Dec. 3, 1940. H. B. POWNALL 2,223,900

REFRIGERATION

Filed May 22, 1939 3 Sheets-Sheet 1

Fig. 1

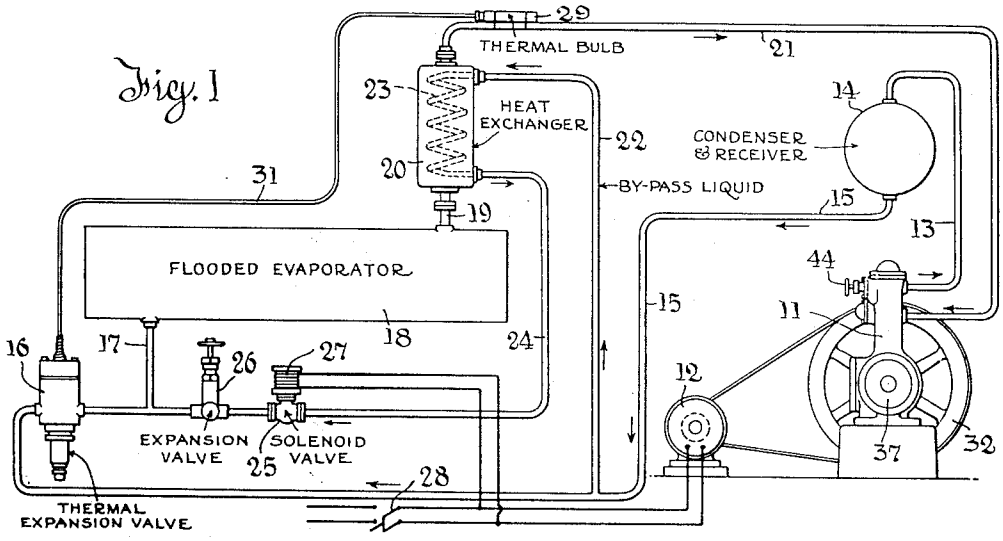


Fig. 2

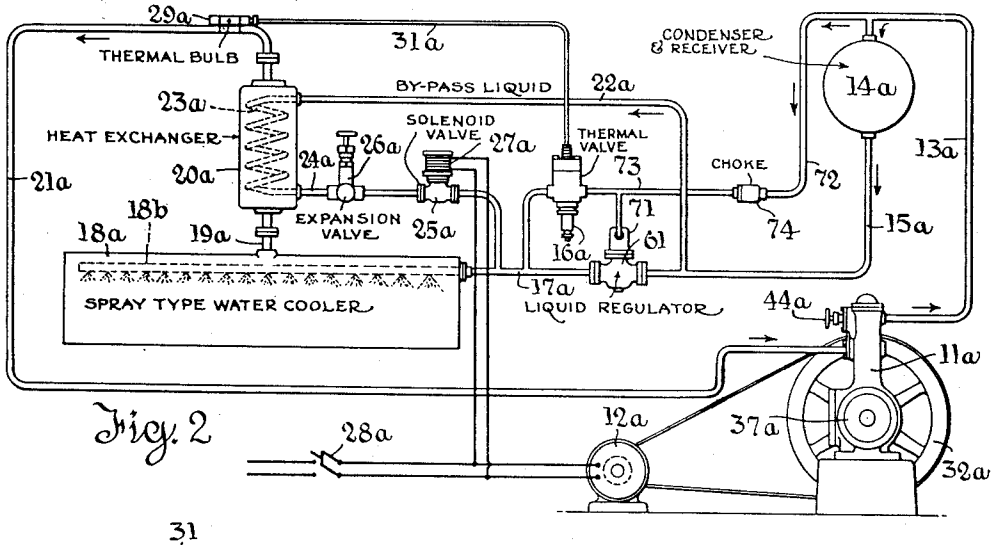
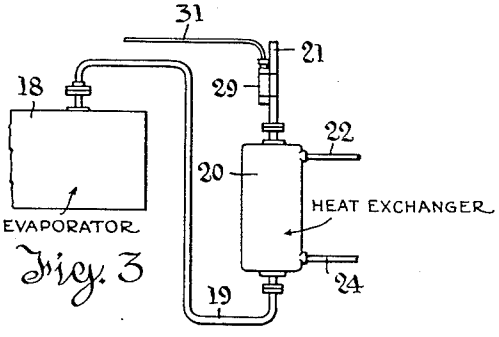


Fig. 3



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REFRIGERATION

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Fig. 4

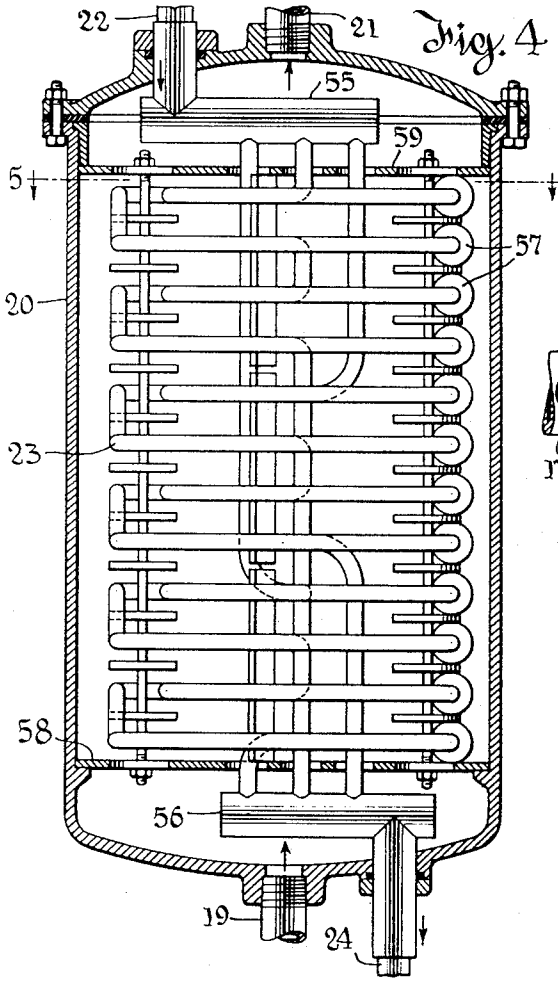


Fig. 6

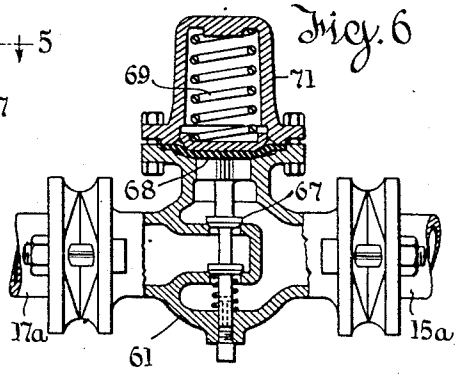
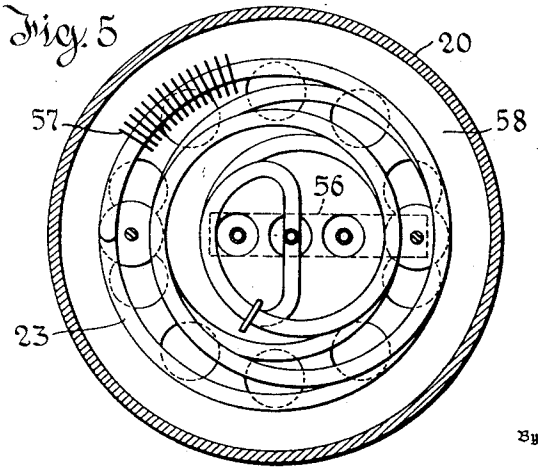


Fig. 5



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REFRIGERATION

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Fig. 7

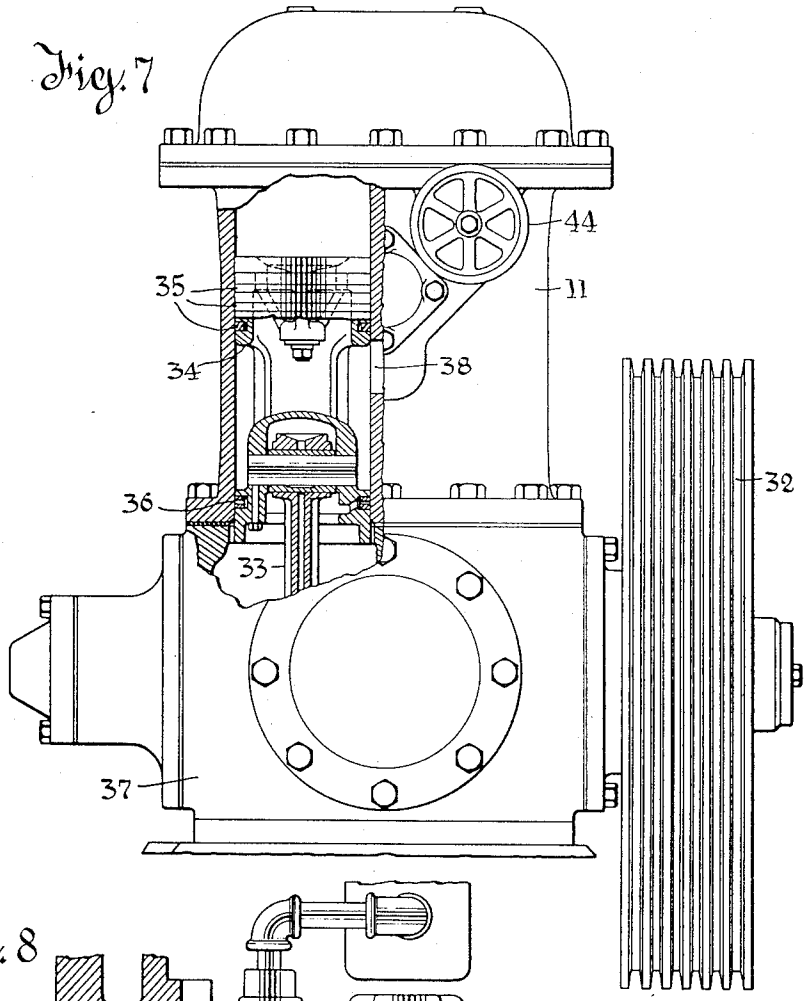
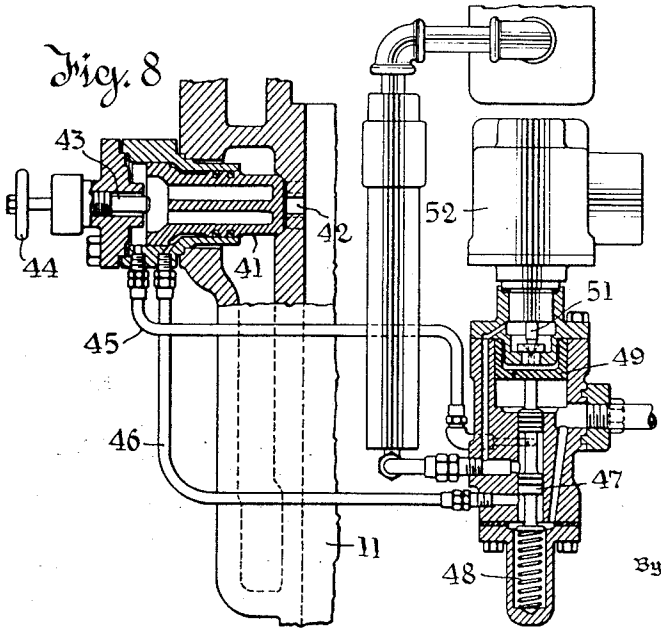


Fig. 8



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Patented Dec. 3, 1940

2,223,900

UNITED STATES PATENT OFFICE

2,223,900

REFRIGERATION

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Application May 22, 1939, Serial No. 275,099

10 Claims. (Cl. 62—115)

This invention relates to refrigeration, and particularly to means for returning to the compressor lubricating oil which leaves the compressor with the compressed refrigerant and, in the absence of special means to remove it, tends to accumulate in the evaporator.

Many schemes have been proposed to accomplish oil return, and those which have been successful have, so far as applicant is aware, involved withdrawal of a mixture of refrigerant and oil from the evaporator, the separation of the refrigerant and oil by fractional distillation, followed by the return of the refrigerant to the circuit and the return of the oil to the compressor, usually to the crank case thereof.

These schemes require the use of considerable additional apparatus with attendant wasteful evaporation of liquid refrigerant, and many of them are characterized by inability to operate satisfactorily over the entire load range of the refrigeration circuit. Difficulty is also encountered with such systems where the compressor runs at various volumetric rates, as is the case in installations in which the compressor is driven at variable speed, or if driven at constant speed is subject to step unloading or progressive unloading.

According to the present invention, the oil is caused to leave the evaporator with the refrigerant and flow toward the compressor through the suction line, the refrigerant which is then in the vapor phase is compressed by the compressor and the oil is returned to the crank case of the compressor. Thus the evaporation of the refrigerant produces useful refrigeration. The invention affords simple means for effecting the desired separation at the compressor inlet. Systems equipped according to the invention will operate satisfactorily even when the compressor runs at different speeds or at variable volumetric rates or is shut down from time to time. The invention can be applied to spray type evaporators, to flooded evaporators, and in fact to nearly any type.

The characteristics of operation vary somewhat with the load, but proper design permits a balance to be struck such that the oil return is substantially uniform. Consequently depletion of oil in the crank case does not occur even if the plant operates at some particular unfavorable load for a long period. This last condition is one which impairs the operation of many of the prior art schemes.

Generally stated, the compressor delivers to the usual condenser and the main path of refrigerant from the condenser to the evaporator is by way of a thermal expansion valve. The thermostatic bulb of such valve is applied to the suction line and thus controls the rate of supply of refrigerant to the evaporator in such a way that all refrigerant flowing to the compressor is slightly superheated and is, therefore, necessarily in the vapor phase.

Interposed between the evaporator proper and that portion of the suction line to which the thermal bulb just mentioned is applied, is a superheating heat-exchanger comprising a shell through which the refrigerant leaving the evaporator flows. In the shell is a finned heat exchange coil, through which a small quantity of liquid refrigerant, less than the minimum amount needed by the evaporator, flows.

Such refrigerant is tapped off from the liquid line between the condenser and the thermal expansion valve, flows thence through the coil of the heat exchanger, thence through a normally open stop valve (such as a magnet valve) which closes as an incident to stoppage of the compressor and thence through an expansion valve, which, for simplicity, is of the manually adjusted type, to the evaporator. This manually adjusted expansion valve is so set that the heat exchanger will superheat refrigerant leaving the evaporator under all load conditions encountered. Consequently the superheater affects the operation of the thermal expansion valve, with the result that the expansion valve supplies to the evaporator the variable demand for refrigerant in excess of that supplied by the manually operable expansion valve.

It is important at this point to explain the effects of variable heat load on the evaporator.

If the quantity of oil in the compressor crank case is to be approximately constant, the quantity of oil which is permitted to remain in the evaporator must be approximately constant, it being obvious that the quantity of oil in other parts of the system is not subject to material variation.

The thermal expansion valve is set to feed more refrigerant to the evaporator as the superheat in the gas passing through the suction line increases. When the evaporator is subjected to a sudden change from a relatively heavy to a lighter heat load, the liquid level in the evaporator lowers. As a result of the lowered liquid level the gaseous refrigerant leaving the heat exchanger tends to increase in superheat. The thermal valve responds to increase the flow of refrigerant to the evaporator, thus raising the

liquid level therein. If the liquid level rises until slugs of liquid enter the heat exchanger, the superheat decreases and the expansion valve reduces the rate of feed of refrigerant. Ultimately equilibrium is reached between the rate of liquid feed and the heat load.

Conversely, when the evaporator is subject to a change from a relatively light to a heavier load, the heat supplied to the liquid causes active evaporation and the entrainment of liquid refrigerant tends to reduce the superheat of the gaseous refrigerant leaving the heat exchanger. The thermal valve responds and reduces the rate of liquid feed to the evaporator until a balance is reached between heat load and liquid feed. In this phase the liquid level lowers until the superheat causes the thermal valve to open sufficiently to establish equilibrium.

An important element of the invention is the heat exchanger which not only superheats the gaseous refrigerant passing the thermal bulb so that the thermal valve functions more effectively regardless of changes of heat load on the evaporator, but also serves to heat the mixture of gas and entrained oil under conditions which prevent reflux of the oil to the evaporator.

Whether this refrigerant-oil mixture enters the exchanger in the form of foam, or assumes the foam form in the exchanger (and either or both actions may occur, according to load conditions) the effect is to separate the gaseous refrigerant from the mixture at a point where the gaseous refrigerant is flowing rapidly so that the oil is caused to flow with the gaseous refrigerant through the suction line.

The effect is to assure oil return under all conditions of operation and stimulate the return as the oil concentration in the evaporator increases, so that the absolute quantity of oil in the evaporator remains nearly constant. This action can be explained as follows:

Under conditions of light load the evaporator is nearly filled, so that liquid refrigerant with a small percentage of oil slops into the heat exchanger where the liquid refrigerant is evaporated and slightly superheated. When the refrigerative load is heavy, and less liquid refrigerant is in the evaporator, the concentration of oil in the refrigerant exceeds the point at which active foaming in the evaporator will occur. Consequently, the evaporator fills with oil filmed bubbles and the vaporous refrigerant flowing from the evaporator into the superheater takes the form of a stream of oily bubbles. The refrigerant is superheated in the heat exchanger and as before entrains the oil and carries it to the compressor.

Thus the oil return has two phases of operation in each of which the heat exchanger performs the useful function of supplying the final superheat. When the concentration is below that necessary to produce foaming, the tendency of liquid refrigerant to enter the heat exchanger is at a maximum and ensures oil return. When the concentration reaches the foaming stage, the foam carries the oil through the relatively longer vertical interval above the liquid level in the evaporator to the heat exchanger.

The amount of liquid refrigerant flowing through the coils in the heat exchanger is fixed by the manually adjustable expansion valve and is less than the minimum demand for refrigerant imposed by the evaporator when the system is operating. The thermal expansion valve stabilizes the entire circuit and controls the liquid level in the evaporator in accordance with the heat load. If the compressor is stopped, the solenoid stop valve closes and terminates the flow of liquid refrigerant through the heat exchanger until the compressor is again put in operation. The action is such that the total quantity of oil in the evaporator tends to remain uniform, and this fact is an important concept underlying the invention.

Preferred embodiments of the invention will now be described with reference to the accompanying drawings, in which

Figure 1 is a diagrammatic view of a refrigerating circuit using a flooded evaporator and having the invention applied.

Figure 2 is a similar view showing a refrigerating circuit of a slightly different type with the invention applied. In this view the evaporator is shown as a spray type water cooler and the circuit is supposed to be one of relatively large capacity, so that a thermostatic valve of limited capacity is used to control a liquid regulating valve of a capacity adequate for the system, the regulating valve performing the actual control of the main flow of liquid from the condenser to the evaporator.

Figure 3 is a fragmentary view showing an alternative location of the heat exchanger which might be used in the structure of either Figure 1 or Figure 2.

Figure 4 is a vertical axial section through the heat exchanger indicating a construction which has been successfully used in practice.

Figure 5 is a section on the line 5—5 of Figure 4.

Figure 6 is a view partly in section and partly in elevation, showing the liquid regulator valve.

Figure 7 is a view partly in elevation and partly in section showing a two-cylinder compressor with trunk pistons and side suction inlets, the inlet valves being in the piston heads. This is a type of compressor with which the invention has been successfully used.

Figure 8 is a diagrammatic view partly in section and partly in elevation of an unloader mechanism which is incorporated in the compressor of Figure 7.

While the invention will be here described as used with a compressor having a variable volumetric rate, this showing is made chiefly to include an affirmative disclosure of such an arrangement. The invention can be applied with compressors which are merely started and stopped and which, when they operate, operate at a constant volumetric rate.

The particular unloading mechanism here illustrated is described and claimed in the patent to Aldinger, No. 2,036,847, April 4, 1936, and so far as the present invention is concerned, is simply an example of one means for varying the volumetric rate of the compressor. Variation of the volumetric rate is the essential point and any known means for causing it may be used. One reason for using compressors of varying volumetric capacity is to stabilize evaporator temperature despite load variations. The invention is applicable to such systems but is not in any sense limited thereto.

Referring first to Figures 1, 4, 5, 7 and 8, the refrigerating circuit comprises a compressor 11 driven by a motor such as the electric motor 12 and connected by way of the high pressure discharge line 13 with the combined condenser and receiver 14 of any suitable type.

From the condenser 14 the liquid line 15 leads

to a thermal expansion valve 16 which delivers refrigerant through connection 17 to a flooded evaporator 18. The evaporator 18 is connected by way of connection 19 with the shell 20 of the heat exchanger. The top of the shell is connected to the suction line 21 which leads to the suction of the compressor 11, hereinafter described in greater detail.

From the liquid line 15 a branch 22 leads to the upper end of the shell 20, where it communicates with a heat exchange coil generally indicated by the numeral 23 on Figure 1, and this coil connects at the lower end of the shell 20 with a line 24 which leads through the solenoid stop valve 25 and manually adjustable expansion valve 26 to the inlet connection 17 in the evaporator. Valve 25 is urged closed by a spring (not shown) and opens when winding 27 is energized. Such solenoid valves are well known.

The winding 27 of the solenoid valve and the motor 12 are both controlled by a switch diagrammatically illustrated at 28 as a knife switch, so that when the switch is closed and the motor 12 runs, the solenoid valve 25 is wide open, and when the motor 12 is stopped, the solenoid valve 25 is closed tightly by the spring above-mentioned.

The expansion valve is of a type common in the art, in which the refrigerant pressure on the discharge side of the valve acts on a diaphragm or other movable abutment connected with the valve in such a way that rising pressure on the discharge side of the expansion valve tends to close the expansion valve.

Connected with the expansion valve 16 is a thermal bulb 29 which is applied to the suction line 21 near the point at which this line is connected with the upper end of the shell 20. The bulb 29 is subject to the temperature of the suction line and contains an expansible fluid of suitable character which develops pressure on rising temperature, the pressure being transmitted through the tube 31 to a diaphragm or abutment within the valve 16 in such a way that rising temperature at the bulb 29 tends to open the expansion valve.

Such valves commonly include an adjustable biasing spring. Any type of thermostatic expansion valve may be used, provided it is of a type suited to control the superheat in the suction line 21 near its connection with the shell 20. Inasmuch as such expansion valves are standard articles of commerce, no attempt to illustrate details has been made.

The expansion valve 26 is just a manually adjustable throttling valve of conventional form.

The compressor 11 is preferably of the type illustrated in Figure 7. The grooved belt wheel 32 drives a crank shaft (not shown) and this shaft drives through connecting rods 33 a pair of trunk pistons, one of which appears at 34. These trunk pistons have packing rings on the upper head at 35 and the inlet valve is mounted in the piston and is of the suction operated type. The lower end of the piston is simply a guiding member and carries an oil scraper ring indicated at 36. This ring is not a packing ring, but is designed to act on oil on the cylinder walls and feed it downward into the crank case 37. Such rings are in extensive use in compressors and in gasoline engines.

The suction line 21 enters through the cylinder wall by way of the port 38 and thus communicates with the middle portion of the piston between the packing rings 35 and the oil ring 36. It follows that the crank case is substantially at suction pressure, although the communication between the crank case and the suction connection is somewhat restricted.

Such compressors may be run at variable speed, but because most driving motors are of the alternating current type, the usual practice is to run the compressor at constant speed and provide an unloader to vary its volumetric rate. The compressor shown in the drawings is so equipped.

Referring to Figure 8, the combined unloader piston and valve 41 is shown in its loading position in which it closes the by-pass or unloading port 42 in the cylinder wall. These ports are located at part stroke so that they will be overtraveled by the piston, say at half stroke. If the ports are open the piston does not start compression until it has overtraveled the ports.

The stem 43 operated by hand wheel 44 is merely a manually adjustable means, operable to lock the piston 41 in its loading position. Normally it is retracted, as shown.

The unloading piston and valve is operated by refrigerant under pressure preferably derived from the discharge connection of the compressor, as explained in the Aldinger patent above mentioned. Two pressure connections are shown at 45 and 46 and they are controlled by a piston valve 47 which is biased in one direction by a spring 48 and may be shifted against such bias by a piston 49. The pressure acting on this piston is controlled by a needle valve 51 which is closed when the winding in the case 52 is deenergized and open when that winding is energized. The energization of the winding 52 may be controlled by any means, but since such control is not material to the present invention, none is illustrated.

Known controls, for example, are thermostatic switches responding to various temperatures, and pressure switches responding to suction pressure in the system. So far as the present invention is concerned, the material point is that the oil return system will operate satisfactorily with compressors operating at varying volumetric rates. It will also operate with compressors which operate at constant volumetric rates and are merely started and stopped.

The construction of the superheating heat exchanger is shown in detail in Figures 4 and 5. The shell 20 is provided with a removable head as shown. The coil structure indicated generally by the numeral 23 in Figure 1 takes the form of three adjacent coils connected in parallel between an upper inlet header 55 to which the branch 22 is connected and a lower discharge header 56 from which the line 24 leads. These coils are spirally finned, the fins being indicated at 57 in Figures 4 and 5, but omitted elsewhere because they would obscure the view.

To all intents and purposes, the shell 20 is filled with a mass of finned tubing. The particular form of the finned coils is not deemed to be vitally important, provided it is adapted to exert what may be described as an eliminator effect. By this is meant that the fins and the tubes arrest liquid which may enter the shell 20 through the connection 19 and hold the liquid in contact with the heat exchange surface until the refrigerant is completely evaporated. The oil will adhere to the extended finned surface in such a way that it may readily be entrained by the vaporous refrigerant flowing rapidly from the shell 20 to the suction line 21.

It will be observed that perforated plate diaphragms 58 and 59 extend across the shell 20

above and below the coils and that the lower diaphragm 58, together with the lower discharge header 56, resists the too free surging of liquid entering at 19 upward through the shell 20.

The effect is to offer a large heat exchange and liquid arresting surface to enable heat to flow from the warm liquid to the refrigerant leaving the evaporator so that the heat performs a final refrigerant superheating function under conditions which favor the entrainment of oil and its delivery to the suction line 21.

As has been described, when the heat load on the evaporator is low, the thermal valve causes the evaporator to fill nearly full. The refrigerant evaporating in the evaporator then causes liquid to slop through the connection 19 into the shell 20 where the liquid refrigerant component will be evaporated by heat derived from the warm liquid flowing through the coil structure 23. Thus oil concentration occurs in the heat exchanger, foam forms in the exchanger and the oily foam flows to the compressor.

Under heavy load conditions, the liquid level in the evaporator is lower, the oil concentration is higher, and active foaming occurs in the evaporator itself. This foaming carries oil into the shell 20 and this oil is entrained with off-flowing refrigerant as before.

Under intermediate conditions there may be different degrees of foaming in the evaporator and different degrees of slopping of liquid refrigerant into the shell 20.

It is difficult to prescribe the operation precisely because under operating conditions it is impracticable to see what takes place, but extensive tests have shown that the operation of the system is stable, so that the absolute quantity of oil in the evaporator tends to remain nearly constant, regardless of the heat load or volumetric rate of the compressor, with the result that the quantity of oil in the crank case 37 also tends to remain nearly constant. The fact of operation is demonstrated. The best known theory of what occurs has been stated.

The arrangement shown in Figure 2 is essentially similar to that shown in Figure 1. Those parts which are identical with similar parts in Figure 1 are given the same reference numeral with the letter "*a*".

The differences will now be specified. The expansion valve 16*a* does not directly control the main flow of refrigerant. The liquid line 15*a* from the condenser 14*a* leads to a liquid regulator valve 61 which is shown in detail in Figure 6. This has a balanced poppet valve unit 67 operated by a diaphragm 68 which is loaded by a spring 69 in a closing direction and which is subject in a valve opening direction to the pressure in the high pressure liquid line 15*a*. The space above the diaphragm 68 is closed by a cap 71 which serves not only as a seat for the spring 69 but also as a pressure chamber for loading the diaphragm 68 in a closing direction.

A choke 74 is interposed in a branch line 72 and supplies refrigerant to the line 73 which is connected to the inlet of the thermal expansion valve 16*a* and with the space in cap 71 above diaphragm 68. Expansion valve 16*a* delivers to the line 17*a*.

Thus the valve 16*a* responds to the temperature at the bulb 29*a* acting in opposition to the pressure in the line 17*a* and exercises a control on the valve 67 which thus responds to the combined effect of temperature and pressure. Hence the liquid regulator 61 is simply a large capacity valve whose opening and closing has the same flow controlling characteristics as does the expansion valve 16*a*. This is simply a convenient way of getting large capacity regulation by the use of a small thermal expansion valve. No novelty is here claimed for that arrangement.

The evaporator 18*a* is indicated as a spray type water cooler, the liquid refrigerant being fed by the connection 17*a* into a spray pipe 18*b*. The water to be cooled passes through tubes within the evaporator over which the refrigerant showers. These tubes are not illustrated as they would simply complicate the drawings. The structure of the cooler is well known and is not a feature of the invention.

Figure 3, on which are applied the same reference numerals as those used in Figure 1, involves no material change of construction but only a change of location. The purpose of the view is merely to indicate that the shell 20 need not be directly above the evaporator 18, although that location is preferred.

Where head room does not permit, the exchanger may be located to one side of the evaporator and even at a slightly lower level, as shown in Figure 3. The operative characteristics are essentially similar. Obviously the arrangement shown in Figure 3 could be used with the scheme of Figure 2 in case of necessity.

With reference to all embodiments of the invention, it may be noted that the system operates with a wet discharge from the evaporator, that is, the evaporator is so designed that it foams or primes or foams and primes. The final evaporation and superheat of refrigerant are secured by heat exchange with a portion only of the liquid refrigerant flowing to the evaporator. It has not been found practicable to use in the heat exchanger all the liquid refrigerant flowing to the evaporator because where such arrangements have been attempted, a persistent and objectionable cycling takes place.

The thermal expansion valve is used to ensure changes of level of liquid in the evaporator as an incident to changes of load. The device is so contrived that when the oil concentration is high and foaming occurs, the foam is caused to carry the oil into the heat exchanger drum. When the oil concentration is low, and foaming is less or does not occur, priming and slopping over into the heat exchanger serve to carry oil into the exchanger.

Close spacing of the coils and of the finned surface in the exchanger functions to arrest liquid refrigerant, ensure its evaporation, and to retain the oil in a dispersed condition favorable to entrainment by the rapidly off-flowing vaporous refrigerant.

While the controlling factors in design cannot be stated with precision, it has been found that adjustment of the expansion valve 26 will permit the operation to establish a stable condition under which there is an inherent tendency for the absolute quantity of oil in the evaporator to remain constant. Because it does remain constant, there is an inherent tendency for the quantity of oil in the compressor crank case to remain constant.

The idea of returning the oil with the refrigerant to the inlet of a trunk piston, side inlet compressor is an important feature of the invention. It offers a remarkably good oil separation. While the theory may be erroneous and is not material to the invention, it is believed that the oil contacts the piston and adheres to it

by capillary attraction, so that the refrigerant flows freely through the inlet valve and the oil flows to the oil ring and is progressively returned to the crank case by the oil scraping action thereof.

However this may be, the oil return mechanism involves very little accessory apparatus. The superheat imparted to the refrigerant in the shell 20 is derived from a portion of the liquid refrigerant flowing to the evaporator, so that the efficiency of the circuit is not penalized. The system requires no periodic attention. It contains no flow traps and no by-pass connections from the evaporator to the compressor. Consequently, the efficiency of the refrigerating circuit is not impaired and the circuit cannot be bypassed by the accidental sticking of any by-pass controlling valve. The separation of the oil and the refrigerant occurs at a time when the refrigerant is superheated and hence is in the vapor phase. From this it follows that liquid refrigerant does not find its way to the crank case of the compressor.

While several embodiments of the invention have been described in considerable detail, these descriptions are intended to be illustrative of the best known embodiments, and are not intended to be limiting. The scope of the invention is defined in the claims.

What is claimed is:

1. The method of limiting the accumulation of oil in the evaporator of a refrigerator circuit of the compressor-condenser-evaporator circuit type charged with a refrigerant in which the oil is soluble which comprises the refrigerant in the evaporator to foam, causing said foam to flow toward the compressor; heating said foam as it flows toward the compressor by heat exchange with warm liquid refrigerant flowing to the evaporator until substantially all refrigerant is in the vapor phase, varying the rate of total supply of refrigerant to the evaporator according to evaporator pressure and the temperature of the foam after such heat exchange to assure at least slight superheating of the refrigerant component of the foam and at least before passage through said compressor, breaking down said foam and causing the oil and vaporous refrigerant to flow in divergent paths.

2. The method of operating a refrigerating system of the compressor-condenser-evaporator circuit type in which a volatile refrigerant in which the compressor lubricant is soluble is used, which method comprises balancing the tendency of oil to flow with refrigerant to the evaporator and accumulate in the evaporator by fractional distillation, by entraining lubricant with refrigerant flowing from the evaporator to the compressor, such entrainment being produced by so operating the evaporator that the refrigerant foams and primes, causing foam to flow toward the compressor, vaporizing liquid refrigerant in such foam by heat exchange with a portion only of the liquid refrigerant flowing from the condenser to the evaporator, varying the rate of supply of refrigerant to the evaporator according to evaporator pressure and the temperature of the foam after such heat exchange to assure at least slight superheating of the refrigerant component of the foam, and breaking down such foam as it enters the compressor whereby the lubricant resumes its lubricating function and the vaporous refrigerant approximately freed of lubricant is compressed.

3. The method of operating a refrigerating system of the compressor-condenser-evaporator circuit type, in which the circulated refrigerant is of a type in which the lubricant used in the compressor is soluble, which method comprises limiting the rate of admission of refrigerant to the evaporator sufficiently to assure moderate superheating of vaporous refrigerant flowing to the compressor, without inhibiting priming and foaming in the evaporator; subdividing the liquid refrigerant flowing from the condenser to the evaporator into two streams which are distinct at least to the extent that they are substantially out of heat exchange relation with each other; supplying heat to the foam leaving the evaporator by heat exchange with one of said streams; and feeding the foam to the compressor whereby the foam is caused to break down, the lubricant serving to lubricate the compressor and the vaporous refrigerant being compressed and discharged to the condenser at least partially freed of lubricant.

4. A refrigerating circuit containing a volatile refrigerant and comprising in combination, a compressor provided with lubricating means using a liquid lubricant soluble in said refrigerant; a condenser into which the compressor discharges; expansion valve means of the superheat control type fed by the condenser and including a thermal element; an evaporator fed by said expansion valve means; a heat exchanger including a shell connected to the evaporator above the maximum liquid level therein and containing a heat exchange element offering a flow path for a heating medium; a suction line leading from said shell to the suction intake of said compressor, said thermal element being subject to suction temperature in said line; means for separating lubricant from gaseous refrigerant as they approach the compressor; and a by-pass leading from said condenser to and through said heat exchange flow path to said evaporator, said by-pass including means to control the rate of flow therethrough.

5. A refrigerating circuit containing a volatile refrigerant and comprising in combination, a compressor provided with lubricating means using a liquid lubricant soluble in said refrigerant; a condenser into which the compressor discharges; expansion valve means of the superheat control type fed by the condenser and including a thermal element; an evaporator fed by said expansion valve means; a heat exchanger including a shell connected to the evaporator above the maximum liquid level therein and containing a heat exchange element offering a flow path for a heating medium; a suction line leading from said shell to the suction intake of said compressor, said thermal element being subject to suction temperature in said line; means forming part of said compressor for separating lubricant from refrigerant and comprising a cylinder scraping ring carried by the piston and arranged to divert lubricant and means for causing the entering mixture to impinge upon the piston; and a by-pass leading from said condenser to and through said exchange flow path to said evaporator, said by-pass including means to control the rate of flow therethrough.

6. The combination defined in claim 4 in which the flow through the shell from the evaporator and the flow of liquid refrigerant through the flow path of the heat exchange element toward the evaporator are in generally opposite directions.

7. The combination defined in claim 4 in

which the heat exchange element takes the form of a plurality of closely spaced finned tubes, the fins being generally parallel with the direction of flow through the shell and the coils extending across substantially the entire transverse extent of the shell.

8. The combination defined in claim 4, in which the means controlling the rate of flow through the by-pass is a manually adjustable expansion valve located between the heat exchange flow path and the evaporator.

9. The combination with the structure defined in claim 4 of a stop valve interposed in said by-pass; and valve controlling means arranged to open said stop valve when the compressor operates and close it at other times.

10. The combination with the structure defined in claim 4 of means for varying the volumetric rate of the compressor.

HENRY B. POWNALL.

CERTIFICATE OF CORRECTION.

Patent No. 2,223,900. December 3, 1940.

HENRY B. POWNALL.

It is hereby certified that error appears in the printed specification of the above numbered patent requiring correction as follows: Page 5, first column, line 35, claim 1, after the word "comprises" insert --causing--; and that the said Letters Patent should be read with this correction therein that the same may conform to the record of the case in the Patent Office.

Signed and sealed this 14th day of January, A. D. 1941.

(Seal)

Henry Van Arsdale,
Acting Commissioner of Patents.