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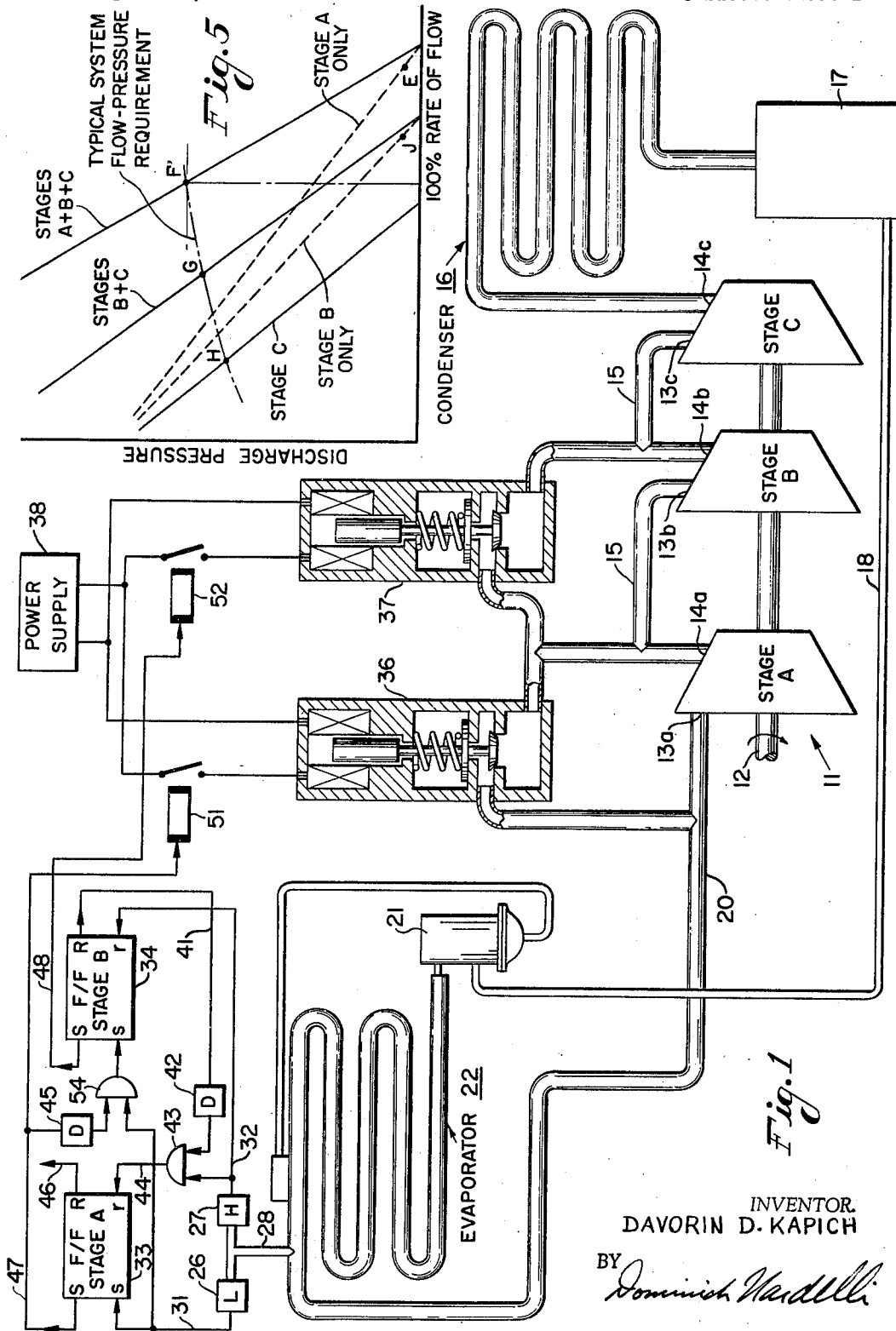
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3,495,418

REFRIGERATION SYSTEM WITH COMPRESSOR UNLOADING MEANS

Filed April 18, 1968

3 Sheets-Sheet 1



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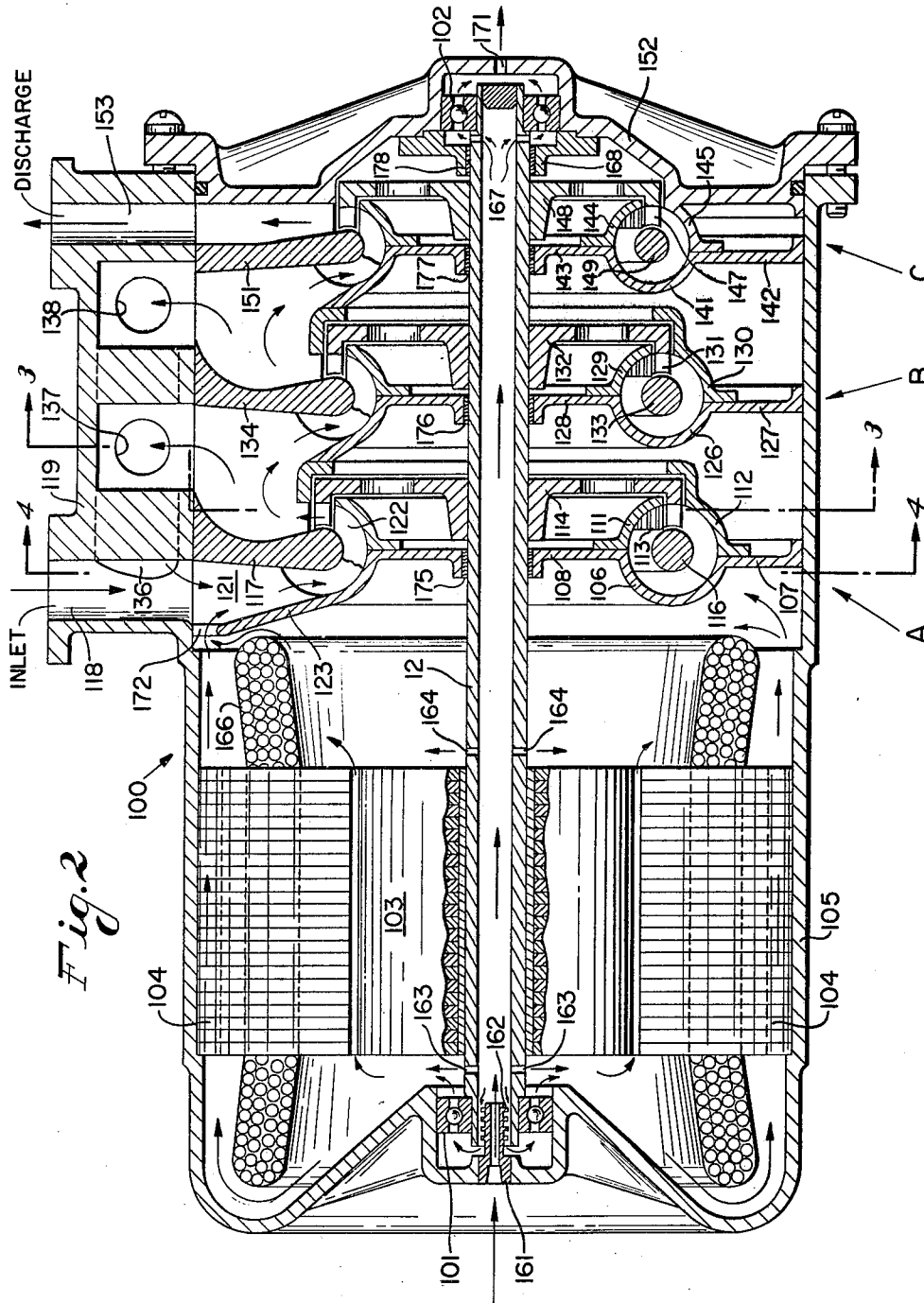


Fig. 2

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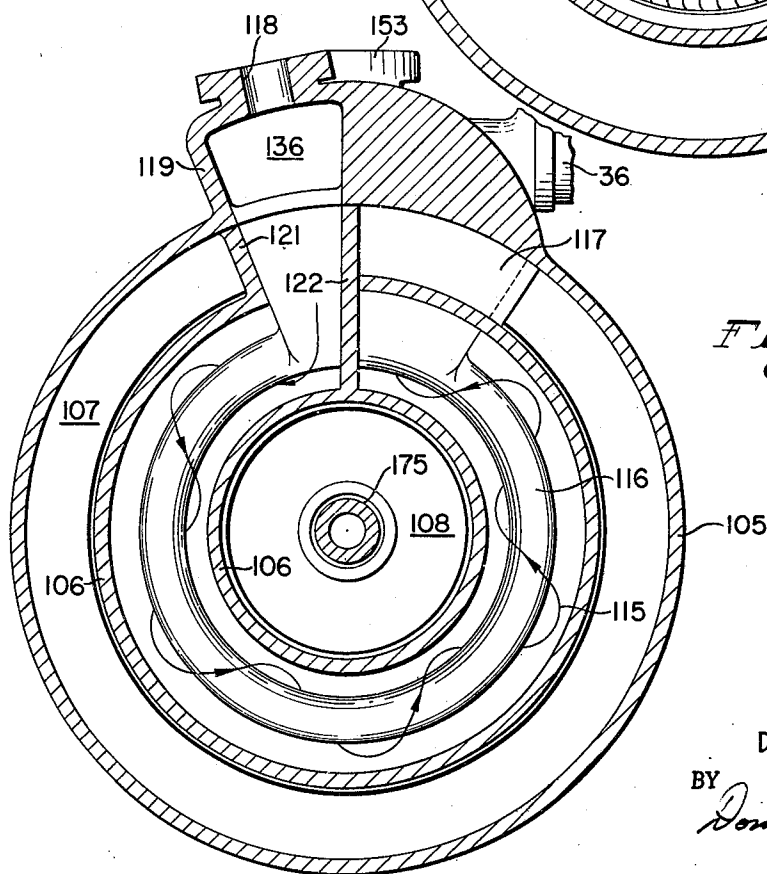
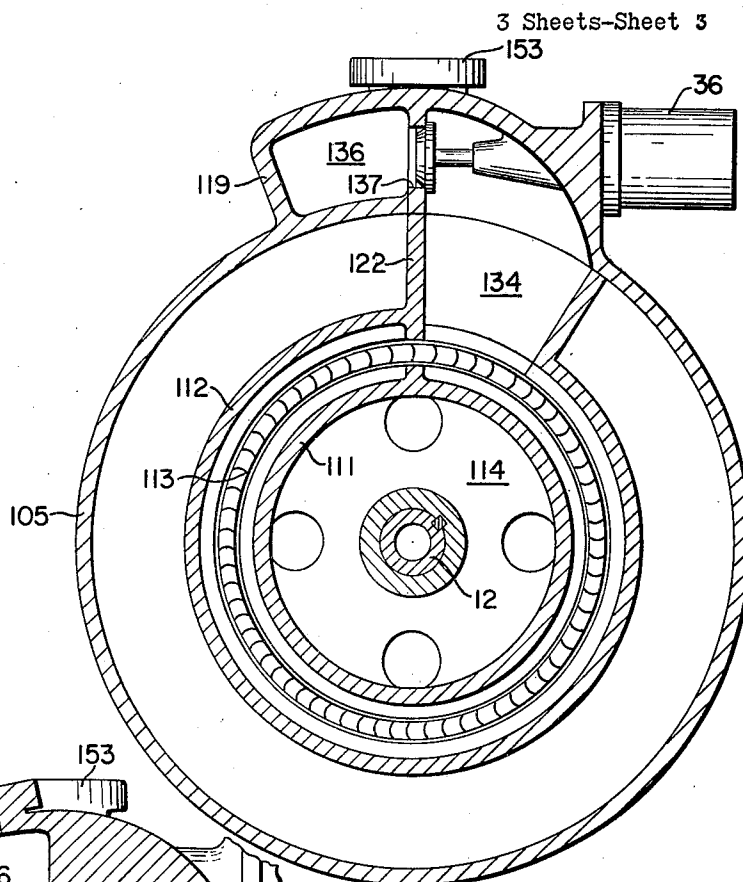


Fig. 4

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REFRIGERATION SYSTEM WITH COMPRESSOR UNLOADING MEANS

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5 Claims

ABSTRACT OF THE DISCLOSURE

A refrigeration system having a constant speed centrifugal compressor with two or more stages arranged in series and means for unloading and reloading the stages one at a time as the cooling requirements decrease and increase respectively.

Centrifugal compressors have the inherent characteristic that at a fixed speed the discharge pressure increases when the flow requirements decrease. Therefore, when centrifugal compressors are used for refrigeration purposes, means are needed to lower the compressor capacity when the load requirement is low. Decreasing the compressor capacity keeps the output pressure within reasonable limits and also prevents liquid refrigerant from "slugging" the compressor, i.e., prevents liquid from entering the inlet of the compressor. Obviously, continuous stopping and starting of, for example, a five horsepower motor operating at high speed is not practical. In the past, the capacity of a high speed centrifugal compressor had been decreased by bleeding some of the hot, high pressure gas discharging from the compressor into the intake of the compressor where the gas vaporizes any liquid in the intake to insure that the compressor is filled with a gas. However, this method is inefficient because a large portion of flow is being throttled from discharge to inlet.

An object of this invention is to provide a relatively low speed centrifugal compressor operating at constant speed wherein the refrigerant is compressed in steps as it moves from one stage to the next. Then, as the cooling requirements drop, means are provided for unloading the stages one at a time so that the intake pressure remains within given limits, achieving at the same time reduction in refrigerant flow delivered to the condenser.

Another object of this invention is to provide an improved relatively low speed, centrifugal compressor that compresses gases to a relatively high pressure wherein "liquid slugging" does not deteriorate the impeller blades.

Briefly, the invention comprises a constant speed centrifugal compressor having two or more stages in series of the type taught in United States Patent No. 3,292,899. The gas is first compressed within one stage, then coupled to the next stage to be further compressed, etc., until the gas is at the desired condensing pressure. In a condenser, the vapor is cooled to a liquid which is evaporated in a suitable evaporator to absorb heat. Valve means are provided between each stage to allow the refrigerant to flow unrestricted between discharge and inlet of selected stages when required. Control means are provided to operate and control the valve means in response to the cooling demands of the system so that when the cooling demand is at the lowest requirement only one stage is used to compress the gas and when the cooling demand is at the highest requirement all the stages are used to compress the gas. The compressor inherently operates at relatively low speeds so that, if slugs of liquid enter the first stage, the compressor is not damaged.

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These and other objects of this invention become apparent as the following description is read in conjunction with the drawings in which:

FIG. 1 is a schematic diagram showing the refrigeration cycle of the invention.

FIG. 2 is an axial cross-section of a relatively low speed compressor that incorporates the refrigeration cycle shown in FIG. 1;

FIG. 3 is a section taken substantially on line 3—3 in FIG. 1 and viewed in the direction of the arrows;

FIG. 4 is a section taken substantially on line 4—4 in FIG. 1 and also viewed in the direction of the arrows; and

FIG. 5 is a graph of the discharge pressure versus refrigerant volumetric flow rate based on the inlet vapor density of the compressor shown in FIG. 1.

Referring to the drawings and to FIG. 1, in particular, there is shown a schematic refrigeration system incorporating the novel refrigeration cycle. Shown schematically is a centrifugal compressor 11 having three stages, stage A, stage B and stage C. The compressor 11 is powered by a suitable constant speed motor (not shown) which rotates a shaft 12 that is coupled to the three stages. The stages A, B and C have suitable inlets 13a, 13b and 13c, and outlets 14a, 14b and 14c, respectively, which are connected by suitable tubing 15 through which the refrigerant flows. Outlet 14c is coupled to a condenser 16 wherein the refrigerant is liquefied and stored in a reservoir 17. The liquid leaves the reservoir through tubing 18 and expands through a conventional refrigerant expansion valve 21. The refrigerant enters an evaporator 22 and vaporizes. The vapor or gas is ducted to compressor 11 through a tubing 20.

Since shaft 12 is continuously rotated by the motor at a constant speed, this invention includes a means that is independent of the rotational speed for controlling the cooling capacity of the system in relation to the cooling requirements. The means utilizes the fact that in a standard compression-expansion refrigerant system, the pressure of the refrigerant leaving the evaporator 22 is directly related to temperature. For example, the means includes two pressure transducers 26 and 27 which are coupled to the evaporator 22 through a T-branch tubing 28. The function of transducer 26 is to apply a signal, for example, a positive voltage to a lead 31 whenever the pressure in the evaporator 22 is below a set minimum value. The function of transducer 27 is to apply a signal, for example, a positive voltage to a lead 32 whenever the pressure in the evaporator 22 is above a set maximum value. The signals outputted by transducers 26 and 27 are fed through suitable gates to a stage A flip-flop 33 and a stage B flip-flop 34, which will be described more fully hereinafter. In addition, the means includes two solenoid valves 36 and 37 wherein the valve 36 is connected across stage A and valve 37 is connected across stage B through suitable tubing. Valves 36 and 37 are two normally closed solenoid valves. When both valves 36 and 37 are closed, all the stages A, B and C are in series, i.e., the refrigerant is compressed to successively higher pressure in each stage.

Whenever only valve 36 is energized by a power supply 38, valve 36 opens, and only stages B and C contribute effectively to the compression process. Stage A is more or less "windmilling" the refrigerant though without significant pressure rise. The operating point of the stage A is as shown on FIG. 5 as point E. Operating point of stages B+C combined is shifted from point F (when all three stages A+B+C were operating in series) to a point G on the B+C operating curve. Thus, the opening of the valve 36 has resulted in a reduction in the net flow delivered to the condenser from point F

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to point G. The amount of flow bypassed through the valve 36 from stage A discharge to the stage A inlet is then equal to the difference between the flow at point E, minus the flow at point G. Since the refrigerant vapor density at the compressor inlet is not changing appreciably, the volumetric flow represents at the same time the weight flow rate which is directly proportional to the cooling capacity of the system.

Whenever both valves 36 and 37 are energized, they both open and only stage C compresses the refrigerant. Stages A and B are both "windmilling" the refrigerant though without significant pressure rise. The operating point of the stage C is now shown on FIG. 5 as point H, thus reducing the net flow delivered to the condenser from point G (when B+C stages were operating in series) to the flow indicated by the point H. The amount of flow bypassed through the valve 37 is now equal to the difference in flow indicated by a point J minus the flow indicated by a point H. Flow bypassed through the valve 36 equals the difference in flow between points E and H. Stage A is now operating at the point E, and the stage B is operating at point J. In this embodiment valve 37 is preferably closed while valve 36 is closed, and is open if valve 36 is open. However, valve 36 could be open while valve 37 is closed as will be explained hereinafter.

The refrigeration system operates as follows: Initially, lead 32 has a high positive voltage coupled thereto by transducer 27 and lead 31 is at ground potential since the complete system is at ambient temperature and the pressure in the evaporator is relatively high. Lead 32 couples the positive voltage to the reset input *r* of the stage B flip-flop 34 to reset the flip-flop. When the flip-flop 34 is reset, the reset output R applies a positive voltage to lead 41. The positive signal in lead 41 is fed through a suitable delay circuit 42 (for reasons that will become apparent hereinafter) and into one of the inputs of an AND gate 43. The other input of AND gate 43 has coupled thereto the positive signal on lead 32. Since the AND gate 43 is of the type that only outputs a positive voltage when both inputs have positive voltage coupled thereto, its output lead 44 couples a positive voltage to the reset input *r* of stage A flip-flop 33 to reset the flip-flop so that its reset output R couples a positive voltage on a lead 46. Flip-flops 33 and 34 are of the type that switch states at the leading edge of a positive voltage and remain in that state until the leading edge of a rising voltage is coupled to the other input. When both flip-flops 33 and 34 are reset, their respective set outputs S are at ground potential thereby placing leads 47 and 48 at ground potential. Leads 47 and 48 are coupled to suitable relay switches 51 and 52, respectively, which are in series with a power supply 38 and the respective valves 36 and 37. The relay switches 51 and 52 operate so that they close the respective circuits only when the respective leads 47 and 48 have a positive voltage coupled thereto.

As the rate of refrigerant liquefied in the container 17 and expanded in evaporator 22 diminishes, the temperature and the pressure of the refrigerant in the evaporator 22 and tubing 20 decreases because of the compressor tendency to extract constant amount of flow from the evaporator. When the pressure drops below the pressure set in transducer 27, the positive potential is removed from lead 32, and both leads 31 and 32 are at ground potential. If the temperature cannot be controlled by expansion valve 21, the temperature of and the pressure in the evaporator drops to the value whereby transducer 26 couples a positive voltage to lead 31. Lead 31 is coupled to the set inputs of stage A flip-flop 33 and also to an AND gate 54 whose output is coupled to the set inputs of flip-flop 34. The flip-flop 33 switches to the set state causing the set output S to couple a positive voltage to the lead 47. A delay circuit 45 prevents the positive potential on the lead 47 from being coupled immediately to AND gate 54. The positive voltage on lead 47 causes solenoid 51 to open valve 36. Now, the net refrigerant

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flow delivered by the compressor decreases from point F to point G (FIG. 5) as previously described. Since less flow is now being taken from the evaporator, the pressure in tubing 20 rises above the set pressure in transducer 26 and the potential on lead 31 drops to ground. The function of delay circuit 45 is to prevent the positive potential on lead 31 from being coupled to AND gate 54 until the pressure rises in the evaporator due to one stage being unloaded. Flip-flop 34 remains in the reset state.

In this condition with one stage unloaded the refrigeration system could operate in one of three ways. First, if the refrigeration requirements stay substantially the same, the pressure in tubing 20 remains substantially constant and the expansion valve 21 controls the flow of refrigerant. Second, if the refrigeration requirement increases, the pressure in tubing 20 rises indicating that the evaporator temperature is rising. When the pressure rises sufficiently, transducer 27 places a positive potential on lead 32. Since the lead 41 has already had a positive potential coupled thereto, delay 42 has coupled the positive voltage to one lead of AND gate 43 therefore gate 43 passes a positive potential to the reset input *r* of flip-flop 33 and solenoid 51 is de-energized. Stage A is again loaded in the system and, since the refrigerant is compressed sequentially in all three stages (shown as point F on FIG. 5), the pressure in tubing 20 drops. Third or last, if the refrigeration requirement further decreases when valve 36 is open, transducer 26 again places a positive potential on lead 31 because the pressure in the evaporator drops. With flip-flop 33 being set for some time, the delay 45 coupled to lead 47 has coupled the positive signal to one of the input leads of AND gate 54. On the rising leading edge of the rising voltage in lead 31, stage B flip-flop 34 is switched to the set stage and the set output S couples to lead 48 a positive potential. Solenoid 52 is also energized and valve 37 is opened along with valve 36. The refrigerant flow now decreases to point H (FIG. 5) for reasons previously described.

When the refrigeration system is in this low output condition and if the refrigeration requirement increases, the pressure in tubing 20 rises. When the pressure is equal to the pressure set in transducer 27, a positive potential is coupled to lead 32. The flip-flop 34 is reset immediately, but, since delay 42 is placed in series with flip-flop 34 and AND gate 43, flip-flop 33 is not reset. At this point, valve 37 is closed and valve 36 is open. The delay time of delay 42 is sufficiently long to allow the compressor to drop the pressure in tubing 20 to cause the positive potential on lead 32 to be removed. If the cooling requirement is still relatively large, transducer 27 will again place a positive potential on lead 32 to cause flip-flop 33 also to be switched to the reset state. At this point both valves 36 and 37 are closed and the refrigerant is now compressed in all the three stages to provide maximum refrigeration.

Referring to FIG. 2, there is shown a compressor 100 employing the teachings of this invention, and also the teachings disclosed in the above-mentioned United States Patent No. 3,292,899. The compressor 100 includes the three stages A, B and C powered by the shaft 12. The shaft 12 is preferably hollow for reasons that will be described hereinafter and is bearing mounted at its end by bearings 101 and 102. The left side of the shaft 12 is coupled to a suitable electric motor. For example, a cage 103 is fixed on the shaft and is surrounded by suitable magnetic field poles 104 to form a standard constant speed A.C. (alternating current) induction motor. Field poles 104 are mounted within a housing 105.

The three compressor stages A, B and C are mounted on the right side and include the following components, for example, the toroidal compartment for stage A includes a circular half member 106 having an outwardly extending flange 107 that is suitably fixed to the housing 105. Half member 106 has also an inwardly extending

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flange 108 to support a suitable seal 175 between it and the shaft 12. The toroidal compartment also is made of an inner circular quarter-member 111 and an outer circular quarter-member 112 which are both suitably fixed to flanges 108 and 107, respectively, forming a circular opening or torus. Within the opening extends a plurality of impeller blades 113 that protrude axially from a wheel 114 that is, in turn, keyed to shaft 12 so that the wheel 114 and impeller blades 113 rotate with the shaft. Disposed within the toroidal compartment is a ring 116 that is held in place by an outwardly extending lug 117 more clearly shown in FIG. 4. In the region of lug 117 suitable passageways are provided for the refrigerant to ingress and egress from the stage. The refrigerant enters through a port 118 formed in an appendage 119 to the housing 105. The refrigerant is guided into the toroidal compartment by suitable baffles, for example, radial baffles 121 and 122 (FIG. 4) that are axially disposed, the lug 117, and inclined baffle 123 (FIG. 2) that is tangentially disposed to half member 106.

The impeller blades 113 and wheel 114 are rotating, for example, counter-clockwise as viewed in FIG. 3. Therefore, the refrigerant flows in a helical path that encircles the ring 116 and also the axis of the compressor, as illustrated by a line 115 (FIG. 4) with arrowheads. The refrigerant flows and is compressed according to the principles taught in U.S. Patent No. 3,292,899. The refrigerant leaves stage A through a suitable passageway or diffuser formed on the right side of lug 117 (as viewed in FIG. 2) and enters stage B. Stage B is made similar to stage A with a half member 126, outer flange 127, inner flange 128, inner quarter member 129, and outer quarter member 130 forming the toroidal passageway. Stage B also includes a plurality of impeller blades 131 mounted on a wheel 132 also keyed to shaft 12. A ring 133 is also provided in stage B and held in place by a lug 134. Stage B propels the refrigerant in the same manner that stage A propels it to further compress the refrigerant. The refrigerant egresses from stage B through a suitable passageway or diffuser formed on the right side of lug 134 (as viewed in FIG. 2) and enters stage C. Stage C also is made similar to stage A and includes a half member 141, outer flange 142, inner flange 143, inner quarter member 144 and outer quarter member 145 forming the toroidal passageway. There is provided a plurality of impeller blades 147 mounted on a wheel 148 also keyed to shaft 12. A ring 149 is provided in the toroidal passageway of stage C and held in place by a lug 151. Outer quarter member 145 is attached to an end cover 152 which supports the bearing 102 for the shaft. The refrigerant is further compressed in stage C and then egresses through a passageway 153 to the condenser 16 since the refrigerant in one stage is at a higher pressure than the refrigerant in a preceding stage, suitable seals 175, 176, 177 and 178 are provided around the shaft 12 to contain the refrigerant within the compressor 100 and within the stages. The loading and unloading means for the stages of the compressor is provided within appendage 119. As shown in FIGS. 3 and 4, appendage 119 has an axially disposed passageway 136 with ports 137 and 138 (FIG. 2) disposed transversely. Passageway 136 communicates with inlet port 118 and communicates with stage B through port 137. The solenoid valve 36 of FIG. 1 is used to open and close port 137 as shown in FIG. 3. As in the schematic system shown in FIG. 1, when valve 36 opens port 137, the refrigerant flows relatively unrestricted from discharge to inlet of the stage A. The valve 37 similarly couples and uncouples discharge of the stage B with the compressor inlet 118 via passageway 136.

The compressor embodiment shown in FIGS. 2, 3 and 4 is a preferred embodiment because, as shown in FIG. 5, the pressure at the discharge end varies at a relatively larger rate than and inversely to refrigerant flow rate. When both valves 36 and 37 are open the performance of the compressor is illustrated by the curve marked stage

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C. When valve 36 is the only valve open, the performance of the compressor is illustrated by the curve marked stages B+C. When all valves are closed, the performance is illustrated by the curve marked stages A+B+C. As mentioned, the curves are relatively steep, and therefore, when the pressure becomes too large for the rate of flow, the compressor is made to operate on the adjacent curve and still maintain a pressure head that is sufficient to liquefy the refrigerant. If the system is only operating with stage C compressing the refrigerant, and the temperature of the evaporator drops to a relatively low temperature, liquid slugging would not degrade the compressors since the flow of refrigerant is relatively low and the speed of the impeller blades is relatively low.

The compressor 100, shown in FIG. 2, also includes a feature for lubricating and cooling. This feature is provided for by liquid refrigerant coupled to a nozzle 161 that is fixed to the left end of housing 105 and extends into the shaft 12. The nozzle 161 is provided with labyrinths 162 around which the refrigerant expands, providing liquid-vapor mixture to cool and lubricate the bearing 101. The refrigerant also expands out of radial holes 163 and 164 formed in the shaft to spray cold refrigerant on the field coils 166. At the other end of shaft 12, bearing 102 is cooled and lubricated by refrigerant expanding from radial holes 167 also formed in shaft 12. The refrigerant that flows over the bearing 102 passes through a port 171 which is coupled to a suitable tube, not shown, and fed into the intake 118. The refrigerant that cools bearing 101 and the motor enter stage A through a port 172 formed in member 123. Port 172 could be placed in such a position that gravity would cause any accumulated liquid to flow into inlet 118.

Since many changes could be made in the above construction and many apparently widely different embodiments of this invention could be made without departing from the scope thereof, all matter contained in the above description or shown in the accompanying drawings is intended to be interpreted as illustrative and not in a limiting sense.

What is claimed is:

1. A refrigeration system comprising:

- a compressor having a plurality of stages for compressing a refrigerant in each stage to a higher pressure;
- a condenser for liquefying the refrigerant after it is compressed by said compressor;
- an evaporator for vaporizing the liquid into a vapor to be again compressed by said compressor;
- means for sensing the refrigeration temperature of said system;
- means, responsive to said means for sensing, for unloading the stages one at a time as the refrigeration requirements drop and for reloading the stages one at a time as the refrigeration requirements increase, said compressor being a centrifugal type and each of said stages operating at a steady rotational speed,
- said means for unloading including a by-pass duct for at least one stage and a valve operated by said means for sensing,
- at least one of said stages including
 - a vaneless toroidal housing through which the refrigerant passes to absorb energy;
 - a plurality of impeller blades extending into said housing and disposed to move with respect to said housing;
 - a wheel, on which said impeller blades are mounted, said wheel being disposed coaxially with said toroidal housing;
 - a shaft for carrying said wheel in rotation,
 - a casing for enclosing said stages,
 - an electric motor disposed within said casing for rotating said wheel,
 - said shaft being hollow,
 - means for feeding liquid refrigerant within said rotating shaft, and

port means on said shaft for causing the liquid to leave said shaft and expand within said casing to provide cooling for the components disposed within said casing.

2. A refrigeration system comprising:

- a compressor having a plurality of stages for compressing a refrigerant in each stage to a higher pressure;
- a condenser for liquefying the refrigerant after it is compressed by said compressor;
- an evaporator for vaporizing the liquid into a vapor to be again compressed by said compressor;
- means for sensing the refrigeration temperature of said system;
- means, responsive to said means for sensing, for unloading the stages one at a time as the refrigeration requirements drop and for reloading the stages one at a time as the refrigeration requirements increase, at least one of said stages including—
- a vaneless toroidal housing through which the refrigerant passes to absorb energy;
- a plurality of impeller blades extending into said housing and disposed to move with respect to said housing;
- a wheel on which said impeller blades are mounted and said wheel is disposed coaxially with said toroidal housing; and
- a shaft for carrying said wheel in rotation.

3. The system of claim 2 wherein:

- a casing is provided for enclosing said stages,
- an induction electric motor is disposed within said casing for rotating said shaft,
- said shaft is hollow,
- means are provided for feeding liquid refrigerant within said rotating shaft,
- port means on said shaft for causing the liquid to leave said shaft and expand within said casing to provide cooling for the components disposed within said casing.

4. A refrigeration system comprising:

- a compressor having an inlet and a first outlet whereby said refrigerant normally enters said inlet and exits from said outlet at a higher pressure;
- a condenser;
- an evaporator;
- first means for sensing the refrigeration temperature of said evaporator;

- said compressor having at least another outlet between said inlet and said first outlet;
- second means for connecting said other outlet with said inlet;
- third means disposed in said second means so that said other outlet communicates with said inlet in response to said first means whenever said refrigeration temperature tends to drop,
- said compressor including—
- a vaneless toroidal housing;
- a plurality of impeller blades extending into said housing and disposed to move with respect to said housing;
- a wheel on which said blades are mounted and which wheel is disposed coaxially with said toroidal housing;
- a shaft for carrying said wheel in rotation,
- a casing for enclosing said stages,
- an electric motor disposed within said casing for rotating said wheel,
- said shaft being hollow,
- means for feeding liquid refrigerant within said rotating shaft, and
- port means on said shaft for causing the liquid to leave said shaft and expand within said casing to provide cooling for the components disposed within said casing.

5. The system of claim 4 wherein:

- said second means includes a duct connecting said other outlet with said inlet; and
- said third means includes a valve.

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45 62—228, 510; 230—26