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AUTOMATIC REFRIGERATION SYSTEM

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Fig. 1.

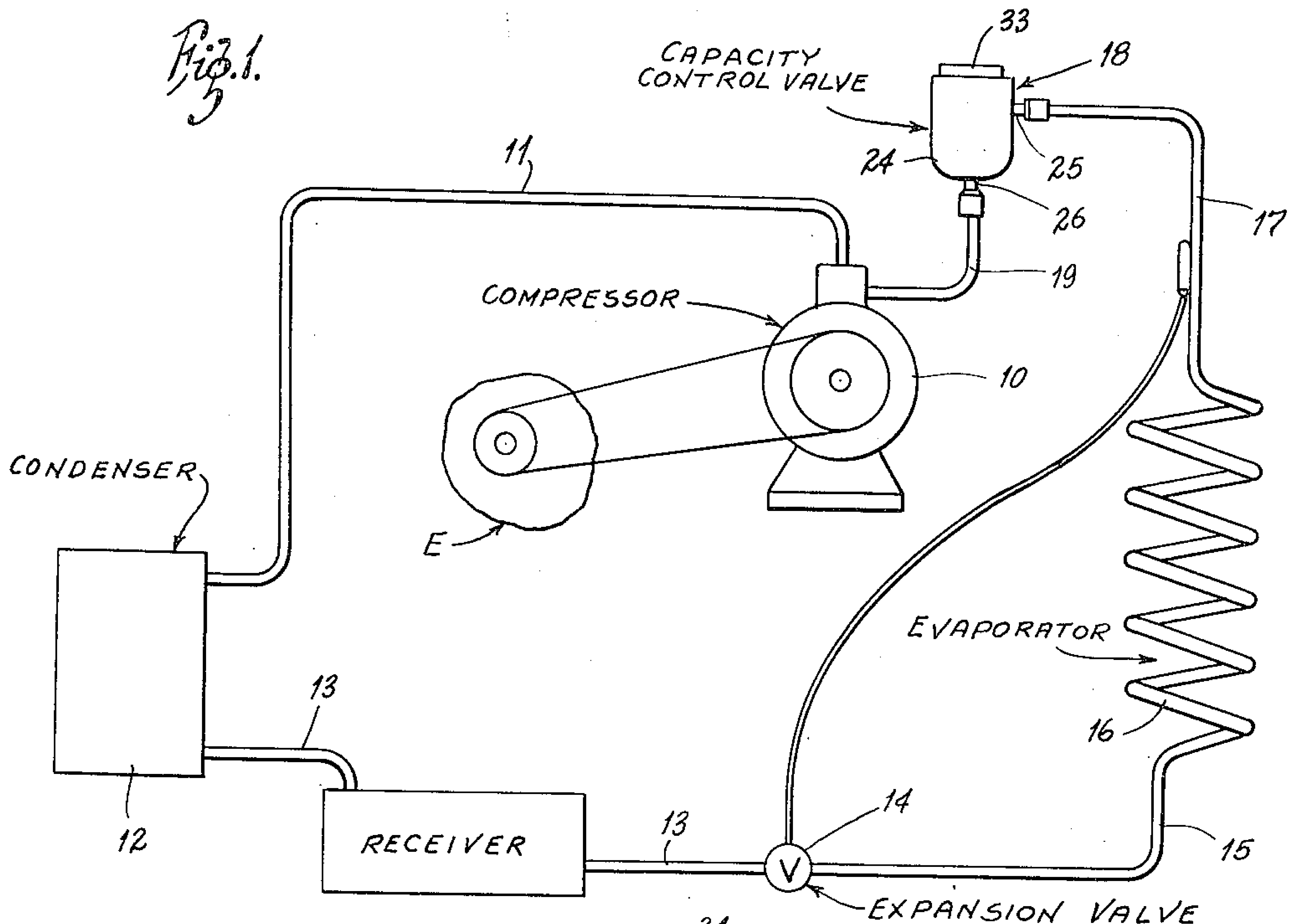
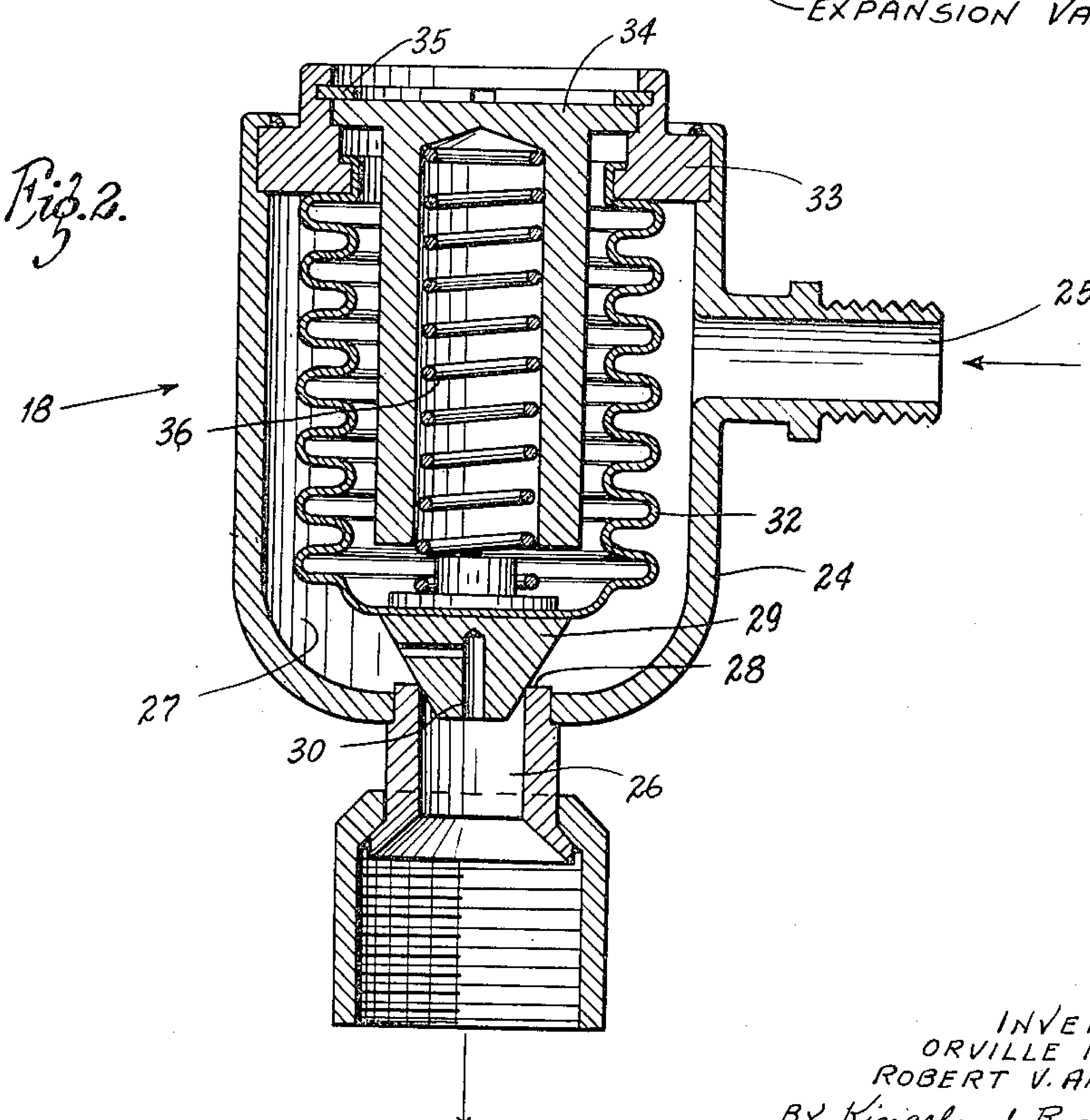


Fig. 2.



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AUTOMATIC REFRIGERATION SYSTEM

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9 Claims. (Cl. 62—3)

The present invention relates to an automatic refrigeration system, and especially to one which is subject to certain variable conditions. More particularly, it relates to a refrigeration system wherein the compressor is subject to wide variation in speed of operation, and consequently in the volume of refrigerant that it circulates, and the head pressures that it produces.

The conditions the present system is designed to overcome are typically those found in automobile air conditioning systems wherein there is a compressor motor that is driven directly from some part of the engine of the automobile. In such systems, the compressor speed varies with the speed of the car, which may have no relationship to the amount of refrigeration required.

Since the refrigeration system must operate under all normal driving conditions, it must have the capacity to cool the automobile after it has been standing closed in a hot place. Not only that, the system must have the capacity to cool the car in a short length of time when it is driven at slow city speeds. With the foregoing minimum capacity limitation on the refrigeration system, it is evident that the system will have excessive capacity when the automobile is driven at high speeds, or when it is driven at normal speeds but when the refrigeration demand is comparatively low.

As a result of the foregoing problem, it has been not uncommon in the past to have the automobile air conditioning system frost the coil completely over, and even to discharge pieces of ice into the air circulating system under conditions of high speed highway driving.

Heretofore, various efforts have been made to overcome the problems aforesaid, especially those relating to excessive refrigeration capacity. One effort has been to unload the compressor when the refrigeration capacity exceeds the load, through a by-pass circuit controlled by a solenoid or modulated pressure-operated valve. This has disadvantages in that it causes the compressor head pressures to be excessive at low speeds; and it actually does not eliminate excessive evaporator freezing. It results in a slow cooling-down period at the starting of the vehicle. It eliminates refrigeration and permits humidification to occur during the time the by-pass circuit is effective. It increases the load on the compressor and engine, because it raises back pressure and attempts to lower head pressure. The solenoid control devices give trouble. In unloading by shunting the compressor, the solenoid valves conventionally open against the upstream side to enable modestly-powered solenoids to open against load. As a result, they may leak when the system is first started up and the compressor pulls a normal operating pressure on its suction side, with a high pressure on its condenser side. Leakage of this kind causes a slower initial cooling down period because of such by-passing of the compressor.

It is an object of the present invention to design an air conditioning system for use under the conditions aforesaid, that will not overcool and will not form ice. It is also an object of the invention to provide such a system

which will accomplish its desired results without complex refrigerant circuits, valves, solenoids, and other equipment, and without altering the overall efficiency of the system.

Other objects of the invention include providing a control that will unload the compressor, or at least hold its output down without rendering the evaporator ineffective, when the compressor is subjected to a sudden surge of acceleration, with the system already operating at sufficient capacity to satisfy the load. This is of especial importance when the automobile is required to accelerate rapidly to meet emergency road conditions. Heretofore, the air conditioner load was firmly fixed onto the engine, and increased when the engine accelerated. This produced an undesirable reduction in engine-power available for driving the vehicle. The automatic unloading of the compressor in the present system accomplishes the desirable objective of rendering more, or all, of the engine power available for driving, when the air conditioning apparatus has already attained enough capacity to satisfy the load requirements. And it does so without rendering the evaporator coil ineffective as a cooling and dehumidifying means.

Objects of the invention also include providing a control arrangement that will automatically reduce the output of the system when the flow of air circulating over the coil is reduced, instead of letting the capacity continue high with such reduced load, with consequent excessive icing of the evaporator.

Other objects will appear from the description to follow.

In the drawings:

Figure 1 is a schematic view of a refrigeration circuit of the present invention; and

Figure 2 is a medial section of a valve used in the present system.

Figure 1 shows a compressor 10. The compressor is diagrammatically illustrated as driven from the automobile engine E, although this is not intended to be a limitation on the operating means for the compressor. And the engine is any variable speed engine or motor subject to conditions comparable to those of an automobile engine.

The compressor delivers through a hot gas line 11 into a condenser 12. The outlet side of the condenser is connected by a pipe 13 to an expansion device 14. Preferably, the expansion device is a thermostatic expansion valve.

From the expansion valve, the pipe 15 leads into an evaporator coil 16. The evaporator coil 16 is located either in a space to be cooled or in a duct communicating with the space to be cooled. In the case of an automobile air conditioning device, the evaporator 16 can be located in the rear deck or under the hood; with suitable duct means connecting it with the interior of the car, and air circulating means to force air over the evaporator and into the automobile.

The evaporator 16 connects by a pipe 17 with a capacity control valve 18. This valve 18, in turn, connects into the compressor 10 by a pipe 19, and is located as near to the compressor inlet as practicable, so that it will respond most promptly to changes in conditions produced in the compressor.

The valve 18 is shown in section in Figure 2. It includes a main body shell 24 having an inlet 25 to which the pipe 17 from the evaporator is connected. The body member 24 also has an outlet 26 connected to the compressor suction inlet. There is a valve chamber 27 in the shell 24 between the inlet and the outlet, and it has a valve seat 28. This valve seat is adapted to be closed by a valve head 29, which is ported at 30, or otherwise provided with bleeder means to prevent complete cutoff when the valve is seated. The valve head 29 is mounted on a bellows 32, the other end of which bellows is connected to a cap 33 secured to the body member 24. The cap 33 has a center insert part 34 held in place by a snap ring 35. The insert 34 forms a removable retainer for a spring 36 acting to urge the valve head 29 against the seat 28 in opposition to the fluid pressure coming into the inlet 25. The spring force may be altered by removing the insert 34 and adding a loading shim, or the like, to the spring mounting.

The valve 18 is designated as a capacity control valve. The valve seat is large compared to the bellows size, and the valve head is tapered and enters the valve seat port. The purpose of these characteristics is to have the valve respond to rate of flow through it, so that an increase of quantity of fluid flowing through it with a given back pressure or inlet pressure will be accompanied by an increase in pressure drop through the valve, and hence the valve will tend to close. The valve can modulate to assume positions that throttle as a function of the quantity of flow. When the valve is entirely shut off, the flow is only that through the bleeder port 30, and the bellows responding to pressure in the inlet side will be the principal control on the valve. When the valve is in throttling position, the control will be modified by the quantity of flow through the valve and the velocities thereof.

Operation

In the use of this system, the valve mechanism 18 is set for about 25 p. s i. minimum pressure, assuming that the system is using Freon 12 as the refrigerant, and that it is being used for air conditioning in an automobile or the like.

It is assumed that the system has sufficient capacity to cool the interior of the car operating at about 10 miles per hour, and that its capacity will be substantially greater than the load when the automobile is operating at higher speeds, such as highway speeds of 50 or 60 miles per hour. It is also assumed that the system has capacity to produce its cooling on very hot sunny days. This means that the compressor, operating at low speed, may meet load requirements, but when it is speeded up, it may substantially exceed load requirements.

It is further assumed that at the start of an operation the automobile has been standing closed with the engine not running, and that as soon as the vehicle is started the air conditioning system is also started.

At the starting point, the evaporator pressure is high because of the high temperature conditions. With a thermostatic constant superheat valve as the expansion device, the high temperature conditions acting upon the bulb at the evaporator outlet result in a wide open thermal valve 14. Also, the high pressure in the line 17 is transmitted into the valve 18 and causes that valve to be wide open, since that valve is operated into an opening position in response to elevated pressures on its inlet side.

When the compressor starts, it builds up head pressure on the condenser side, and lowers back pressure on the evaporator side. Refrigerant flow occurs and refrigeration takes place. As pressure and temperature fall in the evaporator, the expansion valve operates to maintain constant superheat in the evaporator outlet by modulating the rate of flow of liquid refrigerant to satisfy the superheat setting of the valve. This may be followed by a corresponding modulation of the valve 18.

As the vehicle speed increases above the low starting speeds—and disregarding the effects of speed-ratio changes between the engine and the wheels—the compressor capacity exceeds the load requirements. Increase of rate of flow of the refrigerant and automatic demand by the thermostatic expansion valve of constant superheat, could cause the temperature of the evaporator to be reduced to a degree so cold that the vehicle is chilled or the evaporator is frozen over. Lowering of evaporator temperature is accompanied by lowered evaporator pressure. The valve 18 is set to maintain a minimum pressure in the valve chamber 27 for a given rate of flow

through the valve, and also in the evaporator. This corresponds to a certain temperature of the evaporator that is suitable to avoid freezing over in this installation. Typically, in the present installations, 19–20 p. s. i. is maintained at the valve 18, with about 24–25 p. s. i. resulting at the evaporator.

At a certain point during the increase in compressor operation beyond load requirements, the suction line pressure will drop so that the valve 18 begins to throttle. Thereafter it will modulate to maintain the optimum refrigeration rate as a function of flow in the suction line and evaporator, and thereby to prevent the latter from freezing over. When the pressure conditions produce a drop greater than 2–4 p. s. i. below the opening pressure, the valve will close entirely with a snap action.

While one controlling force on the valve 18 is its own inlet pressure, the present installation, in contrast with ordinary evaporator or suction line regulator practice, must deal with a compressor subject to speed changes, and consequent capacity changes, and one that is heavily overcapacity for many operating conditions. Therefore, the compressor produces large and frequently sudden changes in the rates of fluid flow. These are reflected in the valve 18. Indeed, the valve 18 is mounted as close to the inlet as practicable, so that it can react quickly to the changes caused by speed changes in the compressor.

When the compressor operates at speeds to overproduce load requirements, and the valve 18 throttles to prevent excess cooling of the evaporator, it starves the compressor, and thereby lowers rate of delivery, despite the speed of the compressor. Consequently, the valve 18 checks the tendency of the system to overrefrigerate. At the same time, it reduces the load on the compressor as it throttles, so that the load on the automobile engine is reduced.

While the foregoing is important in ordinary driving because it represents a saving in refrigeration work, it includes a characteristic of notable importance in rapid acceleration of the engine to meet sudden road conditions. Heretofore, acceleration surges of the engine have been accompanied by corresponding surges of the refrigeration system and of additional power used in refrigeration. Normally, such additional power is not needed in the refrigeration system, but the permanent connection of the engine and compressor means that it is added anyway.

With the present system, a surge of acceleration of the compressor causes abrupt increase in pressure drop through the valve, and this causes the valve 18 to throttle and possibly even to close. Throttling the valve 18 starves the compressor and unloads it so that it is not required to draw the added power from the engine. That power, or a large part of it, is then available to drive the vehicle. The valve 18 may close entirely, since the differential is normally only about 2–4 p. s. i., which almost completely unloads the compressor at the moment that maximum power is needed at the wheels or driving element of the vehicle. The return to normal pressures throughout the system is not instantaneous, because of the differentials in the control elements, the normal sluggishness of any fluid system, and the reduction in compressor efficiency that follows lowering of its inlet pressure. Therefore, the acceleration may be completed before the compressor load is restored, and if the high compressor speed is maintained, then the valve 18 may function to hold minimum pressure in the evaporator, and to continue reduced flow through the compressor inlet.

The valve 18 aids in cooling adjustment inside the vehicle, which has been a problem. With this system, if the vents for the circulating air be throttled to reduce cooling effect, the evaporator may become overcapacity. The valve 18 will then prevent too high a flow of refrigerant in the evaporator, and will correspondingly cut down on compressor input and output to reduce the compressor load corresponding to the reduction of cooling air being circulated.

It may be noted that typical compressor head pressure in this system is 180 p. s. i. This is lower than that required for the same system without the valve **18**, but with a compressor by-pass unloading arrangement. Also, it is characteristic of the bypassing unloader system to give increased suction pressures when the unloader circuit is opened, which is an undesirable effect not produced when the present valve **18** throttles to accomplish the same end of minimizing overcapacity.

The reason for the bleeder passage **30** in the valve **18** is that it is possible to close both valves **14** and **18** at once, and thereby produce an almost complete vacuum on the valve head **29**, resulting in possible damage to compressor reeds and improper oil circulation. Of more importance is the fact that the system must be evacuated of air when it is charged with refrigerant. If vacuum be pulled onto the system, the valve **18**, without the port **30**, could be closed tightly and held closed, thus preventing evacuation of the air. These results cannot happen when the port **30** is present.

It will be seen that with a variable speed compressor such as one driven from the engine of a vehicle, and one having overcapacity, the present capacity control valving arrangement increases very materially the efficiency of the system. The function of the valve **18** in producing regulation of the compressor is of great significance in the present type of air conditioners. The present system has proved far better than other conventional overcapacity controls, in all respects.

What is claimed is:

1. In a refrigeration system, a compressor, a condenser, an expansion device, and an evaporator, the compressor having a capacity to supply the refrigeration load on the evaporator at low compressor speeds, means for operating the compressor at various speeds, including speeds above that required to satisfy the load, means to prevent the compressor from reducing the temperature of the evaporator below a predetermined minimum despite speeds in excess of those required to produce the minimum, and for reducing the load on the compressor to below that corresponding to its speed, said means comprising a fluid-flow responsive valve device between the evaporator and the compressor, the valve device having means to throttle flow through the compressor in response to excess capacity of the compressor, and comprising an inlet on the evaporator side, a partition having a valve port and a valve seat, and an outlet on the compressor side; a valve cooperable with the valve seat, the valve being adapted to position itself as a function of the rate of fluid flow past it through the valve port, and to that end the valve port and the outlet being large enough relative to the size of the inlet and the valve and valve seat having

relative shapes such that the valve produces a pressure drop between the inlet and outlet that varies with the degree of throttling of the valve port by the valve; a collapsible-walled chamber connected to the valve and having its collapsible wall exposed to inlet pressure; and yieldable force means acting on the valve to apply predetermined closing pressure thereon, the size of the valve seat being large relative to the size of the collapsible wall but smaller than the said wall, so that the valve is positioned primarily as a function of pressure drop across the valve, but can be opened from a closed position by pressure on the collapsible wall.

2. The combination of claim 1, wherein the valve device is connected close to the compressor and located at least several feet from the evaporator, so as to quickly respond to changes in speed of the compressor and to restrict fluid flow.

3. The combination of claim 1, wherein the expansion device in the system is a constant superheat, thermostatic expansion valve.

4. The combination of claim 1, wherein the valve apparatus outlet is connected immediately onto the compressor inlet.

5. The combination of claim 1, wherein there is means to limit maximum opening movement of the valve to a position wherein it always maintains pressure drop across the valve port when fluid flows through the port.

6. The combination of claim 1, wherein the valve is tapered and actually enters the valve port when in throttling position.

7. The combination of claim 3, wherein there is restricted passage between the inlet and outlet sides of the partition, permitting equalization of pressures on opposite sides of the valve when it is closed.

8. The combination of claim 1, wherein there is a restricted passage between the inlet and the outlet sides of the partition, permitting pulling a vacuum on the system to evacuate it of air as a part of the operation of charging the system with refrigerant, despite a fluid flow past the valve that produces a pressure drop sufficient to close the valve.

9. The combination of claim 8, wherein the passage is in the valve itself.

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