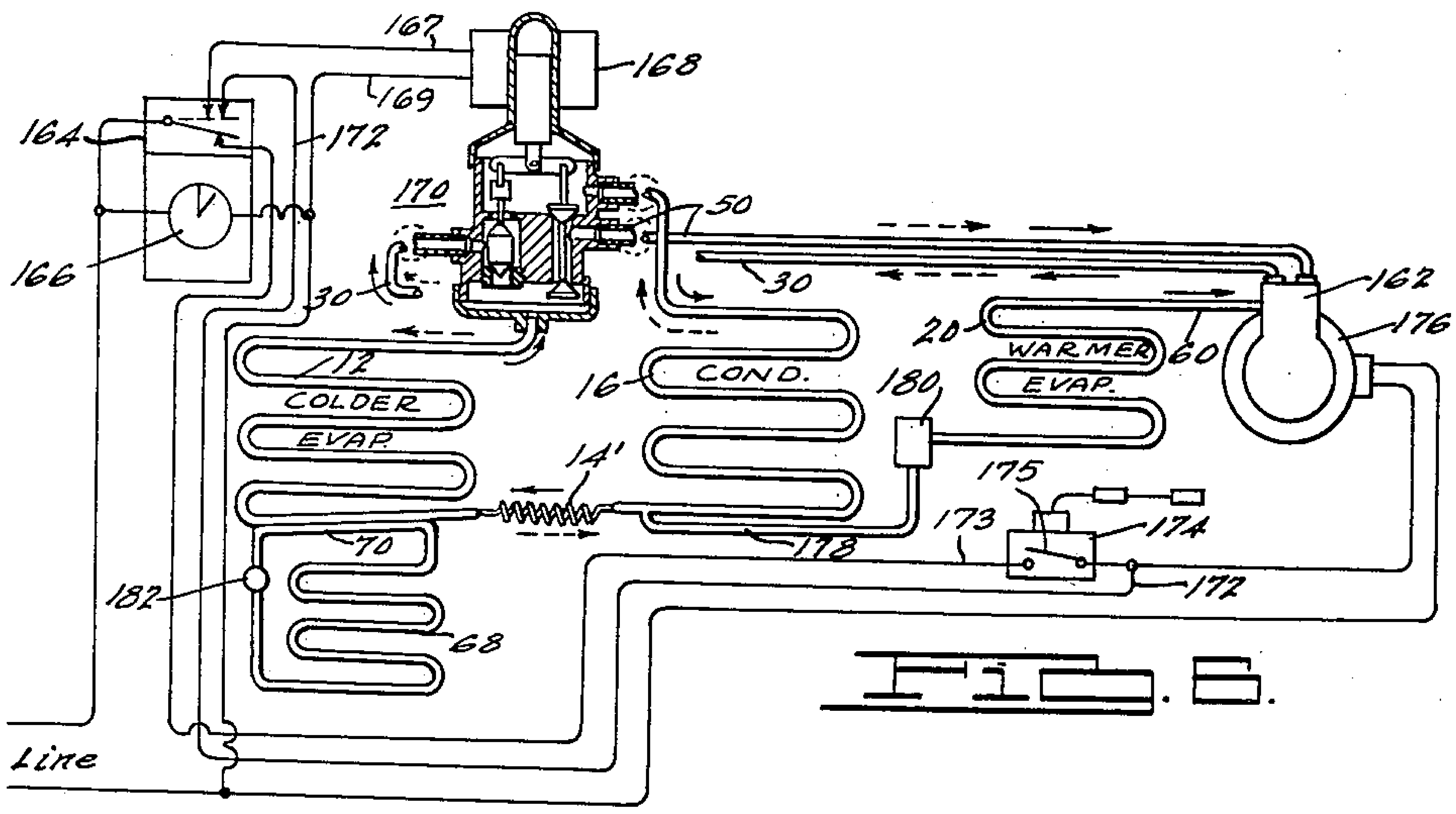

FIG. 6.

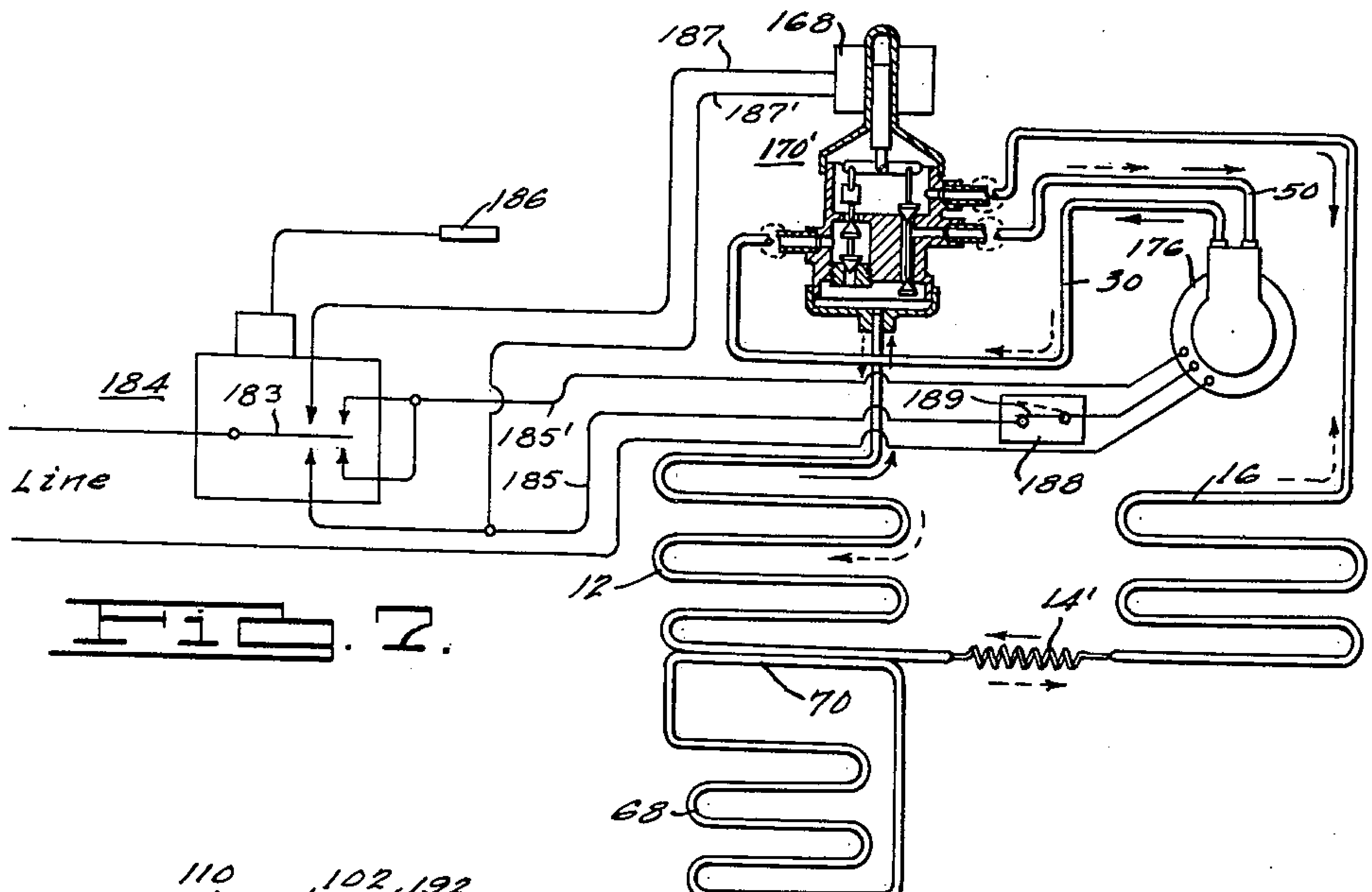

FIG. 7.

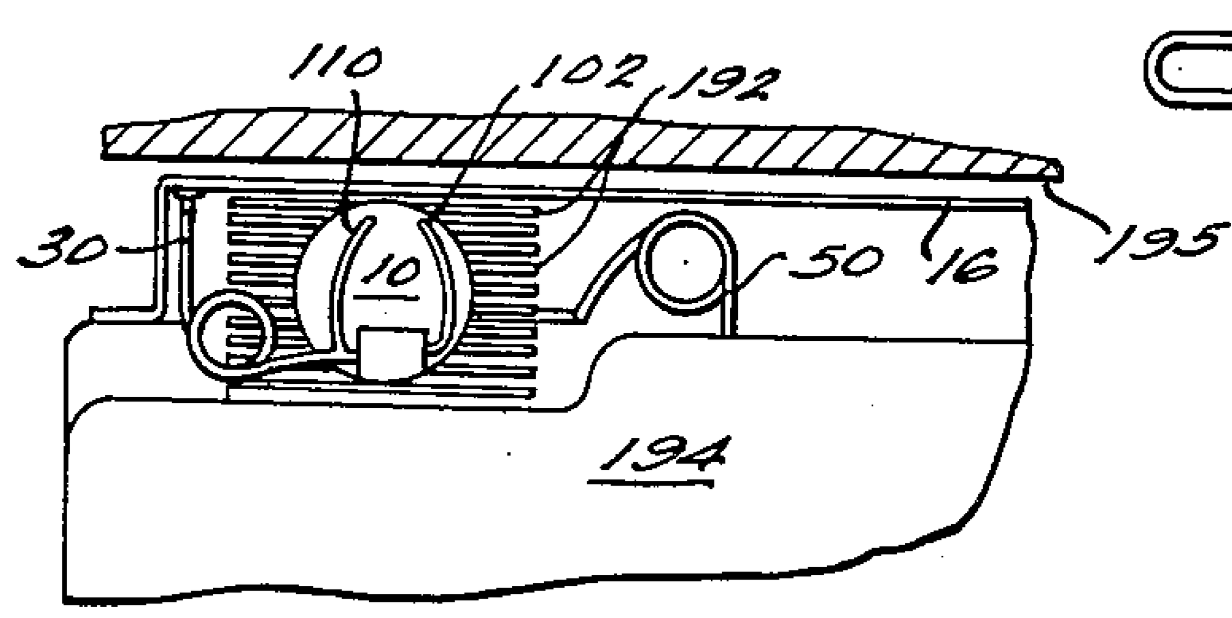

FIG. 8.

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REVERSIBLE REFRIGERATING SYSTEMS

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Original application Sept. 19, 1951, Ser. No. 247,239, now Patent No. 2,844,945, dated July 29, 1958. Divided and this application Sept. 11, 1957, Ser. No. 683,335

15 Claims. (Cl. 62—160)

This application is a division of my copending United States application Serial Number 247,239, filed September 19, 1951, issued July 29, 1958, as Patent No. 2,844,945, and relates to defrosting and to reversing the direction of fluid flow with or without the direction of compressor rotation being reversed.

There is a need for a reversible compressor and/or valve mechanism in reverse-cycle refrigerating and air conditioning systems, and particularly for defrosting freezer evaporators of two-temperature refrigerators.

No refrigeration compressor now on the market is suitable for reversing refrigerant flow, hence complicated valve arrangements are coming into use. Piston type compressors continue to pump in the same direction when their direction of rotation is reversed. Rotary compressors in which the piston or rotor is carried by a crank pin or eccentric require discharge valves which prevent operation of the compressor in reverse. The gear or lobed impeller type of compressor reverses its direction of fluid flow when the direction of rotation is reversed, but this type has not proven satisfactory and is not currently in production in the refrigeration field because of noise and the fact that gas leakage increases with wear. The type of rotary compressor in which a rotor is eccentrically located with respect to its cylinder and mounted concentrically on the drive shaft usually carries one or more sliding radial vanes which sweep the clearance space. This latter type of compressor does reverse its direction of pumping when reversed in rotation if equipped with three or more vanes, but such reverse pumping is inefficient since the suction and discharge ports are not suited for reversing their functions.

When four or more radial vanes are carried by the rotor it is practical to dispense with the usual check valves located in the inlet and discharge ports, but for best results the discharge port should be smaller than the suction port and the latter should extend farther around the cylinder in order to allow proper filling of the displacement space. This has prevented the development of a satisfactory reversible compressor.

It is an object of this invention to provide a compressor which reverses its direction of pumping when its direction of rotation is reversed and has good efficiency when operated in either direction.

Another object is to provide a compressor having a pair of ports which interchange their functions of inlet and discharge when the compressor rotation is reversed and to provide an additional port which serves as an auxiliary inlet port for both directions of rotation.

Another object is to provide self-actuating valve means for connecting the last mentioned port with the suction conduit and shifting this connection to another suction conduit when the compressor rotation is reversed.

Another object is to provide a compressor which allows high and low side pressures to equalize when idle.

Another object is to provide a valve mechanism responsive to the starting of the compressor in either of

two manners to close and open the proper ports in relationship to the direction in which the compressor has been started.

Another object is to provide reverse-cycle defrosting of a freezer evaporator in a primary system which is combined with a secondary system of which the evaporator is not to be defrosted.

Another object is to provide a pair of opposed check valves so interconnected as to cause the closing of one valve to open the other when the system is started and to allow each valve to assume a partly open position each time the compressor is stopped.

Another object is to provide a check valve and its associated chamber so proportioned that when the valve is in its idle partly open position it nearly closes the passage so that a slight flow of fluid will cause the check valve to close, yet when the valve is fully opened it offers no serious obstruction to the flow of fluid in the direction which would otherwise tend to close the valve.

Another object is to provide for defrosting the colder evaporator only in a multiple-temperature system using either a conventional compressor or a compressor with the "multiple-effect" or dual-suction arrangement which provides two distinct suction pressures.

Still another object is to achieve the utmost simplicity in a two-temperature system with automatic defrosting of the colder evaporator, such defrosting being accomplished automatically by a simple reversal of motor rotation or shifting of valves without affecting the temperature of the space cooled by the warmer evaporator.

In the drawings:

Fig. 1 is a diagrammatic view of a two-temperature refrigerating system employing a new type of compressor and valve mechanism which combine to reverse flow in the system.

Fig. 2 is a longitudinally sectional view of a compressor such as illustrated diagrammatically in Fig. 1, but showing the preferred arrangement of ports, with the constant inlet port at the top.

Fig. 3 is a sectional and diagrammatic view of a similar compressor as used in a heat-pump or "reverse-cycle" system with a reversible valve incorporated in the motor-compressor unit and arranged to control vapor flow instead of liquid flow.

Fig. 4 is a top view of a reversing valve mounted on the head of a conventional compressor, to provide reversal of both discharge and suction where the compressor does not in itself reverse flow when its direction of rotation is reversed.

Fig. 5 is a detail sectional view on the line 5—5 of Fig. 4.

Fig. 6 is a diagrammatic view of a system similar to Fig. 1 but using a reciprocating multiple-effect compressor and a clock switch to control the valves.

Fig. 7 is a diagrammatic view of a similar system but using a conventional compressor and having the electrically actuated valves under thermal instead of clock control.

Fig. 8 is a bottom view of a sealed motor-compressor unit designed for use in a system having its high side on the rear wall of a two-zone refrigerator to conserve space and aid in disposal of defrost water.

Fig. 1 shows the compressor **10** driven clockwise, which is the direction of rotation for defrosting the freezer evaporator **12** with hot high pressure refrigerant vapor, which condenses therein and flows through the restrictor tube **14** to the condenser **16**, in which the condensed refrigerant evaporates, and returns to the compressor at the port **18** which serves as an inlet port during this defrosting operation. Since it is assumed that the system was operating shortly before defrosting, there will be some liquid refrigerant remaining in the warmer evaporator 20 when defrosting starts, although the inlet to this evaporator is now shut off by the valve 22. Vapor will continue to flow from the evaporator 20 to the auxiliary inlet port 24 of the compressor during a considerable portion if not all of the short defrosting period of the freezer evaporator 12. The rotation of rotor 26 of the compressor and the path of refrigerant flow during this defrosting period are indicated by solid arrows.

At the completion of the defrosting operation the compressor rotation stops and then restarts in the opposite direction as indicated by the dotted arrow on the rotor 26. This causes the port 28 to serve as a suction inlet to the compressor, which draws refrigerant vapor from the freezer evaporator 12 and discharges it at port 18 through the tube 30 to the condenser 16. Liquefied refrigerant flows from the condenser through the vapor-lock restrictor 14 to the chamber 32 of the reversing valve 34. The impact of the resulting jet of liquid refrigerant mixed with its flash gas striking the vane 36 causes this vane to tilt upon its pivot 38, opening the valve 22 and closing the valve 40. Liquid refrigerant now leaves the chamber 32 by way of the tubes 42 and 44 after filling the lower portion of the chamber 32 with liquid. When a sufficient pressure had developed within the tube 42 in excess of the pressure existing in the freezer evaporator 12 the weighted check valve 46 lifts and refrigerant liquid flows through the tube 48 into the lower temperature evaporator 12, from which vapor flows through the tube 50 to the inlet port 28 of the compressor.

It will be noted that the spacing of the ports 28 and 24 is such that they are always divided by one or more of the radial vanes 52, 54, 56, and 58 so that vapor withdrawn from the evaporator 12 is trapped in the space between two vanes, such as 52 and 54, before this space comes into communication with the auxiliary inlet port 24, at which time higher pressure suction vapor from the evaporator 20 flows into the space between these two vanes. This space reaches its maximum expansion at the time the two vanes reach their symmetrical position straddling the port 24, but due to the velocity of suction vapor in the tube 60 it will continue to flow from tube 60 into the compressor until approximately the cut-off point at which the vane 54 cuts the compression space off from the auxiliary inlet port 24.

The trapped vapor is now compressed and delivered to the port 18, such discharge continuing until the vane 54 opens the port 18 to the next compression space between the vane 54 and the vane 56. There will be some reverse flow between the port 18 and these arcuate spaces, but with the proper arrangement of ports and the proper number of blades there can be no actual leak-back except the limited amount of unavoidable leakage past the working surfaces of the compressor.

The shaft 62 carrying rotor 26 is preferably the shaft of a two-pole alternating current motor which, on 60 cycle current, will operate at about 3400 to 3500 r.p.m. It is desirable to keep the motor diameter down to a minimum for reasons which will appear later herein, hence in Fig. 1 it is assumed that the rotor and stator of the motor are hidden back of the compressor cylinder 64. The motor and compressor are preferably enclosed within a sealed casing to which the tube 60 leads and from which vapor flows into the port 24. The cylinder 64 is provided with end plates or heads in the customary manner and it is preferred that the ports 18, 24 and 28 be formed by recesses in one or both of these heads rather than as shown diagrammatically in the cylinder bore in Fig. 1. The blades of rotary compressors are often provided with springs to hold them in engagement with the cylinder bore, but such springs may be omitted in this case because of the high speed of rotation which provides ample centrifugal force to hold the blades against the cylinder bore.

In order to prevent the development of a partial vacuum in the slot back of a blade, which would interfere with the operation of centrifugal force, and to prevent the accumulation of oil in a slot which would interfere with the free movement of a blade, I have shown a counterbore 65 connected with all of the slots by means of radial holes, which are indicated by dotted lines.

When the motor is stopped with a considerable pressure difference existing between the tubes 30 and 50 the compressor is free to reverse its direction of rotation and act as a vapor-expansion motor until these pressures are nearly balanced. It is desired that pressures within the system equalize during idle periods of the compressor and the tendency of the compressor to act as a motor aids the electric motor in starting its reverse rotation when the controls of the system cause an instantaneous switch from cooling to defrosting or vice-versa.

The housing 66 which encloses the weighted check valve 46 may be incorporated with the valve assembly 34 to eliminate the tubes which connect these two housings, but it may be preferred to keep them separate so that the housing 66 can be located adjacent to the colder evaporator 12 while the valve assembly 34 is located adjacent to the warmer evaporator 20. In either case it is preferred that the internal volume of the chamber 32 below the level of the various ports be kept at a minimum to conserve refrigerant liquid. All four of the ports connecting with the chamber 32 may be located at the same level, though shown at two different levels in Fig. 1 so that the refrigerant flow paths can be more easily traced. The freezer evaporator 12 may be located either above or below the warmer evaporator 20 and the weighted check valve 46 may be located either above or below the reversing valve assembly 34, hence this system is adaptable for use in a two-zone refrigerator with the freezer compartment either above or below the main food compartment.

Fig. 1 illustrates the arrangement of ports and refrigerant passages for cooling a pair of evaporators simultaneously, one at a lower evaporating pressure than the other. The condenser 16 receives refrigerant from both evaporators while they are being cooled, but when the colder evaporator 12 is being defrosted by reserve operation of the compressor the warmer evaporator 20 is not affected since it is isolated by the closed valve 22.

A third evaporator 68, comprising a part of a secondary refrigerating system, may be combined with this system by locating the secondary condenser 70 in heat exchange with a portion of the evaporator 12, as indicated in Fig. 1. If a secondary system such as 68—70 is used in place of the warmer evaporator 20, or only one evaporator is required, the valves 34 and 46 may be eliminated, connecting the vapor-lock restrictor 14 directly with tube 48. Such a system is illustrated in Fig. 3. The elimination of evaporator 20 makes the port 24 available for withdrawal of refrigerant vapor from the evaporator 12, but it is then preferred to stop it off from evaporator 12 during the defrosting period, as will be explained later with reference to Fig. 3. The secondary evaporator 68 will not be affected by the defrosting of evaporator 12 since the heating of the secondary condenser 70 merely stops the flow of vapor from 68 to 70, thus trapping the entire charge of secondary refrigerant in the evaporator 68.

Fig. 2 shows a preferred arrangement of the compressor of Fig. 1 with ports in the end walls 72 and 74 instead of in the cylinder barrel 64. The auxiliary intake port 24 in the wall 74 is open to the interior of the sealed housing 76 and hence placed at the top to be above the level of oil 78. The suction tube 60 enters the housing on the far side of the motor 80 so that suction vapor from evaporator 20 aids in cooling the motor. The ports 18 and 28 are formed in the end plate 72 which closes the housing. Tubes 30 and 50, connected with these ports, are brazed to the end plate 72. An oil slinger 82, shown in the form of a drawn cup attached to the rotor of the motor, has two or more arms 84 which, in either direction of rotation, throw oil against the sheet metal baffle 86 which is attached to the plate 74 and arranged to carry oil into the pocket 88, which leads to port 24 and also to the shaft through the hole 90.

The stator of the motor is pressed into the housing 76 and provided with a longitudinal oil passage 92 so that the level of oil 78 is maintained on both sides of the motor. Four terminals of the motor winding, as 94 of Fig. 2, are connected with a suitable reversing switch, as 96 in Fig. 3, which may be operated manually, by temperature or pressure changes, by count of cycles or openings of cabinet or by a clock, for the purpose of starting the motor in either direction of rotation. In one closed position of switch 96 one of the wires of the line may be connected with wires 97 and 98 while the other line wire is connected with 99 and 100. In the other closed position of switch 96 one of the line wires would then be connected with 97 and 99 while the other line wire connects with 98 and 100 for the reverse rotation of the compressor.

Referring now to Fig. 3, which shows a rotary compressor similar to those of previous figures. This compressor is assumed to be of the open type, i.e. not incorporated in a sealed unit, and to be connected with two heat exchangers 12 and 16 in series with the restrictor 14 between them. The rotor 26, which is concentric with and driven by an electric motor, carries four blades 52, 54, 56 and 58 which are free to slide in their radial slots in the rotor so that they are held in contact with the bore of cylinder 64 by centrifugal force whenever the compressor is operating. Since there are no springs holding the blades in contact with the cylinder bore, there are no check valves which remain closed when the compressor is idle and the compressor may be driven in either direction by excess pressure in one of the lines 30 or 50, it is seen that high and low side pressures will equalize very soon after the compressor is stopped.

In Fig. 3, as in Fig. 1, it is assumed that low pressure refrigerant vapor is flowing to the compressor from the tube 30, as indicated by the solid arrow and that high pressure refrigerant vapor is being discharged through the tube 50 as indicated by the solid arrow. This means that the rotor is being driven counterclockwise, as indicated by solid arrow, and that there is high pressure refrigerant vapor in the tube 102 which leads to the check valve 104. This pressure holds the check valve 104 closed and thereby through the medium of the rocker 106 the check valve 108 is held open. Suction vapor is thus free to enter the compressor through the suction port 24 and by way of the branch tube 110 and the port 18. Low pressure refrigerant vapor enters the increasing clearance pocket between adjacent blades 52 and 54. This clearance pocket increases in volume after passing out of communication with the first inlet port 18, but soon thereafter it comes into open communication with the port 24 which extends a considerable angle on each side of the vertical center line. Suction vapor will continue to enter the arcuate compression space between blades after this space has started to reduce in volume, but before there is any appreciable back flow of vapor from the compression space into the elongated port 24 this port will be cut off by blade 54. From this point on the compression space decreases in volume and substantially all of the vapor is discharged into the tube 102. After the blade 54 has passed the port 28 there will be a return flow of high pressure vapor into the next compression pocket but not beyond it. It is thus seen that the customary discharge check valve is not required in the port 28.

The valve 104 will be held firmly closed by high pressure refrigerant vapor as long as the compressor is operated counterclockwise as indicated by solid arrow in Fig. 3. Due to the arrangement of the tubes 102 and 50 it will be seen that oil is centrifugally separated from the discharge vapor to collect in the chamber 112 ahead of the closed check valve 104. This oil remains so trapped until the next idle period of the compressor, at which time it will flow into port 24 or port 28 and reenter the compressor when next started in its reverse direction of rotation.

As shown in Fig. 3 the valves 104 and 108 are each hinged to 106 and 108 is additionally supported by one of the ears 109 formed on 106, thus the open valve 108 and the rocker 106 exert gravitational forces tending to open the valve 104, and their neutral position of rest is with each valve partly opened. It will be seen in Fig. 3 that there is only a small clearance between each valve and the recess which it enters in its neutral position. In the event of vapor flow toward the valve 104 from the tube 102 this valve will be pushed closed and the valve 108 pushed to its position of maximum opening, thus no matter in which direction the compressor is started the valves 104 and 108 will adjust themselves to the proper relationship which allows suction vapor flow to the port 24 from the low pressure side of the system and stops flow to port 24 from the high pressure side of the system.

As shown in Fig. 3, with counterclockwise rotation of the compressor rotor 26, high pressure refrigerant vapor is being discharged at the port 28 to the tube 102 which leads upward to the chamber 112, this chamber being stopped off from the port 24 by the valve 104 which is closed, hence the high pressure refrigerant vapor is delivered to the tube 50 which leads to the heat exchanger 12, now serving as the condenser of the system. Condensed refrigerant flows through the restrictor 14 to the heat exchanger 16, now serving as an evaporator, and the evaporated refrigerant returns through the tube 30 to the chamber 114, which is now serving as a suction chamber. The refrigerant vapor is free to flow through the tube 110 to the primary intake port 18 and past the open valve 108 to the secondary intake port 24.

When the compressor is stopped the pressure within the system will substantially equalize due to the open restrictor 14, to the fact that the compressor may be turned in reverse by high pressure vapor from heat exchanger 12 and to the absence of centrifugal force allowing vapor to pass the vanes carried by rotor 26. The weight of the rocker 106 and of the open valve 108 now cause the rocker 106 to drop to its neutral position at which both valves 104 and 108 are open. In this neutral position each valve enters its counterbore 116 where it offers considerable resistance to vapor flow without closing tightly.

Assuming now that the compressor is started in the direction of rotation indicated by the dotted arrow on the rotor 26, either by reversing the motor which drives the shaft 62 or by the use of a reversing mechanism between this shaft and its source of power. The suddenly applied torque causes the body of the compressor, which is mounted on flexible supports 118, to rotate slightly so the right at the instant of starting. The rocker 106 and the two valves pivoted thereto will, due to their inertia, lag behind the sudden clockwise jerk of the compressor body thus causing the valve 108 to close while at the same instant vapor discharged through the port 18 to the tube 110 impinges upon the valve 108 to aid in holding it closed until the pressure within the chamber 114 builds up to positively hold the valve 108 closed.

This closing of the valve 108 causes the valve 104 to be lifted farther from its seat and clear of its counterbore 116 so that suction vapor is now free to flow from the tube 50 to the auxiliary intake port 24 as well as to port 28, which now becomes the primary suction port. This clockwise rotation of the compressor causes the heat exchanger 16 to begin operation as the condenser of the system. Liquid refrigerant now flows to the left through the restrictor 14 as indicated by the dotted arrow and it evaporates in the heat exchanger 12, which now serves as the evaporator of the system, delivering refrigerant vapor to the tube 50, which is now serving as the suction tube. Suction vapor is free to enter the main suction port 28 of the compressor and also to pass the open valve 104 from the chamber 112 to the auxiliary suction port 24.

It is thus seen that the ports 18 and 28 have exchanged their functions and now serve as discharge and suction ports respectively, whereas the auxiliary suction port 24 continues to operate as the auxiliary suction port, being open to heat exchanger 12 instead of to heat exchanger 16. This use of the auxiliary suction port to receive vapor from the same evaporator from which vapor is flowing to the active suction port 18 or 28 is suitable for a system of the "heat pump" type in which reversed operation may continue for several hours instead of for a few minutes as explained in connection with Fig. 1.

Fig. 1 represents the "multiple effect" use of the compressor and its use for the purpose of defrosting the colder evaporator, whereas in Fig. 3 the same principle is shown as utilized in an air conditioning system of the so-called "reverse cycle" type, which either heats or cools the room. In the latter case there is no need for two distinct evaporating temperatures and normally neither one of the two heat exchangers collects frost during its operation as the evaporator of the system. In both cases gravity, inertia and pressure differences combine to actuate the valves, the required one being held closed by refrigerant pressure during operation of the compressor in either direction. The valve mechanism 34 of Fig. 1 might be mounted on the compressor or on its casing and actuated by the sudden applied torque, as explained in connection with Fig. 3, or the valve mechanism of Fig. 3 might be mounted independently of the compressor and operated solely by refrigerant flow, as is the valve 34 of Fig. 1.

Assuming that the compressor of Fig. 3 is incorporated in a sealed unit and rigidly connected with the stator of an electric motor enclosed by the same sealed casing, it will be seen that the direction of the starting jerk applied to the compressor body is now in the opposite direction from that of rotation of the motor rotor, the shaft 62 and the compressor rotor 26, due to the resultant torque being applied through the motor stator to the body of the compressor, with or without transmission of such resultant torque through the casing which encloses the motor and the compressor.

Fig. 4 shows how the principle of actuating a reversing valve by means of inertia can be applied to a conventional piston type compressor to accomplish the result of Fig. 3. Assuming that suitable provisions have been made for lubrication, a reciprocating compressor of the piston type may be driven in either direction of rotation, but the reversal of rotation does not reverse the direction of refrigerant flow. The valve mechanism seen in Fig. 4 is intended to replace the usual cylinder head of an open type reciprocating compressor in which both intake and discharge ports are through the regular valve plate of the compressor. The plan is to design a replacement cylinder head to be bolted on top of the valve plate in place of the original cylinder head which is connected with the suction and discharge tubes, thus converting a conventional rotary or reciprocating compressor into a flow-reversing compressor.

The special cylinder head includes the valve body which is shown in section as 130 in Fig. 4. The chamber 132 connects with the chamber into which high pressure vapor flows from the regular discharge valve of the compressor, and the chamber 132 connects in a similar manner with the chamber from which suction vapor is drawn through the regular intake valve of the compressor. It is therefore only necessary to consider Fig. 4 as a top view of a conventional reciprocating compressor of which the crank shaft is indicated at 136. The actual valve mechanism is very similar to the one shown in Fig. 6 of my copending U.S. patent application Serial No. 50,101, filed Sept. 20, 1948, now Patent No. 2,672,016, but the actuating

mechanism shown in Figs. 7 and 8 of this earlier patent application of mine is omitted. The valve stems 138 and 140 are supported by guides 142 and 144 respectively and are free to slide therein. We thus have a pair of check valves 146 and 148 in the discharge chamber 132, but these check valves are rigidly connected together so that one must open when the other is closed. Likewise we have a pair of check valves 150 and 152 arranged to close one or the other of two ports which lead into the chamber 134 from which suction vapor flows to the regular intake valve of the compressor.

Assume now that the valves are as shown in Fig. 4 or in neutral positions, none being fully closed, that the compressor body is mounted on springs or other flexible supporting means 118, as shown in Fig. 3, and that torque is suddenly applied to the shaft 136 in the direction indicated by the solid arrow. During the first compression stroke of a piston in the compressor the compressor body will jerk suddenly to the right, causing valves 146 and 152 to close. This compression stroke delivers compressed vapor into the chamber 132, thus holding the valve 146 closed and the valve 148 open so that the discharge vapor can flow freely past valve 148 into the chamber 154 and out through the port 156 which now serves as the discharge connection leading to the condenser. The discharge vapor filling the chamber 154 aids in holding the valve 152 closed so that the valve 150 is held open, allowing vapor to be drawn from the port 158 through the chamber 160 and past the valve 150 into the chamber 134, which is connected with the regular intake port of the compressor. So long as the compressor continues to operate in this direction, each of the valves 146 and 152 is held closed by high pressure refrigerant, thereby holding their mating valves 148 and 150 open. The result is to deliver compressed refrigerant vapor to condenser 16, where it condenses and then flows through restrictor 14 to the evaporator 12 from which its vapor flows to port 158. When the compressor is stopped the high and low side pressures may be allowed to equalize or not as desired. They will substantially equalize if a vapor-lock restrictor is used as shown in Fig. 4. Should the next start of the compressor be in its opposite direction of rotation the result will be to close the valves 148 and 150 and to open valves 146 and 152, thus coming back to the position shown in Fig. 4 with flow as indicated by the solid arrows.

As in Figs. 1 and 3 the secondary evaporator 68 of Fig. 4 is cooled only when its condenser 70 is colder than 68, with the result that refrigeration is suspended in the secondary evaporator 68 while the evaporator 12 is being defrosted. This arrangement is suitable for use in a two-temperature household refrigerator. The arrangement of Fig. 4, with the secondary system 68—70 omitted, is also suitable for reverse-cycle air conditioning systems, as the conventional reciprocating compressor is equally efficient in its two directions of rotation and may be operated for long periods in either direction, assuming that proper provision has been made for lubrication.

Again it will be understood that in the event that the compressor of Fig. 4 is enclosed within a sealed unit and rigidly associated with the stator of the motor, as is customary, using either internal or external spring mounting of the sealed unit, the sudden jerk which moves the valves will be caused by resultant torque, hence the valves will operate in exactly the reverse manner. The effect however is the same because high pressure refrigerant is delivered through port 156 when the compressor is started in one direction and it is delivered through port 158 when the compressor is started in the opposite direction. In each case the former discharge tube becomes the suction tube. It is also within the scope of this invention to mount the inertia-actuated valves on the motor which drives the compressor or on the casing of a sealed unit. In any case the valve or valves can be actuated by inertia due to starting of the compressor.

Fig. 5 needs no explanation, being a detail section of Fig. 4 to show that valve guides **142** and **144** are a part of the casting **130**.

Fig. 6 shows a system similar to that of Fig. **1**, but employing a conventional reciprocating compressor **162** of the multiple-effect type, i.e. one having two suction ports for two separate suction pressures. Since this is not a reversible compressor the reversal of flow is obtained by means of a valve mechanism such as shown in my co-pending U.S. patent application Serial Number 45,343, filed August 20, 1948, now Patent No. 2,654,227. Another change from Fig. 1 is that the two pressure reducing devices are located in branch lines instead of in series. Fig. 6 also shows the use of a clock-actuated switch to cause defrosting to occur at a preselected time, preferably between midnight and daybreak.

The switch **164** includes blade **165** which is normally actuated by the clock **166** on a time cycle, but may also be operated manually when occasion requires. The lifting of this switch blade **165** breaks the circuit through wire **173** and the switch **174** regardless of the position of its blade **175** and energizes the circuit through wire **167**, the solenoid **168** and wire **169** to lift the valves of **170** to their defrosting positions, as explained in the earlier application above mentioned. At the same time the switch closes a circuit through the wires **172** and **172'** to short out the thermostatic switch **174** and start the compressor motor **176** if it is not already running. The clock mechanism allows the switch **164** to drop to the position shown at the end of a short period which is established just long enough to insure that the freezer evaporator **12** is defrosted. This allows return of the valves to their normal positions as shown, due to the combined weight of the movable parts of **170** and **168** which is ample to overcome the upward liquid pressure on the valves which were closed during the defrosting operation. As an additional provision to insure the downward movement of the valves due to gravity when the solenoid **168** is de-energized I propose to employ loose fits for lost motion in the pivots which connect these valves with the armature of solenoid **168**.

The valve assembly **34** and **66** of Fig. 1 could be used in Fig. 6 in the same manner, but I have shown the warmer evaporator **20** fed with liquid through the branch tube **178** and expansion valve **180**, thus eliminating the valve assembly **34**. Normal flow of refrigerant is as indicated by solid arrows, the restrictor **14'** being designed to produce a greater pressure drop than the expansion valve **180** so that evaporator **12** operates at below freezing while evaporator **20** operates at a non-frosting temperature or defrosts itself during each idle period.

Closing the defrost switch **164** produces reverse cycle operation of evaporator **12** and condenser **16** without feeding either liquid or vapor to evaporator **20**. Any liquid in evaporator **20** at the time defrosting starts is evaporated therein and the vapor flows through tube **60** to the auxiliary inlet of the compressor as in Fig. 1. There will be substantially no flow of liquid through the expansion valve **180** during the short defrosting operation, though there may be some if the pressure in evaporator **20** is pulled down to below that of the condenser **16** which operates as a low pressure evaporator during the defrosting of evaporator **12**. Dotted arrows indicate the flow during the defrost period. In the event that only one evaporator is required or the secondary system **68—70** is used, the evaporator **20** may be omitted along with expansion valve **180** and tubes **178** and **60**. This arrangement allows the use of any conventional compressor in place of the multiple-suction compressor **176**.

The thermostatic switch **174** may be equipped with two bulbs so as to open when both the space cooled by evaporator **12** and the space cooled by evaporator **20** have been pulled down to their required temperatures. For such use I show two bulbs connected with the thermostat **174** and it is assumed that the thermostat is charged with a volatile fluid in such quantity that the bulb associated with evaporator **20** will contain only vapor at the cut-out point.

The secondary system **68—70** operates as described in connection with Figs. 1, 3 and 4 and may or may not be provided with the thermostatic control valve **182**.

Fig. 7 shows a modified electrical system particularly suited for use in air conditioning or when the reverse-cycle periods are apt to last for hours instead of minutes. The blade **183** of switch **184** is actuated in response to temperature changes of bulb **186**, which is located in the space where temperature is to be controlled. A rise of temperature of bulb **186** moves the blade **183** of switch **184** downwardly, thus energizing the motor **176** through wires **185** to start and **185'** to run without energizing the solenoid **168**. Thus operation of the system is started normally with flow as shown by solid arrows, causing **12** to operate as the evaporator and **16** as the condenser. When the switch blade is moved in the opposite direction, either manually or in response to a drop of temperature of the bulb **186**, the motor **176** and solenoid **168** are both energized through wires **187**, **187'** and **185** so that the flow of refrigerant follows the dotted arrows, causing **12** to function as a condenser and **16** as an evaporator, thus heating the controlled space. It will be seen however that the solenoid **168** remains energized only so long as the starting circuit breaker **188** remains closed with its blade **189** in the solid line position. In this case the armature of the solenoid and the movable parts of the valve assembly **170'** are made light in weight relative to valve port sizes so that the high side pressures effective on the valves by the time the circuit breaker opens will be ample to hold the valves in the positions to which they have been moved by the solenoid. Thus it is only necessary to lift the valves at the start of the run and they will thereafter be held in their lifted positions by high side refrigerant pressure until the next time the compressor is stopped. As some time will always elapse between the need for heating and the need for cooling, the pressures within the system will equalize to allow the valves to drop before the system is restarted.

The wiring of Fig. 7 puts the solenoid **168** in series with the starting winding of the motor **176**, which is permissible with a solenoid winding which offers little resistance to flow of current. The solenoid and the starting circuit could be wired in parallel, as they are in Fig. 6, at a slight additional cost but this is not considered necessary.

The secondary system **68—70** of Figs. 6 and 7 operates as in Figs. 1, 3, 4 and 5, cooling evaporator **68** whenever evaporator **12** is cooled. The secondary system remains idle when the evaporator **12** is functioning as a condenser.

Figure 8 is a bottom view of a preferred design and location of a sealed motor-compressor unit **10**, such as shown in Figs. 1 to 4 inclusive. It will be understood that the lubricating system of Fig. 2 and the valves of Fig. 3 may be modified to fit the vertical axis arrangement of Fig. 8, as by putting the valves and ports **18**, **28** and **24** of Fig. 3 in the end plate **72** of Fig. 2. The casing of unit **10** is flexibly mounted on the rear wall of a refrigerator and provided with parallel fins **192** instead of the usual radial fins. This is to reduce the horizontal dimension between the back of the refrigerator **194** and the wall **195** of the room. It is highly desirable to locate the motor-compressor unit in this manner and to keep one of the horizontal dimensions down to the minimum which allows use of a suitable motor. One of the reasons for preferring a two-pole motor, is to reduce its diameter for use in this location. This arrangement puts the fins in vertical planes, thus adapting the unit for cooling by gravity circulation of air upwardly over it.

The tube **50**, which normally carries suction vapor, is preferably connected as at **60** of Fig. 2, thus the motor is normally cooled by suction vapor and high pressure vapor flows through the casing during the short defrost period only. A port not shown connects the chamber **112** (Fig. 3) with the interior of the motor casing of Fig. 8. I propose to locate the unit **10** or a portion of the condenser near the bottom of the cabinet to provide heat for evaporating drip water to ambient air.

The electrical system of Fig. 8 may be similar to that of Fig. 6, except that the switch **164** will also reverse the motor, normally closing the motor circuit through the thermostatic switch **174** while during defrost it opens this circuit and closes the one which reverses motor rotation. In the event that defrosting is to be controlled in response to frost accumulation, door openings, a temperature change, manually or otherwise than on a time cycle the clock may be omitted.

In the event that the system of Fig. 1 is to be used in Fig. 8 the tubes **102** and **110** will disappear and we will see tubes **30** and **50** enter the near (bottom) end of **10** while tube **60** comes out the back of the cabinet and disappears on the far (top) end of **10**, where it enters the casing, as in Fig. 2. This eliminates the valves of Fig. 3 and substitutes the valves of Fig. 1, which are preferably located inside of or adjacent to their respective refrigerated spaces within the refrigerator cabinet.

The word "inertia" is used in this specification and the appended claims as defined in Webster's Dictionary thus: "That property of matter by which it tends when at rest to remain so, and when in motion to continue in motion, and in the same straight line or direction, unless acted on by some external force."

It will be understood that a change of design from one in which the compressor body is rigidly connected with the stator of the motor, and the "jerk" is caused by resultant torque, to one in which the compressor body is separate from the motor stator and the starting "jerk" is caused by frictional drag, or vice versa, will not change the fact of the valves of Fig. 3 or 4 being actuated by their inertia. It would however change the marking of switch **96** as used in either Fig. 3 or Fig. 4 and in Fig. 3 it would call for crossing the tubes **102** and **110** to connect with **114** and **112** respectively. Instead of L (left) and R (right) the switch might be marked C (cooling) and D (defrosting) to match the detailed design.

The showing of two bulbs for switch **174** in Fig. 6 will be understood by reference to Fig. 10 of my issued U.S. Patent No. 2,349,367 or to the multiple bulbs of switch 83 in Fig. 13 of my U. S. Patent No. 2,359,780.

The modifications here shown represent only a few typical designs which utilize the principles of this invention, the main features of which are obtainable in designs having many other modifications of mechanical details.

I claim:

1. In a refrigerating system employing a volatile refrigerant, a compressor, means for starting said compressor in either direction of rotation, a valve for controlling flow of said refrigerant, a seat for said valve, and means associated with said valve for moving it relative to its seat, said means comprising a mass which is movable relative to the main body of said compressor and thus caused by its own inertia to move the valve in one direction relative to its seat when the compressor is started in one direction of rotation and in the opposite direction when the compressor is started in the reverse direction of rotation.

2. In a refrigerating system, a compressor, an intake port for said compressor, means forming two passages leading to said intake port, valve means for stopping flow to said port from said passages one at a time, and means responsive to the flow of refrigerant vapor from said compressor for actuating said valve means.

3. In a refrigerating system, a compressor, said compressor being reversible as to direction of its pumping action, an intake port for said compressor, means forming two passages leading to said intake port, valve means for stopping flow to said port from said passages one at a time, and means actuated by flow of refrigerant vapor from said compressor for actuating said valve means to stop flow from one of said passages for one direction of said pumping action and to stop flow from the other of said passages for the other direction of said pumping action.

4. In a refrigerating system, a compressor, a control device for reversing the direction of rotation of said compressor to reverse its direction of pumping action, a valve, a seat for said valve, and means for utilizing inertia to cause said valve to move relative to its seat at the instant the compressor is started.

5. In a refrigerating system, a motor-compressor unit including flexible supporting means, means for starting said compressor, a valve and a seat therefor associated with the said compressor, and inertia means for actuating said valve to move it relative to its seat as a result of torque reaction on the body of said unit caused by the starting of said compressor.

6. In a refrigerating system of reversible heat pump type, a compressor, a motor for driving said compressor in either direction of rotation, an inlet port of said compressor which serves as inlet port regardless of the direction of compressor rotation, a pair of passages leading to said inlet port, valve means for closing one at a time of said passages while opening the other, a mass associated with said valve means, means utilizing the inertia of said mass at the moment of starting said compressor in either direction to actuate said valve means in accordance with the starting direction of rotation, and means for employing the discharge pressure of said compressor to hold said valve means in the position which maintains the closure of one of said passages during continuous operation of the compressor in the direction in which it was started.

7. In a refrigerating system, a compressor, means for starting said compressor, a refrigerant passage, a valve adapted to close said passage, and means mounting said valve so that it movably responds to dual influences, one of said influences being refrigerant flow and the other of said influences being an inertia resultant of torque suddenly applied to said compressor.

8. In a refrigerating system, a rotary compressor having two inlet ports for low pressure vapor and a discharge port, a suction conduit having a branch leading to each of said inlet ports, means for reversing said compressor to cause said discharge port to become an inlet port and one of the first said inlet ports to become a discharge port, and a check valve in the branch of said suction conduit which leads to the other one of the first inlet ports, whereby said branch is closed to flow of discharge vapor to said other one of the first said inlet ports when said compressor is reversed.

9. In a refrigerating system employing a volatile refrigerant, a pair of heat exchangers of which one acts as a condenser and the other acts as an evaporator, a compressor for supplying refrigerant vapor to the heat exchanger acting as a condenser and drawing vapor from the one acting as an evaporator, control means for stopping said compressor and restarting it in either direction of rotation, a valve for changing the path of refrigerant flow when said compressor is restarted in the opposite direction of rotation from that in which it was last operated, said changing of the path causing the first said one of the heat exchangers to act as an evaporator and said other to act as a condenser, and inertia means responsive to the starting of said compressor only during acceleration thereof for actuating said valve.

10. In a refrigerating system, a reversible compressor, and inertia-actuated valve means for establishing the direction of refrigerant flow in said system in accordance with compressor rotation each time the compressor is started or reversed in its direction of rotation.

11. In a refrigerating system, a compressor, reversible driving means for said compressor, and a valve closed by an inertia force which is effective only while said compressor is being accelerated, said valve being thereafter held closed by the discharge pressure produced by said compressor during its operation.

12. In a refrigerating system employing a volatile refrigerant, two evaporators, a condenser, two pressure reducing devices including one for each of said evaporators, a first one of said devices being of a type adapted to control refrigerant flow in either direction, the other of said devices being pressure-responsive and adapted to regulate the flow of refrigerant in one direction only, and means for reversing the flow of refrigerant in a portion of said system whereby one of said evaporators is caused to serve as a condenser for the purpose of defrosting it while the first said condenser serves to evaporate liquid refrigerant which it receives from the defrosting evaporator now serving as a condenser by way of said first one of the pressure reducing devices, the other of said evaporators meanwhile having its inlet stopped.

13. A fluid flow control valve for use in a refrigeration system of the type having a reversible compressor and means for alternatively driving said compressor in its forward and its reverse directions, said valve comprising a body member for mounting on an element of the system that moves in one direction in response to starting of the compressor forwardly and moves in a different direction in response to starting of the compressor reversely, and a valve member movable relative to said body member by inertia in response to movement of the element caused by starting of the compressor in a predetermined direction.

14. A fluid flow control valve for use in a refrigeration system of the type having a reversible compressor and means for selectively driving said compressor in one or the other of two directions, said valve comprising a body member for mounting on an element of the system that moves in one direction in response to starting of the compressor forwardly and moves in a different direction in response to starting of the compressor reversely, and a

valve member movable relative to said body member by inertia in response to movement of the element caused by starting of the compressor in a predetermined direction, said valve member being arranged when it is moved by inertia to one position to be held in said one position by fluid pressure within said valve body.

15. A fluid flow control valve for use in a refrigeration system of the type having a reversible compressor and means for selectively driving said compressor in a first or a reverse direction, the compressor having an outlet port and an inlet port, and the direction of fluid flow through said outlet and inlet ports being constant regardless of the direction of drive of the compressor, said valve comprising a body member for mounting on an element of the system that moves initially in one direction in response to forward starting of the compressor and moves initially in a direction different from said one direction in response to reverse starting of the compressor, said body member defining a chamber and a first port communicating therewith for connection to one of the ports of the compressor, said body also having a second port and a third port communicating with said chamber, and a reciprocatable valve member for alternatively closing one of said second and third ports in response to a movement of the element on which said body member is mounted caused by starting of the compressor, the inertia of said valve member being effective to shift it relative to said body member in response to movement of the element caused by starting the compressor in a direction opposite from the direction in which it was last operated.

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UNITED STATES PATENT OFFICE

CERTIFICATION OF CORRECTION

Patent No. 2,976,698 March 28, 1961

Glenn Muffly

It is hereby certified that error appears in the above numbered patent requiring correction and that the said Letters Patent should read as corrected below.

Column 4, line 43, for "reserve" read -- reverse --; column 9, line 42, for "assembly" read -- asemblies --.

Signed and sealed this 8th day of August 1961.

(SEAL)
Attest:

ERNEST W. SWIDER
Attesting Officer

DAVID L. LADD
Commissioner of Patents