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(54) Apparatus for recovering and purifying refrigerant.

(57) An apparatus (10) for recovering and purifying refrigerant contained in a refrigeration system has a first mode of operation wherein refrigerant is withdrawn from the system being serviced, compressed, condensed and delivered in liquid form to a refrigerant storage means (86). The pressure ratio across the recovery compressor (44) is monitored, and, when the pressure ratio exceeds a value above which the compressor (44) may be adversely affected withdrawal of the refrigerant from the refrigeration system is terminated. The system is then operated in a closed, cooling mode wherein refrigerant recovered from the system and stored in the storage means (86) is withdrawn therefrom by the compressor (44), compressed condensed, and expanded and returned to the storage means (86) to thereby lower the temperature and pressure of the storage means (86) and the refrigerant contained therein. Means for purifying (32) the withdrawn refrigerant are located upstream from the compressor suction port (42) so that refrigerant purification takes place during all modes of operation. When the temperature in the refrigerant storage cylinder (86) falls to a predetermined level the system is returned to the recovery mode. During the second recovery cycle, because of the substantially lower temperature in the recovery system (10), the refrigerant storage cylinder (86) effectively serves as a condenser.

The system (10) may be operated in a liquid recovery cycle wherein liquid refrigerant is delivered directly to the storage means (86). The system (10) is shifted from liquid recovery to vapor recovery responsive to, either, a decrease in liquid level increase in the storage means (86), or, several system control parameters. Apparatus for sampling the purity (104) of refrigerant flowing through the recovery system is provided.

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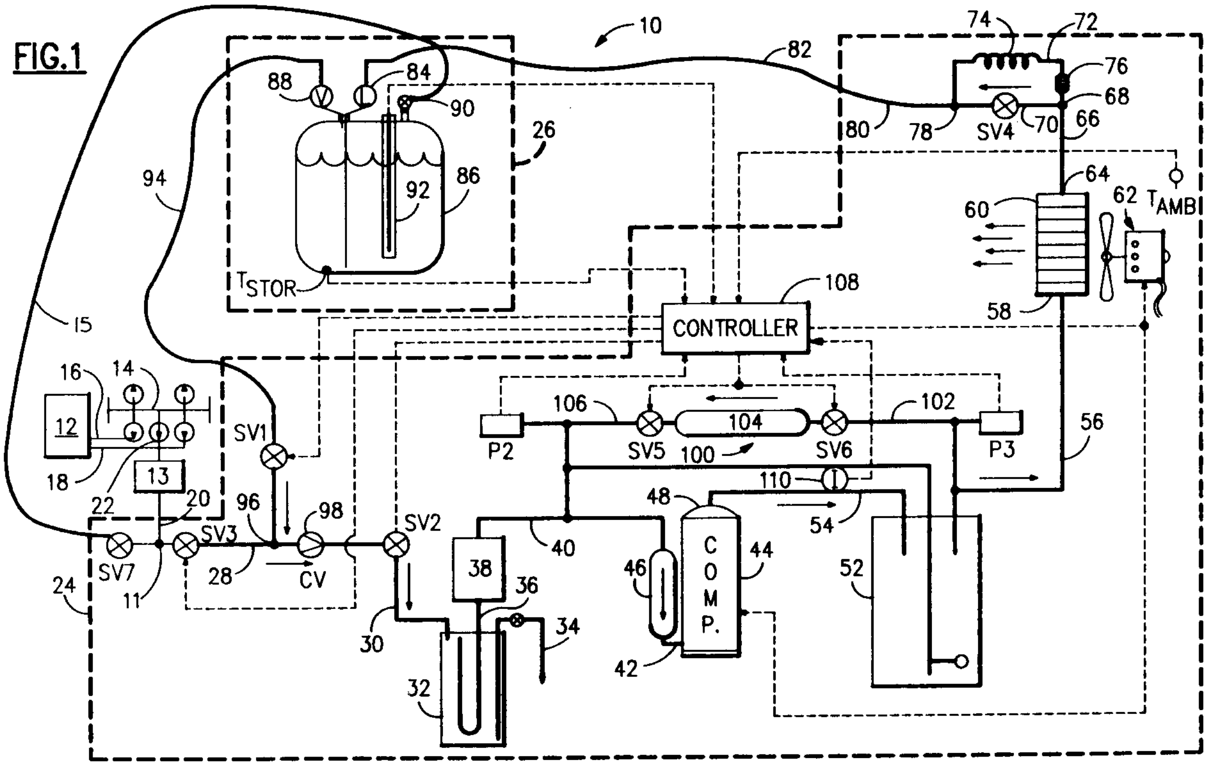


FIG.1
10
82
74
72
76
68
66
70
SV4
78
80
88
84
90
26
94
92
86
64
60
62
TAMB
TSTOR
108
CONTROLLER
58
15
16
14
12
18
13
22
20
SV1
96
98
CV
SV7
SV3
11
28
24
SV2
30
106
P2
SV5
104
100
SV6
102
P3
56
110
48
54
40
38
36
34
32
46
44
COMP.
42
52

A wide variety of mechanical refrigeration systems are currently in use in a wide variety of applications. These applications include domestic refrigeration, commercial refrigeration, air conditioning, dehumidifying, food freezing, cooling and manufacturing processes, and numerous other applications. The vast majority of mechanical refrigeration systems operate according to similar, well known principals, employing a closed-loop fluid circuit through which a refrigerant flows. A number of saturated fluorocarbon compounds and azeotropes are commonly used as refrigerants in refrigeration systems. Representative of these refrigerants are R-12, R-22, R-500 and R-502.

Those familiar with mechanical refrigeration systems will recognize that such systems periodically require service. Such service may include removal, of, and replacement or repair of, a component of the system. Further during normal system operation the refrigerant can become contaminated by foreign matter within the refrigeration circuit, or by excess moisture in the system. The presence of excess moisture can cause ice formation in the expansion valves and capillary tubes, corrosion of metal, copper plating and chemical damage to insulation in hermetic compressors. Acid can be present due to motor burn out which causes overheating of the refrigerant. Such burn outs can be temporary or localized in nature as in the case of a friction producing chip which produces a local hot spot which overheats the refrigerant. The main acid of concern is HCL but other acids and contaminants can be produced as the decomposition products of oil, insulation, varnish, gaskets and adhesives. Such contamination may lead to component failure or it may be desirable to change the refrigerant to improve the operating efficiency of the system.

When servicing a refrigeration system it has been the practice for the refrigerant to be vented into the atmosphere, before the apparatus is serviced and repaired. The circuit is then evacuated by a vacuum pump, which vents additional refrigerant to the atmosphere, and recharged with new refrigerant. This procedure has now become unacceptable for environmental reasons, specifically, it is believed that the release of such fluorocarbons depletes the concentration of ozone in the atmosphere. This depletion of the ozone layer is believed to adversely impact the environment and human health. Further, the cost of refrigerant is now becoming an important factor with respect to service cost, and such a waste of refrigerant, which could be recovered, purified and reused, is no longer acceptable.

To avoid release of fluorocarbons into the atmosphere, devices have been provided that are designed to recover the refrigerant from refrigeration systems. The devices often include means for processing the refrigerants so recovered so that the refrigerant may be reused. Representative examples of such devices are shown in the following United States Patents: 4,441,330 "Refrigerant Recovery And Recharging System" to Lower et al; 4,476,688 "Refrigerant Recovery And Purification System" to Goddard; 4,766,733 "Refrigerant Reclamation And Charging Unit" to Scuderi; 4,809,520 "Refrigerant Recovery And Purification System" to Manz et al; 4,862,699 "Method And Apparatus For Recovering, Purifying and Separating Refrigerant From Its Lubricant" to Lounis; 4,903,499 "Refrigerant Recovery System" to Merritt; and 4,942,741 "Refrigerant Recovery Device" to Hancock et al.

When most such systems are operating, a recovery compressor is used to withdraw the refrigerant from the unit being serviced.

As the pressure in the service unit is drawn down, the pressure differential across the recovery compressor increases because the pressure on the suction side of the compressor becomes increasingly lower while the pressure on the discharge side of the compressor stays constant. High compressor pressure differentials can be destructive to compressor internal components because of the unacceptably high internal compressor temperatures which accompany them and the increased stresses on compressor bearing surfaces. Limitations on the pressure differentials or pressure ratio across the recovery compressors are thus necessary, such limitations, in turn can limit the percentage of the total charge of refrigerant contained within the unit being serviced that may be successfully recovered.

When using such recovery systems in servicing larger refrigeration systems it is particularly advantageous to have the capability of withdrawing refrigerant from the system in the liquid form and delivering it directly to a storage cylinder. The recovery of the refrigerant in liquid form, because of its much greater density, is obviously far quicker than recovery in the vapor state.

Another feature considered desirable in such recovery systems is to have a refrigerant quality test system incorporated in the recovery systems itself.

According to the invention protection of the recovery compressor is achieved by providing an apparatus and method for recovering compressible refrigerant from a refrigeration system and delivering the recovered refrigerant to a refrigeration storage means. The recovery method includes the steps of withdrawing refrigerant from a refrigeration system being serviced and compressing the withdrawn refrigerant in a compressor to form a high pressure gaseous refrigerant. The high pressure gaseous refrigerant is delivered to a condenser where it is condensed to form liquid refrigerant. The liquid refrigerant from the condenser is delivered to the refrigerant storage means. Means are provided for determining the pressure ratio across the recovery system compressor and monitoring the determined pressure ratio. When the monitored pressure ratio exceeds a predetermined value above which the compressor may be adversely affected the system is caused to stop the withdrawal of refrigerant from the refrigeration system being serviced.

At that point, the system begins to withdraw stored refrigerant from the storage means. The refrigerant withdrawn from the storage means is then compressed in the same compressor which was used to compress refrigerant withdrawn from the refrigeration system. This refrigerant is then condensed to form liquid refrigerant which is then passed through a suitable expansion device and delivered back to the storage means to thereby cool the storage means and the refrigerant contained therein. This cooling cycle is performed for a period of time until the temperature of the storage means falls to a predetermined value. At that point the system resumes withdrawal of refrigerant from the refrigeration system being serviced. When the suction pressure of the recovery system compressor falls below a predetermined value the recovery operation is terminated.

According to another feature of the invention a refrigerant recovery system is operated to withdraw compressible refrigerant from a refrigeration system by first drawing liquid refrigerant from the system being serviced through a suitable conduit and delivering the withdrawn refrigerant to a refrigerant storage means. In the refrigerant storage means at least a portion of the refrigerant so withdrawn exits in gaseous form. A portion of this gaseous refrigerant is withdrawn from the storage means and compressed to form a high pressure gaseous refrigerant. The high pressure gaseous refrigerant is then condensed to form a high pressure liquid refrigerant. The high pressure liquid refrigerant is passed through an expansion device where the refrigerant under goes a pressure drop and is at least partially flashed to a vapor. The liquid vapor mixture is then delivered to the storage means where it evaporates and absorbs heat from the refrigerant within the storage means thereby cooling the storage means and lowering the pressure therein, thereby increasing the withdrawal of liquid refrigerant from the refrigeration system through the conduit.

According to one control system, the rate of liquid level change in the storage means is monitored. When that rate reaches a value which indicates that liquid is no longer being withdrawn, the system shifts automatically to the vapor recovery mode of operation.

According to another control system means are provided for sensing a system control parameter which has a detectable change in value which occurs at a time which may be correlated with the time at which the state of the refrigerant being withdrawn from the refrigeration system changes from a liquid to vapor. A signal indicating that the detectable change has occurred causes the system to shift from a liquid recovery mode to a vapor recovery mode.

Purity of the recovered refrigerant may be determined by use of a method and apparatus for sampling the purity of the refrigerant flowing through a refrigeration system. The refrigeration system includes a compressor having an inlet port which has an inlet conduit associated therewith and defining in part the low pressure side of the refrigeration system. The compressor further has an outlet port for having an out conduit associated therewith which defines in part the high pressure side of the refrigeration system. A refrigerant sampling chamber is operatively connected in parallel in fluid flow communication with the compressor. The compressor is then operated to establish the flow of refrigerant through the system and a quantity of refrigerant is withdrawn from the high pressure side of the system. The withdrawn quantity of refrigerant is then passed through the sampling chamber and thence returned to the low pressure side of the system.

Figure 1 is a diagrammatical representation of a refrigeration recovery and purifying system embodying the principles of the present invention;

Figure 2 is a flow chart of an exemplary program for controlling the elements of the present invention in a vapor recovery cycle;

Figure 3 is a flow chart of an exemplary program for controlling the elements of the present invention in a recycle mode of operation; and

Figure 4A is a flow chart of a program for controlling the system in a liquid recovery mode of operation using rate of liquid recovery as a control parameter;

Figure 4B is a continuation of the flow chart of Figure 4A showing a program for controlling the system in a vapor recovery mode of operation;

Figure 5 is a graphical showing of quantity of refrigerant recovered versus time in the liquid recovery mode of operation;

Figure 6A is a flow chart of a program for controlling the system in a liquid recovery mode of operation using a control parameter which may be correlated to the change of state of refrigerant being withdrawn;

Figure 6B is a continuation of the flow chart of Figure 6A showing a program for controlling the system in a vapor recovery mode of operation;

Figure 7 is a graphical showing of storage cylinder temperature versus time in the liquid recovery mode of operation;

Figure 8 is a graphical showing of pressure leaving the compressor versus time in the liquid recovery mode of operation;

Figure 9 is a graphical showing of pressure entering the compressor versus time in the liquid mode of operation;

Figure 10 is a chart showing the operation of the various components of the system during different modes of operation.

An apparatus for recovering and purifying the refrigerant contained in a refrigeration system is generally shown at reference numeral 10 in Figure 1. The refrigeration system to be evacuated is generally indicated at 12 and may be virtually any mechanical refrigeration system.

As shown the interface or tap between the recovery and purification system 10 and the system being serviced 12 is a standard gauge and service manifold 14. The manifold 14 is connected to the refrigeration system to be serviced in a standard manner with one line 16 connected to the low pressure side of the system 12 and another line 18 connected to the high pressure side of the system. A high pressure refrigerant line 20 is interconnected between the service connection 22 of the service manifold and a T connection 11 for coupling the line 20 to the recovery system 10.

Located in the interconnecting line 20 is a filter-dryer 13 which is mounted external of the recovery system. This device as will be seen, is normally installed in the line 20 only when the system is to be operated first in the liquid recovery mode of operation.

The recovery system 10 includes two sections, as shown in Figure 1 the components and controls of the recovery system are contained within a self contained compact housing (not shown) schematically represented by the dotted line 24. A refrigerant storage section of the system is contained within the confines of the dotted lines 26. The details of each of these sections and their interconnection and interaction with one another will now be described in detail.

As will be appreciated as the description of the operation of the system continues there are two refrigerant paths extending from the T-connection 11 at the end of interconnecting line 20. The first path, i.e. the liquid path, extends to the left of the T-11 to an electrically actuatable solenoid valve SV7. This valve will selectively allow refrigerant to pass therethrough when actuated to its open position or will prevent the flow of refrigerant therethrough when electrically actuated to its closed position. Additional electrically actuatable solenoid valves contained in the system operate in the same conventional manner. From SV7 a liquid refrigerant line 15 extends to the refrigerant storage section of the system 26 where it communicates through a valve 90 with a refrigerant storage cylinder 86. In the liquid recovery mode of operation of the system liquid refrigerant passes through the line 15 directly from the refrigeration system 12 to the storage cylinder 86.

When the system is operated in the vapor recovery mode gaseous refrigerant flowing through the interconnecting line 20 flows through the T-11 and to the right to electrically actuatable solenoid valve SV3. From SV3 refrigerant passes through a conduit 28 through a check valve 98 to a second electrically actuatable solenoid valve SV2. From SV2 an appropriate conduit 30 conducts the refrigerant to the inlet of a combination accumulator/oil trap 32 having a drain valve 34. Refrigerant gas is then drawn from the oil trap through conduit 36 to an acid purification filter-dryer 38 where impurities such as acid, moisture, foreign particles and the like are removed before the gases are passed via conduit 40 to the suction port 42 of the compressor 44. A suction line accumulator 46 is disposed in the conduit 42 to assure that no liquid refrigerant passes to the suction port 42 of the compressor. The compressor 44 is preferably of the rotary type, which are readily commercially available from a number of compressor manufacturers but may be of any type such as reciprocating, scroll or screw.

From the compressor discharge port 48 gaseous refrigerant is directed through conduit 50 to a conventional float operated oil separator 52 where oil from the recovery system compressor 44 is separated from the gaseous refrigerant and directed via float controlled return line 54 to the conduit 40 communicating with the suction port of the compressor. From the outlet of the oil separator 52 gaseous refrigerant passes via conduit 56 to the inlet of a heat exchanger/condenser coil 60. An electrically actuated condenser fan 62 is associated with the coil 60 to direct the flow of ambient air through the coil as will be described in connection with the operation of the system.

From the outlet 64 of the condenser coil 60 an appropriate conduit 66 conducts refrigerant to a T-connection 68. From the T 68 one conduit 70 passes to another electrically actuated solenoid valve SV4 while the other branch 72 of the T passes to a suitable refrigerant expansion device 74. In the illustrated embodiment the expansion device 74 is a capillary tube and a strainer 76 is disposed in the refrigerant line 72 upstream from the capillary tube to remove any particles which might potentially block the capillary. It should be appreciated that the expansion device could comprise any of the other numerous well known refrigerant expansion devices which are widely commercially available. The conduit 72 containing the expansion device 74 and the conduit 70 containing the valve SV4 rejoin at a second T connection 78 downstream from both devices. It will be appreciated that the solenoid valve SV4 and the expansion device 74 are in a parallel fluid flow relationship. As a result, when the solenoid valve SV4 is open the flow of refrigerant will be, because of the high resistance of the expansion device, through the solenoid valve in a substantially unrestricted manner. On the other hand, when the valve SV4 is closed, the flow of refrigerant will be through the high resistance path provided by the expansion device. Combination devices such as electronically actuated expansion valves are known which would combine the functions of the valves SV4 and the capillary tube 74, however, as configured and described above, the desired function is obtained at a minimum cost.

From the second T-78 a conduit 80 passes to an appropriate coupling (not shown) for connection of the system as defined by the confines of the line 24, via a flexible refrigerant line 82 to the liquid inlet port 84 of the previously referred to refillable refrigerant storage container 86. The container 86 is of conventional construction and includes a second port 88 adapted for vapor outlet. The storage cylinder 86 further includes a liquid level indicator 92. The liquid level indicator, for example, may comprise a compact continuous liquid level sensor of the type available from Imo Delaval Inc., Gems Sensors Division. Such an indicator is capable of providing an electrical signal indicative of the level of the refrigerant contained within the storage cylinder 86.

Refrigerant line 94 interconnects the vapor outlet 88 of the cylinder 86 with a T connection 96 in the conduit 28 extending between solenoid valve SV3 and solenoid valve SV2. An additional electrically actuated solenoid valve SV1 is located in the line 94. A check valve 98 is also positioned in the conduit 28 at a location downstream of the T-96 which is adapted to allow flow in the direction from SV3 to SV2 and to prevent flow in the direction from SV2 to SV3.

With continued reference to Figure 1 a refrigerant gas contamination detection circuit 100 is included in the system in a parallel fluid flow arrangement with the compressor 44. The contamination detection circuit 100 includes an inlet conduit 102 in fluid communication with the conduit 56 extending from the oil separator 52 to the condenser inlet 58. The inlet conduit 102 has an electrically actuated solenoid valve SV6 disposed there along and from there passes to the inlet of a sampling tube holder 104. The outlet of the sampling tube holder 104 is interconnected via conduit 106 with the conduit 40 which communicates with the suction port 42 of the compressor. An electrically controlled solenoid valve SV5 is disposed in the conduit 106.

The solenoid valves SV5 and SV6, when closed, isolate the sampling tube holder 104 from the system and allow easy replacement of the sampling tube contained therein. The sampling tube holder may be of the type described in U. S. Patent 4,389,372 Portable Holder Assembly for Gas Detection Tube. Further, the refrigerant contaminant testing system is preferably of the type shown and described in detail in U. S. Patent 4,923,806 entitled Method and Apparatus For Refrigerant Testing In A Closed System and assigned to the assignee of the present invention. Each of the above identified patents is hereby incorporated herein by reference in its entirety.

Automatic control of all of the components of the refrigerant recovery system 10 is carried out by an electronic controller 108 which is formed of a micro-processor having a memory storage capability and which is micro-programmable to control the operation of all of the solenoid valves SV1 through SV7 as well as the compressor motor and the condenser fan motor. Inputs to the controller 108 include a number of measured or sensed system control parameters. In the embodiment disclosed these control parameters include the temperature of the storage cylinder Tstor which comprises a temperature transducer capable of accurately providing a signal indicative of the temperature of the refrigerant in the storage cylinder 86. Ambient temperature is measured by a temperature transducer positioned at the inlet to the condenser coil or condenser fan 62 and is referred to as Tamb. The temperature of the refrigerant flowing through the compressor discharge line 50 is sensed by a temperature transducer 110 positioned on the compressor discharge line 50.

Most important in the control scheme of the systems are the compressor suction pressure designated as P2 and the compressor discharge pressure designated as P3. As indicated in Figure 1 a pressure transducer labeled P2 is in fluid flow communication with the suction line 40 to the compressor while a second pressure transducer P3 is in fluid communication with the high pressure refrigerant line 56 passing to the condenser. The pressure ratio across the compressor 44 is defined as the ratio P3/P2. An additional input to the controller 108 is the signal from the liquid level indicator 92.

Looking now at Figure 10 it will be noted that the operating modes of the system are identified and the condition of the electrically actuatable components of the system are shown in the different modes. In the Standby mode the system has been turned on and all electrically actuatable mechanical systems are de-energized and ready for operation. In the Service mode, the electrically actuated solenoid valves SV1 through SV4 are all open thereby equalizing the pressures within the system so that it may be serviced without fear of encountering high pressure refrigerant.

The recovery and purification system 10 is capable of operating in both liquid recovery and vapor recovery modes. It should be appreciated that a vapor recovery cycle may begin under two different sets of circumstances: 1) in the case of a system containing more than five pounds of refrigerant the vapor recovery cycle will follow a previously performed liquid recovery cycle; and 2) in the case of a refrigeration system containing less than five pounds of refrigerant the vapor recover cycle represents the initiation of the recovery sequence.

Further it should be appreciated that the shift from liquid recovery to vapor recovery may be initiated by several different control schemes.

To facilitate a complete understanding of the system in its various modes of operation the system will be described first in the Vapor Recovery and Cylinder Cool modes in connection with Figures 1,2,3 and 10. Following that a combination Liquid-Vapor cycle with liquid level control will be described in connection with Figures 1,4A & B, 5 and 10. Last, a combination Liquid-Vapor cycle with control by a parameter which may be correlated to the change of state of the refrigerant being withdrawn will be described in connection with Figures 1, 6A & B, and 7-10.

The Recover and Cylinder Cool modes will now be described in detail in connection with the flow chart of Figure 2. The Recover mode is the mode in which the device 10 has been coupled to an air conditioning system 12 for removal of refrigerant therefrom. Looking now to Figure 2 it will be noted that the first step performed by the controller 108 when the Recover cycle is selected is to compare the compressor discharge pressure P3 to the compressor inlet pressure P2. If the pressure differential (P3-P2) is greater than 30 psi the controller 108 will open valves SV1-SV4 in order to equalize the pressures within the system. When the difference between P3 and P2 falls to less than 10 psi the system will then go to the Recover mode of operation. If the initial comparison of P3 and P2 shows a difference of less than or equal to 30 psi the system will go directly to the Recover mode. The reason for this comparison is that the compressor may readily start up when the pressure differential is less than or equal to 30 psi, whereas, when the pressure differential is greater than 30 psi, compressor start up is difficult and dictates a reduction in the pressure difference thereacross.

Upon initiation of the Recover mode the controller 108 will open valves SV2, SV3 and SV4, valve SV1 will remain closed. Valves SV5 and SV6 as noted in Figure 4 operate together as a single output from the micro-processor (controller) and the only time these valves are opened is when the contaminant testing process is being carried out. These valves will not be discussed further in connection with the other modes of operation of the system. The compressor 44 and the condenser fan 62 are also actuated upon initiation of the Recover mode.

Looking now at operation of the system in the Recover mode, and referring to Figure 1, with valve SV3 open refrigerant from the system being serviced 12 is forced by the pressure of the refrigerant in the system, and by the suction created by operation of the compressor 44, through conduit 20, through valve SV3, check valve 98, valve SV2 and conduit 30 to the accumulator/oil trap 32. Within the accumulator/oil trap the oil contained in the refrigerant being removed from the system being serviced falls to the bottom of the trap along with any liquid refrigerant withdrawn from the system. Gaseous refrigerant is drawn from the accumulator/oil trap 32 through the filter dryer 38 where moisture, acid and any particulate matter is removed therefrom, and, from there passes via conduit 40, through the suction accumulator 46 to the compressor 44.

The compressor 44 compresses the low pressure gaseous refrigerant entering the compressor into a high pressure gaseous refrigerant which is delivered via conduit 50 to the oil separator 52. The oil separated from the high pressure gaseous refrigerant in the separator 52 is the oil from the recovery compressor 44 and this oil is returned via conduit 54 to the suction line 40 of the compressor to assure lubrication of the compressor. From the oil separator 52 the high pressure gaseous refrigerant passes via conduit 56 to the condenser coil 60 where the hot compressed gas condenses to a liquid. Liquified refrigerant leaves the condensing coil 60 via conduit 66 and passes through the T68 through the open solenoid valve SV4, and passes via the liquid lines 80 and 82, to the refrigerant storage cylinder 86 through liquid inlet port 84.

While refrigerant recovery is going on the controller 108 is receiving signals from the pressure transducers P3 and P2, calculating the pressure ratio P3/P2, and, comparing the calculated ratio to a predetermined value. Compressor suction pressure P2 is also being looked at alone and being compared to a predetermined Recovery Termination Suction Pressure. As shown in Figure 2, the predetermined Recovery Termination Suction Pressure is 4 psia, and if P2 falls below this value the Recover mode is terminated and the controller 108 initiates the refrigerant quality test cycle, identified as Totaltest. This cycle will be described below following a complete description of the other modes of operation. Totaltest is a registered Trademark of Carrier Corporation for "Testers For Contaminants in A Refrigerant".

The selection of the predetermined recovery termination suction pressure of 4 psia results from recovery system operation wherein it has been shown that a compressor suction pressure, P2, of 4 psia or less results in recovery of 98 to 99% of the refrigerant from the system being serviced. Achieving this pressure during the first Recover mode cycle is unusual, however, it is achievable. As an example, P2 may be drawn down to the 4 psia termination value in low ambient temperature conditions where the condensing coil temperature (which is ambient air cooled) is low enough to allow P3 to remain low enough for P2 to reach 4 psia before the pressure ratio limit is reached.

Returning now to compressor pressure ratio, as indicated in Figure 2, in the illustrated embodiment, when the pressure ratio exceeds or is equal to 16 the microprocessor in the controller 108 performs what is referred to as the Recovery Cycle Test. If the Recovery Cycle just performed is the first Recovery Cycle performed and the compressor suction pressure P2 is greater than or equal to 10 psia the system will shift to what is known as a Cylinder Cool mode of operation. If the Recovery Cycle just performed is a second or subsequent recovery cycle and the compressor suction pressure P2 is less than 10 psia the controller will consider the refrigerant Recovery as completed and will initiate the refrigerant contaminant test cycle (Totaltest).

The latter conditions, i.e. second or subsequent recover cycle, and P2 less than 10 psia, are conditions that are found to exist at high ambient temperatures. For example, such conditions may exist when recovering R-22 from an air conditioning system at an ambient temperature of 105°F and above. Under such conditions it has been found that attempts to reduce the compressor suction pressure P2 to values less than 10 psia are counterproductive in that a substantial length of operating time would be necessary in order to obtain a very small additional drop in suction pressure. Further, it has been found, at these conditions, that shifting to the Cylinder Cool mode, which will be described below, also would not substantially increase the amount of refrigerant that would ultimately be withdrawn from the system and accordingly termination of the Recover mode and initiation of the refrigerant contaminant test cycle is indicated.

Assuming that the Recovery Cycle Test has indicated that either: it is the first recovery cycle, or, the compressor suction pressure P2 is greater than or equal to 10 psia, the controller 108 will initiate the Cylinder Cool mode of operation.

In the Cylinder Cool mode, as indicated in Figure 10, the solenoid valves SV1 and SV2 are energized and thereby in the open condition. Solenoid valves SV3 and SV4 are closed, and, the compressor motor and condenser fan motor continue to be energized. The Cylinder Cool mode of operation essentially converts the system to a closed cycle refrigeration system wherein the refrigerant storage cylinder 86 functions as a flooded evaporator. By closing solenoid valve SV3 the refrigerant recovery and purification system 10 is isolated from the refrigeration system 12 being serviced. The opening of solenoid valve SV1 establishes a fluid path between the vapor outlet 88 of the storage cylinder 86 and the conduit 28 which is in communication with the low pressure side of the compressor 44. The closing of solenoid valve SV4 routes the refrigerant passing from the condenser 60 through the refrigerant expansion device 74.

With the control solenoids set as described above, in the Cylinder Cooling mode of operation the compressor 44 compresses low pressure gaseous refrigerant entering the compressor and delivers a high pressure gaseous refrigerant via conduit 50 to the oil separator 52. From the oil separator 52 the high pressure gaseous refrigerant passes via conduit 56 to the condenser coil 60 where the hot compressed gas condenses to a liquid. Liquified refrigerant leaves the condensing coil 60 via conduit 66 and passes through the T-connection 68 through the strainer 76 and, via conduit 72, to the refrigerant expansion device 74. The thus condensed refrigerant, at a high pressure, flows through the expansion device 74 where the refrigerant undergoes a pressure drop, and is at least partially, flashed to a vapor. The liquid-vapor mixture then flows via conduits 78 and 82 to the refrigerant storage cylinder 86 where it evaporates and absorbs heat from the refrigerant within the cylinder 86 thereby cooling the refrigerant.

Low pressure refrigerant vapor then passes from the storage cylinder 86, via vapor outlet port 88, through conduit 94 and solenoid valve SV1 to the T connection 96. From there it passes through the check valve 98, solenoid valve SV2, oil separator/accumulator 32, filter dryer 38 and conduit 40 to return to the compressor 44, to complete the circuit.

As the Cylinder Cool mode of operation continues, the cylinder temperature, as measured by the temperature transducer Tstor, continues to drop as the refrigerant is continuously circulated through the closed refrigeration circuit. Also during this time the refrigerant is passed through the refrigeration purifying components, i.e. the oil separator 32 and the filter dryer 38, a plurality of times to thereby further purify the refrigerant.

Referring again to Figure 2, the Cylinder Cool mode of operation will terminate when any one of three conditions occur; 1) the cylinder temperature, as measured by Tstor falls to a level 70 ° F below ambient temperature (Tamb), or, 2) when the Cylinder Cooling mode of operation has gone on for a duration of 15 minutes, or, 3) when the cylinder temperature Tstor falls to 0 ° F. Regardless of which of the three conditions has triggered the termination of the Cylinder Cool mode the result is substantially the same, i.e., the temperature (Tstor) of the refrigerant stored in the cylinder 86 is now well below ambient temperature. As a result, the pressure within the cylinder, corresponding to the lowered temperature is substantially lower than any other point in the system.

When any one of the Cylinder Cool mode termination events occur, the controller 108 will shift the system to a second Recover mode of operation. In the second Recover mode the solenoid valves, and compressor and condenser motors are energized as described above in connection with the first Recover mode. Because of the low temperature Tstor that has been created in the refrigerant storage cylinder, however, the capability of the system to withdraw refrigerant from the unit being serviced, without subjecting the recovery compressor to high pressure differentials is dramatically increased.

An understanding of this phenomenon will be appreciated with reference to Figure 1. It will be described by picking up a Recover cycle at the point where refrigerant withdrawn from the system being serviced is discharged from the compressor 44 and is passing, via conduit 56, to the condenser 60. At this point the pressure within the system, extending from the compressor discharge port 48 through to and including the storage cylinder 86, is dictated by temperature and pressure conditions within the storage cylinder 86. As a result the storage cylinder 86 now effectively serves as a condenser with the recovered refrigerant passing as a super- heated vapor through the condenser coil, through the solenoid valve SV4 and the conduits 80 and 82 to the storage cylinder 86 where it is condensed to liquid form.

It is the dramatically lower compressor discharge pressure P3 experienced during a second or subsequent Recover mode (i.e. any Recover mode following a Cylinder Cool mode) that allows the recovery compressor 44 to draw the system being serviced 12 to a pressure lower than heretofore obtainable while still maintaining a permissible pressure ratio across the recovery compressor.

It will be appreciated that in a second Recover mode, the pressure ratio P3/P2 could exceed the predetermined value (which in the example given is 16) and, depending upon the other system conditions, as outlined in the flow chart of Figure 2, will result in an additional Cylinder Cool mode of operation or termination.

With continued reference to Figure 2, the system will then operate as described until conditions exist which result in the controller 108 switching to the refrigerant contaminant test (Totaltest) mode of operation. Prior to initiation of a Recover cycle an operator should make sure that a sampling tube has been placed in the sampling tube holder 104. Upon initiation of the Totaltest mode of operation, solenoid valves SV1, SV2, SV4 and SV5/SV6 are all energized to an open position. The solenoid valve SV3 is not energized and is therefore closed. With the flow control valves in the condition described the flow of refrigerant through the recovery system is similar to that described above in connection with the Cylinder Cooling mode except that the solenoid valve SV4 is open and therefore the refrigerant does not pass through the expansion device 74. With the refrigerant flowing through the circuit in this manner, and with the solenoid valves SV5 and SV6 open, the pressure differential existing between the high and low pressure side of the system induces a flow of refrigerant through conduit 102 solenoid valve SV6, the sampling tube holder 104 (and the tube contained therein), solenoid valve SV5 and conduit 106 to thereby return the refrigerant being tested to the suction side of the compressor 44.

A suitable orifice is provided in conduit 102, or in the sampling tube holder 104, to provide the necessary pressure drop to assure that the flow of refrigerant through the testing tube held in the sampling tube holder 104 is at a rate that will assure that the testing tube will receive the proper flow of refrigerant therethrough during the Totaltest run time in order to assure a reliable test of the quality of the refrigerant passing therethrough. With reference to Figure 2 will be noted that the run time of the refrigerant quality test is indicated as X minutes. The normal run time for a commercially available Totaltest system is about ten minutes and the controller may be programmed to run the test for that length of time or different time for different refrigerants. The quality test however may be terminated sooner if the refrigerant being tested contains a large amount of acid and the indicator in the test tube changes color in less than the programmed run time. If this occurs, the refrigerant quality test may be terminated, and, an additional refrigerant purification cycle initiated.

The additional purification cycle is identified as the Recycle mode and a flow chart showing the system operating logic is shown in Figure 3. With reference to Figure 4 it will be noted that the condition of the electrically actuable components is the same in Recycle as it is for the Cylinder Cool mode except that the solenoid valve SV4 is open so that the refrigerant does not flow through the expansion device 74 but flows through the open solenoid valve SV4. This increases the volume flow of refrigerant through the system during the Recycle mode. The function of this mode is strictly to further purify the refrigerant by multiple passes through the oil trap 32 and the filter dryer 38.

With reference to Figure 3 the length of time in which the system is run in the Recycle mode is determined by the operator as a number of minutes "X" which varies as a function of refrigerant type and quality and ambient air temperature. The type of refrigerant is known, the ambient temperature may be measured, and the quality is determined by the operator upon the evaluation of the test tube used in the refrigerant quality test cycle. With continued referenced to Figure 3, upon the end of the selected recycle time the system, if so selected by the operator, will run another refrigerant quality test, and, if the results of this test so indicate another recycle period may initiated following the procedure set forth above.

The object of the system and control scheme described above is to remove as much refrigerant as possible from a system being serviced, under any given ambient conditions, or system conditions, while, at all times monitoring system control parameters which will assure that the compressor of the Recovery system is not subjected to adverse operating conditions. As described above, the system control parameter is the pressure ratio P3/P2, across the recovery compressor 44. In the example given above a value of P3/P2 of 16 was used as the pressure ratio above which the compressor could be adversely affected. It should be appreciated that for different compressors the value of this parameter could be different.

The ultimate goal in the control of this system is to limit compressor operation to predetermined limits to assure long and reliable compressor life. As pointed out above, in the Background of the Invention. the internal compressor temperature is considered by compressor experts to be the controlling factor in preventing internal compressor damage during operation. In the presently disclosed preferred embodiment the pressure ratio has been found to be an extremely reliable effective control parameter which may be related to the internal compressor temperature and has thus been selected as the preferred control parameter in the above described preferred embodiment. Pressure differential, (i.e. P_3-P_2) could also be effectively used to control the system.

It should be appreciated however, that other system control parameters such as the compressor discharge temperature as measured by the temperature transducer 110 in the compressor discharge line 50, or the compressor suction pressure P2 could also be used to control the operation of the system, to limit the system to operation only at conditions at which the compressor is not adversely effected.

With respect to temperature, it is generally agreed that an internal compressor temperature at which the lubricating oil begins to break down is about 325 °F. Above this temperature adverse compressor operation and damage may be expected. In the present system the controller 108 has been programmed such that, should the compressor discharge temperature, monitored by the temperature transducer 110 exceed a maximum of 225 °F regardless of pressure ratio conditions, the system will be shut off.

It is further contemplated that, if the compressor discharge temperature, as measured at the transducer 110 were used as the primary system control parameter that a temperature in the neighborhood of 200 °F would be used to switch the recovery system from a Recover mode to a Cylinder Cooling mode of operation in order to assure that the compressor would not be adversely affected during operation of the system.

According to another embodiment of the invention, as mentioned above, the system control parameter being sensed for compressor protection could be the compressor suction pressure P2. In this case the microprocessor of the controller 108 would be programmed with compressor suction pressures P2 which would be considered indicative of adverse compressor operation, for a range of ambient air temperatures and for the different refrigerants which may be processed by the system. As an example, when processing refrigerant R-22 at an ambient air temperature of 90 °F a suction pressure P2 in the range of 13 psia to 15 psia would be programmed to change the system from a Recover mode to a Cylinder Cooling mode of operation.

The outstanding refrigerant recovery capability of a system according to the present invention is reflected in the following example. The recovery apparatus was connected to a refrigeration system having a system charge of 4.5 pounds of refrigerant R-12 at an ambient temperature of 70 ° F. Such a system is typical of an automobile air conditioning system.

Upon initiation of recovery the system performed a first Recover cycle for 8.67 minutes before the system reached the limiting pressure ratio P_2/P_3 of 16. At that point 3.73 pounds had been recovered from the system. This represents 82.9% of the systems total charge. Typical prior art systems would stop at this point, leaving .77 pounds, or more than 17% of the charge in the system. This .77 pounds would eventually be released to the atmosphere.

At this point, the system shifted to the Cylinder Cool mode of operation. The Cylinder Cool cycle ran for 15 minutes, bringing the cylinder temperature (Tstor) down to 10 ° F. At this point a second Recover cycle was initiated by the system controller. The second Recover cycle ran for 3.8 minutes at which time Recover was terminated when the suction pressure P2 fell to 4.0 psia.

At this point, the total system run time had been 27.5 minutes and a total of 4.42 pounds of refrigerant had been recovered from the system. This represents 98.2% of the total charge of 4.5 pounds, leaving only .08 pounds in the system.

Following completion of recovery and purification, the storage cylinder 86 contains clean refrigerant which may be returned to the refrigeration system. With reference to Figure 10, the Recharge mode, when selected, results in simultaneous opening of valves SV1 and SV3 to establish a direct refrigerant path from the storage cylinder 86 to the refrigeration system 12. All other valves and the compressor and condenser are de-energized in this mode. The amount of refrigerant to be delivered to the system is selected by the operator, and, the controller 108, with input from the liquid level sensor 92 will assure accurate recharge of the selected quantity of refrigerant to the system.

The liquid recovery mode will now be described in detail in connection with the flow chart of figure 4A. It should be appreciated that the liquid recovery mode is designed to be used in larger systems for example systems having a refrigerant charge of greater than 5 pounds of refrigerant. In systems where less than 5 pounds of refrigerant are contained in the system the liquid recover mode of operation may be omitted and the operator may go directly to the previously described vapor recovery mode which will be subsequently described.

At this point it is assumed that a system containing greater than 5 pounds of refrigerant is being serviced and that the device 10 has been coupled to the system 12 for removal of refrigerant therefrom. With preference now to Figure 4A and Figure 10 it will be seen that upon initiation of the Liquid Recover mode the controller 108 will open valves SV1, SV2 and SV7. The valves SV3, SV4, SV5 and SV6 will remain closed. Valves SV5 and SV6 as noted in Figure 10 operate together as a single output from the microprocessor (controller 108) and the only time these valves are open is when the contaminant testing process is being carried out. These valves will not be discussed further in connection with other modes of operation of the system. The motors of the compressor 44 and the condenser fan 62 are also energized upon initiating the liquid recover mode.

Looking now at operation of the system in the liquid recover mode, and referring to Figure 1. With valve SV3 closed and valve SV7 open refrigerant from the system being serviced 12 is forced by the pressure of the refrigerant in the system through conduit 20, through the T-11, through valve SV7 and via liquid refrigerant line 15 to the valve 90 on the refrigerant storage cylinder 86 and directly into refrigerant storage cylinder.

Upon entering the storage cylinder 86 at ambient conditions, a portion of the liquid refrigerant will exist in gaseous form. At this time because, the solenoid valve SV1 is open, a fluid path is directly established between the vapor outlet 88 of the storage cylinder 86 and the conduit 94 which is in communication with the low pressure side of the compressor 44. With the solenoid valve SV4 closed refrigerant passing from the condensers 60 will pass through the refrigerant expansion device 74.

Accordingly with the control solenoids set as described above, during liquid recovery, the compressor 44 acts to withdraw low pressure gaseous refrigerant directly from the storage cylinder 86. This refrigerant passes via conduit 94 and T-96. through the check valve 98, valve SV2 and conduit 30 to the oil separator 32. From the oil separator it passes via conduit 36 to the filter drier 38, and thence via conduit 40 and accumulator 46 to the compressor 44 delivers high pressure gaseous refrigerant via conduit 50 to the oil separator 52. From the oil separator 52 the high pressure gaseous refrigerant passes via conduit 56 to the condenser coil 60 where the hot compressed gas condenses to a liquid.

Liquified refrigerant leaves the condensing coil 60, via conduit 66 and passes through the T-connection 68 through the strainer 76 and, via conduit 72 to the refrigerant expansion device 74. The thus condensed refrigerant, at a high pressure, flows through the expansion device 74 where the refrigerant undergoes a pressure drop, and is at least partially flashed to a vapor. The liquid-vapor mixture then flows via conduit 78 and 82 back to the refrigerant storage cylinder 86 where it evaporates and absorbs heat from the refrigerant within the cylinder 86 thereby lowering the pressure and temperature within the storage cylinder 86. As a result of the lowered temperature and pressure within the storage cylinder 86 the pressure differential between the refrigeration system being serviced 12, which is at ambient temperature, and the storage tank 86 is substantially increased and as a result the flow of liquid refrigerant through the liquid refrigerant line 15 to the storage cylinder is substantially increased.

It will be appreciated, that during this mode of operation refrigerant will continue to recirculate through the cooling and purifying circuit described above.

With reference to Figure 4A it will be seen that the liquid recovery mode is run according to the illustrated embodiment, for two minutes at which time the system is shifted to the Cylinder Cool cycle. With reference to Figure 7, the only difference between the operation of the system in the Cylinder Cool cycle and the liquid recovery cycle is that the solenoid valve SV7 is closed and the system operates in a closed circuit, as described with no connection to the system being serviced. As the Cylinder Cool mode of operation continues the cylinder temperature continues to drop as the refrigerant is continuously circulated through the closed refrigeration circuit. Also during this time the refrigerant is passed through the refrigeration purifying components, i.e. the oil separator 32 and the filter dryer 38, a plurality of times to thereby further purify the refrigerant. The system is run in the Cylinder Cool cycle for five minutes in order to assure that the temperature and pressure within the storage cylinder is reduced such that it is substantially lower than ambient temperature.

At this point, with continued reference to Figure 4A the system returns to liquid recovery operation. As the second liquid recovery cycle continues the controller 108 continues to receive the signal generated by the liquid level sensor 92 which is indicative of the liquid level within the storage cylinder 86. The processor receives a succession of these signals and determines a rate of liquid level increase in the storage cylinder 86. The processor then generates a signal indicative of the rate of liquid level increase. The processor is further programmed to look at the signal indicative of the rate of liquid level increase and determine whether that rate is commensurate with the withdrawal of liquid refrigerant from the system.

Figure 5 illustrates the decrease in the rate of refrigerant recovery, and, accordingly, the decrease in the rate of increase of the liquid level within the cylinder 86 which occurs when the recovery of refrigerant shifts from a liquid to a vapor state. The straight line portion of the graph illustrates the linear increase in the amount of refrigerant recovered as time goes by when recovery is in the liquid state. At the top of the graph where the slope changes dramatically the rate of refrigerant being recovered is in the vapor state. When the microprocessor senses the dramatic change in the rate that refrigerant is being recovered the liquid recovery mode of operation is automatically terminated.

The accuracy of the information which liquid level sensors are able to provide varies widely. The operation of the Liquid Recovery system as described above is such that the system will perform a successful recovery using a level sensor that provides less accurate readings. In a system using an extremely accurate level sensor the Liquid Recovery mode of operation described above, as outlined in Figure 4A, may be performed by omitting the first Cylinder Cool cycle and the return to Liquid Recovery cycle.

With reference to Figure 4A it will be seen that at this point the system shifts to a Cylinder Cool cycle of operation in order to reduce the temperature and pressure of the storage cylinder 86 prior to the beginning of a vapor recovery cycle. With continued reference to Figure 4A, this Cylinder Cool mode of operation will terminate when any one of three conditions occur; 1) the cylinder temperature, as measured by Tstor falls to a level 70°F below ambient temperature (Tamb), or, 2) when the cylinder cool mode of operation has gone for a duration of 15 minutes, or, 3) when the cylinder temperature Tstor falls to 0°F. Regardless of which of the three conditions triggers termination of the Cylinder Cool mode, the result is substantially the same, i.e., the temperature (Tstor) of the refrigerant stored in the cylinder 86 is well below ambient temperature. At this point the system will shift to a vapor recovery mode of operation to complete the withdrawal of the refrigerant from the system being serviced.

The Vapor Recover and Cylinder Cool modes of operation are illustrated in the flow chart of Figure 4B. The operation of the system at this time is the same as the previously described Vapor Recovery and Cylinder Cool cycles and will not be repeated.

Figures 6A and B illustrate the operation of the system in the liquid recovery mode where the shift from liquid recovery to vapor recovery is controlled by a parameter which may be correlated to the change of state of the refrigerant being withdrawn. Operation of the system is the same as that described in connection with Figure 4A and B except for the source of the control signals.

As the second liquid recovery cycle continues, the controller 108 continues to receive signals related to a number of conditions within the system. Specifically the temperature transducer Tstor provides a signal indicative of the temperature of the refrigerant in the storage cylinder 86. The pressure transducer P2 and P3, provide information with respect to the pressure entering and leaving, respectively the compressor 44. These three parameters will collectively be referred to as system control parameters.

Figures 7, 8 and 9 illustrate the value of the system control parameters Tstor, P3 and P2 respectively as a function of the length of time the liquid recovery cycle has been run. With respect to each of these graphical representations it will be noted that at the seven minute mark each of the parameters increases, then stabilizes and then begins to drop. The beginning of the increase of each of the parameters, i.e. the seven minute point represents the beginning of the second liquid recovery cycle. The point at which each of theses parameters begins to drop has been found to be correlatable with the time at which the state of the refrigerant being withdrawn from the refrigeration system 12 changes from a liquid state to a vapor state. The microprocessor of the controller 108 is programmed to terminate the recovery mode of operation automatically when one of these selected system control parameters falls a predetermined amount below its maximum value. As noted in Figure 6A Tstor is the preferred controlled parameter and in the preferred embodiment the termination of liquid recovery occurs when Tstor drops 5°F from its maximum value. In the case of the control parameter being P2 or P3 a drop of 5 psi from the maximum value has been found to cause the shift from liquid recovery to vapor recovery to occur at an appropriate time.

With reference to Figure 6A it will be seen that at this point the system shifts to a Cylinder Cool cycle of operation in order to reduce the temperature and pressure of the storage cylinder 86 prior to the beginning of a vapor recovery cycle. With continued reference to Figure 6A, this Cylinder Cool mode of operation will terminate when any one of three conditions occur; 1) the cylinder temperature, as measured by Tstor falls to a level 70°F below ambient temperature (Tamb), or, 2) when the cylinder cool mode of operation has gone for a duration of 15 minutes, or, 3) when the cylinder temperature Tstor falls to 0°F. Regardless of which of the three conditions triggers termination of the Cylinder Cool mode, the result is substantially the same, i.e., the temperature (Tstor) of the refrigerant stored in the cylinder 86 is well below ambient temperature. At this point the system will shift to a vapor recovery mode of operation to complete the withdrawal of the refrigerant from the system being serviced.

Claims

1. Apparatus of the type for recovering compressible refrigerant from a refrigeration system including;

compressor means (44) for compressing gaseous refrigerant delivered thereto, said compressor means having a suction port (42) and a discharge port (48);

first conduit means (20,30,36,40) for connecting the refrigeration system to said suction port of said compressor means;

condenser means (60) for passing refrigerant therethrough, said condenser means having an inlet (58) and an outlet (64);

second conduit means (54,56) for connecting said discharge port of said compressor means with said inlet of said condenser means;

means for storing refrigerant (86);

third conduit means (66,82) for connecting said outlet of said condenser means with said means for storing refrigerant; wherein the improvement comprises:

fourth conduit means (94) for connecting said means for storing refrigerant with said first conduit means;

first valve means (SV3) operable between open and shut conditions and disposed in said first conduit means upstream from the connection of said fourth conduit means with said first conduit means;

second valve means (SV4) operable between an open condition and an refrigerant expanding condition and disposed in said third conduit means;

third valve means (SV1) operable between open and shut conditions and disposed in said fourth conduit means;

means for sensing a system control parameter (110, P2, P3) related to protection of said compressor, and, for providing a signal having a value indicative of the sensed system control parameter; and

processor means (108) for receiving said signal provided by said sensing means, and, for operating said first valve means (SV3) to an open condition, said second valve (SV4) means to an open condition, and, said third valve means (SV1) to a closed condition in response to said signal having a value within a predetermined range at which said compressor is not adversely effected; and, for operating said first valve means (SV3) to a shut condition, said second valve means (SV4) to a refrigerant expanding condition, and, said third valve means (SV1) to an open condition in response to said signal having a value within a predetermined range at which said compressor is adversely effected.

2. The apparatus of claim 1 wherein said means for sensing a system control parameter comprises;

a first pressure transducer means (P2) for sensing the pressure of refrigerant entering said compressor (44) and for providing a first pressure signal indicative of this pressure; and

a second pressure transducer means (P3) for sensing the pressure of refrigerant leaving said compressor and for providing a second pressure signal indicative of this pressure;

means for processing said first pressure signal and said second pressure signal to determine the pressure ratio across said compressor and provide said signal having a value indicative of the sensed control parameter, said sensed system control parameter being compressor pressure ratio.

3. The apparatus of claim 1 including:

means for purifying (32,38) the refrigerant passing through said first conduit means from the refrigeration system to said compressor means;

fifth conduit means (102) having one end thereof in fluid communication with said second conduit means for allowing withdrawal of a quantity of refrigerant therefrom;

sixth conduit means (106) having one end in fluid communication with said first conduit means downstream from said means for purifying;

means (104) for operably supporting a refrigerant purity sampling tube in sealed fluid flow communication with the other ends of said fifth (102) and sixth (106) conduit means to thereby establish a fluid flow interconnection therebetween;

second valve means (SV6) operable between an open and shut condition disposed in said fifth conduit means;

third valve means (SV5) operable between open and shut conditions and disposed in said sixth conduit;

fourth valve means (SV1) operable between open and shut conditions and disposed in said fourth conduit;

means (108) for energizing said compressor and for actuating said first valve means (SV3) to an open condition and said second, third and fourth valve means (SV6), (SV5), (SV1) to a closed condition to thereby operate the system in a refrigerant recovery mode;

means (108) for energizing said compressor (44) and for operating said first valve (SV3) to a closed position, said second valve (SV6) to a closed position, said third valve (SV5) to a closed position, and said fourth valve (SV1) to an open position to thereby define a closed refrigerant circulation path defining a refrigerant purification mode of operation; and

means (108) for energizing said compressor (44), operating said first valve (SV3) to a closed position, operating said fourth valve (SV1) to an open position, operating said second valve (SV6) to an open position and operating said third valve (SV5) to an open position to thereby define a refrigerant circuit whereby refrigerant is passed through said refrigerant purity sampling tube in a refrigerant quality test mode of operation.

4. The apparatus of claim 1, further including;

fifth conduit means (15) for connecting the refrigeration system (12) to said means for storing refrigerant (86);

fourth valve means (SV7) operable between open and closed conditions and disposed in said fifth conduit means (15);

wherein, said fifth conduit means (15) defines a refrigerant path from said refrigerant system (12) directly to said means for storing refrigerant (86);

wherein, when said first valve means (SV3) is closed, and, both said fourth valve means (SV1) and said fifth valve means (SV7) are open, refrigerant flowing from said means for storing refrigerant (86), through said fourth conduit means (94,30,40), through said compressor means (44), through said second conduit means (54,56), through said condenser means (60), through said third conduit means (66,82), and, said expansion means (74) disposed therein, back to said means for storing refrigerant (86), defines a cooling circuit that reduces the temperature and pressure within said means for storing refrigerant (86) to thereby encourage the flow of refrigerant from the refrigeration system (12) through said first conduit to said means for storing refrigerant; and

wherein, when both said first valve means (SV7) and said fourth valve means (SV7) are closed, and, said third valve means (SV1) is open, said apparatus for recovering will be isolated from the refrigeration system (12) and said cooling circuit will operate in a closed circuit to lower the temperature and pressure within the apparatus for recovering.

5. The apparatus of claim 4, further including means for sensing the level of liquid (92) within said means for storing refrigerant (86) and for generating a signal indicative of the liquid level within said means for storing refrigerant;

processor means (108) for receiving a succession of said signals indicative of liquid level and for determining a rate of liquid level increase within said means for storing refrigerant and for generating a signal indicative of the rate of liquid level increase;

processor means (108) for receiving the signal indicative of the rate of liquid level increase and for operating said first valve means (SV7) to an open condition and said second valve means (SV3) to a closed condition in response to said rate of liquid level increase exceeding a predetermined value of rate of liquid level increase which is indicative of the recovery of liquid refrigerant from the refrigeration system (12), and, for operating said first valve means (SV7) to a closed condition and said second valve means (SV3) to an open condition in response to said signal indicative of rate of liquid level increase falling below said predetermined value of the rate of liquid level increase which is indicative of the recovery of liquid from the refrigeration system.

6. The apparatus of claim 4, further including;

means for sensing a system control parameter (P2, P3, Tstor) which has a detectable change in value which occurs at a time which may be correlated with the time at which the state of the refrigerant being withdrawn from the refrigeration system (12) changes from liquid to vapor, and, for providing a signal indicating that the detectable change has occurred;

means (108) for operating said first valve means (SV7) to an open position, and, said second valve means (SV3) to a shut position, wherein, the apparatus will operate to withdraw liquid refrigerant from the refrigeration system through said first conduit; and,

for operating said first valve means (SV7)-to a shut position and said second valve means (SV3) to an open position to withdraw gaseous refrigerant from the refrigeration system through said fifth conduit in response to said signal indicating that the detectable change has occurred.

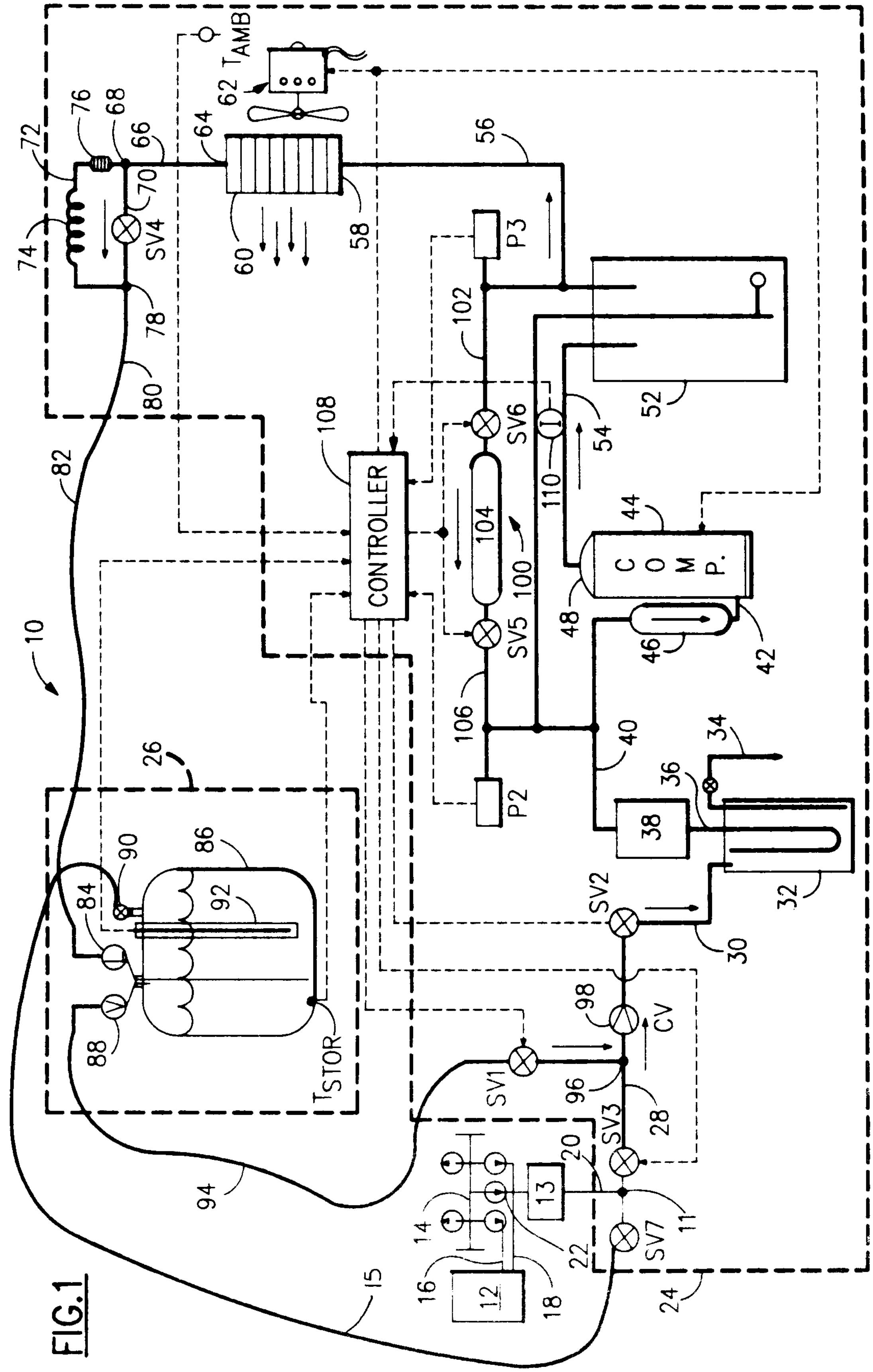

CONTROLLER
C O M P.
SV1
SV2
SV3
SV4
SV5
SV6
SV7
CV
P2
P3
T AMB
T STOR

FIG.1

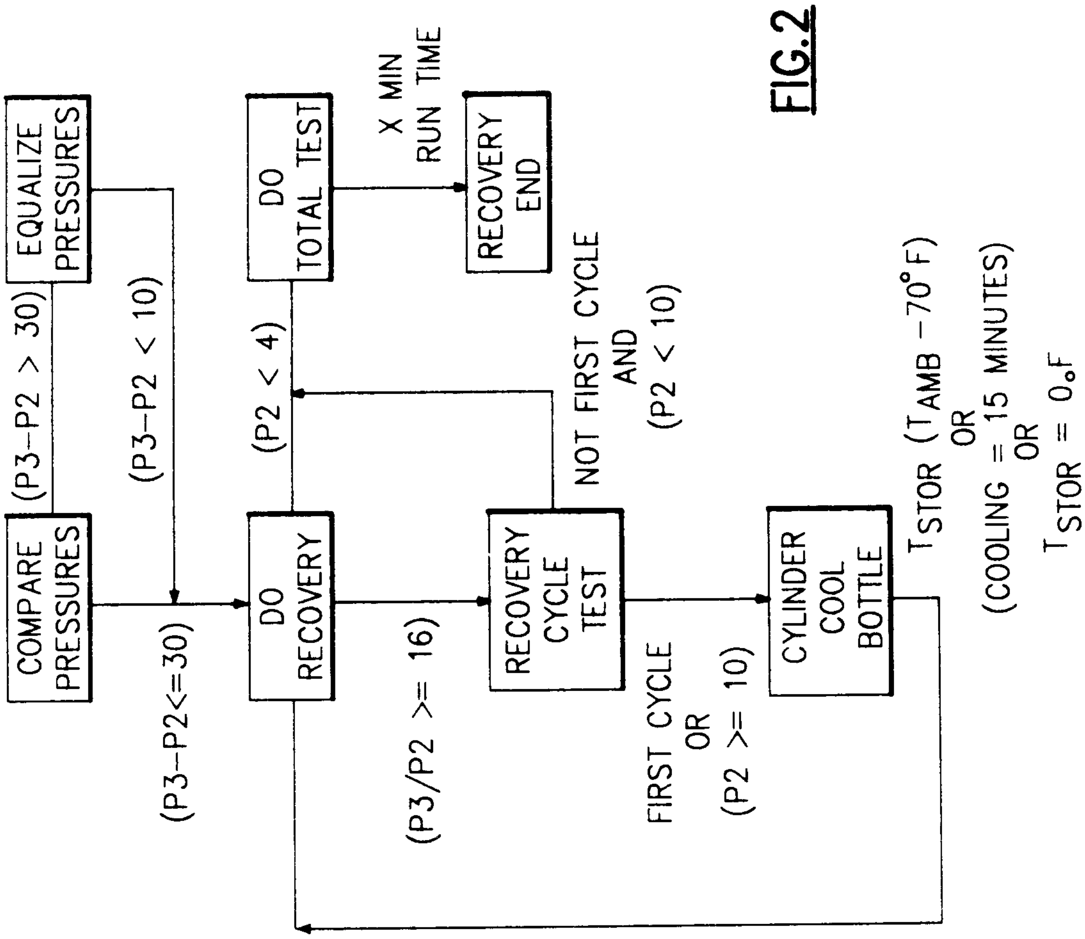

COMPARE PRESSURES
EQUALIZE PRESSURES
(P3–P2 > 30)
(P3–P2 < 10)
(P3–P2<=30)
DO RECOVERY
DO TOTAL TEST
(P2 < 4)
X MIN RUN TIME
RECOVERY END
(P3/P2 >= 16)
RECOVERY CYCLE TEST
NOT FIRST CYCLE AND (P2 < 10)
FIRST CYCLE OR (P2 >= 10)
CYLINDER COOL BOTTLE
TSTOR (TAMB –70° F)
OR
(COOLING = 15 MINUTES)
OR
TSTOR = 0°F

FIG.2

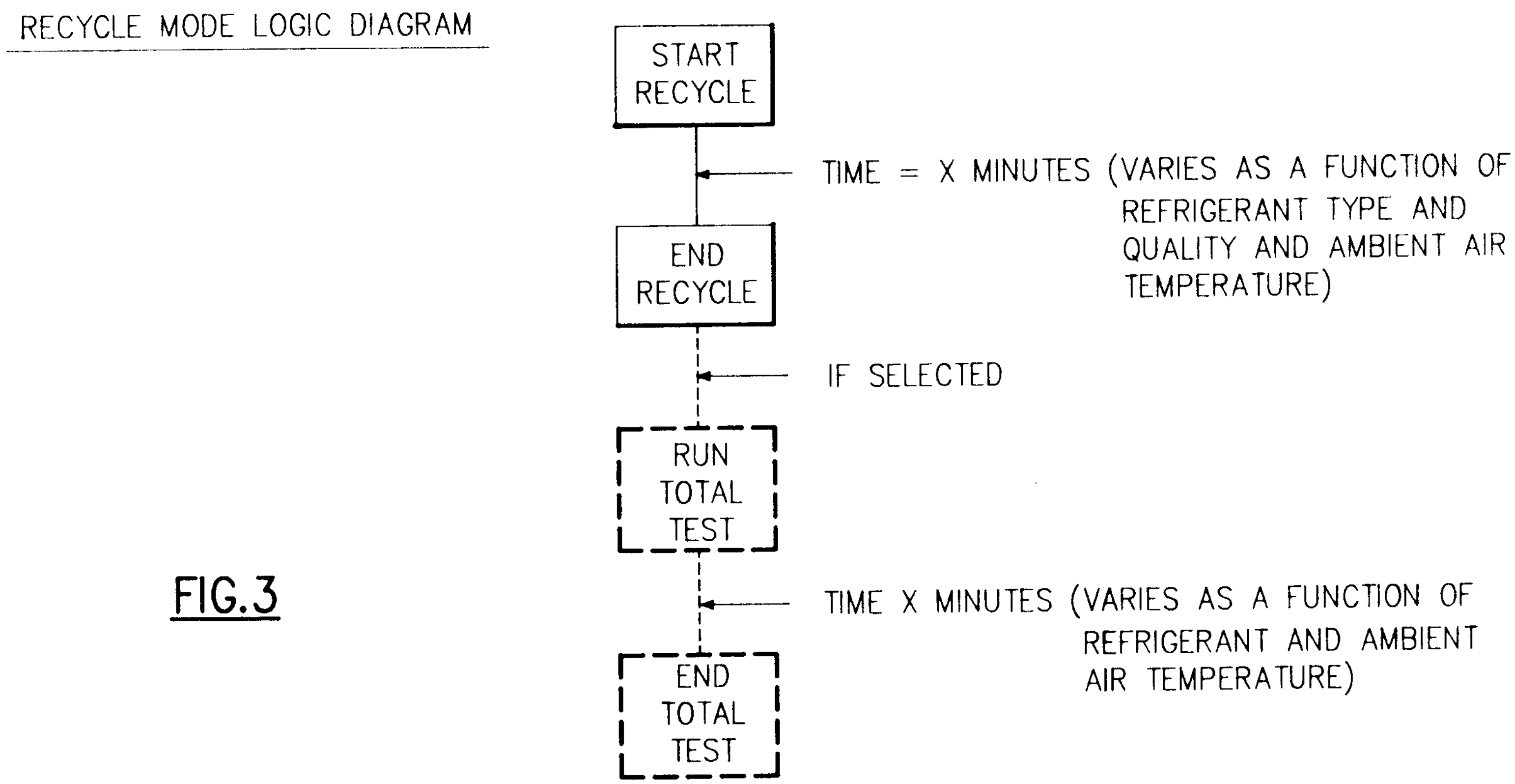

RECYCLE MODE LOGIC DIAGRAM
START RECYCLE
TIME = X MINUTES (VARIES AS A FUNCTION OF REFRIGERANT TYPE AND QUALITY AND AMBIENT AIR TEMPERATURE)
END RECYCLE
IF SELECTED
RUN TOTAL TEST
TIME X MINUTES (VARIES AS A FUNCTION OF REFRIGERANT AND AMBIENT AIR TEMPERATURE)
END TOTAL TEST

FIG.3

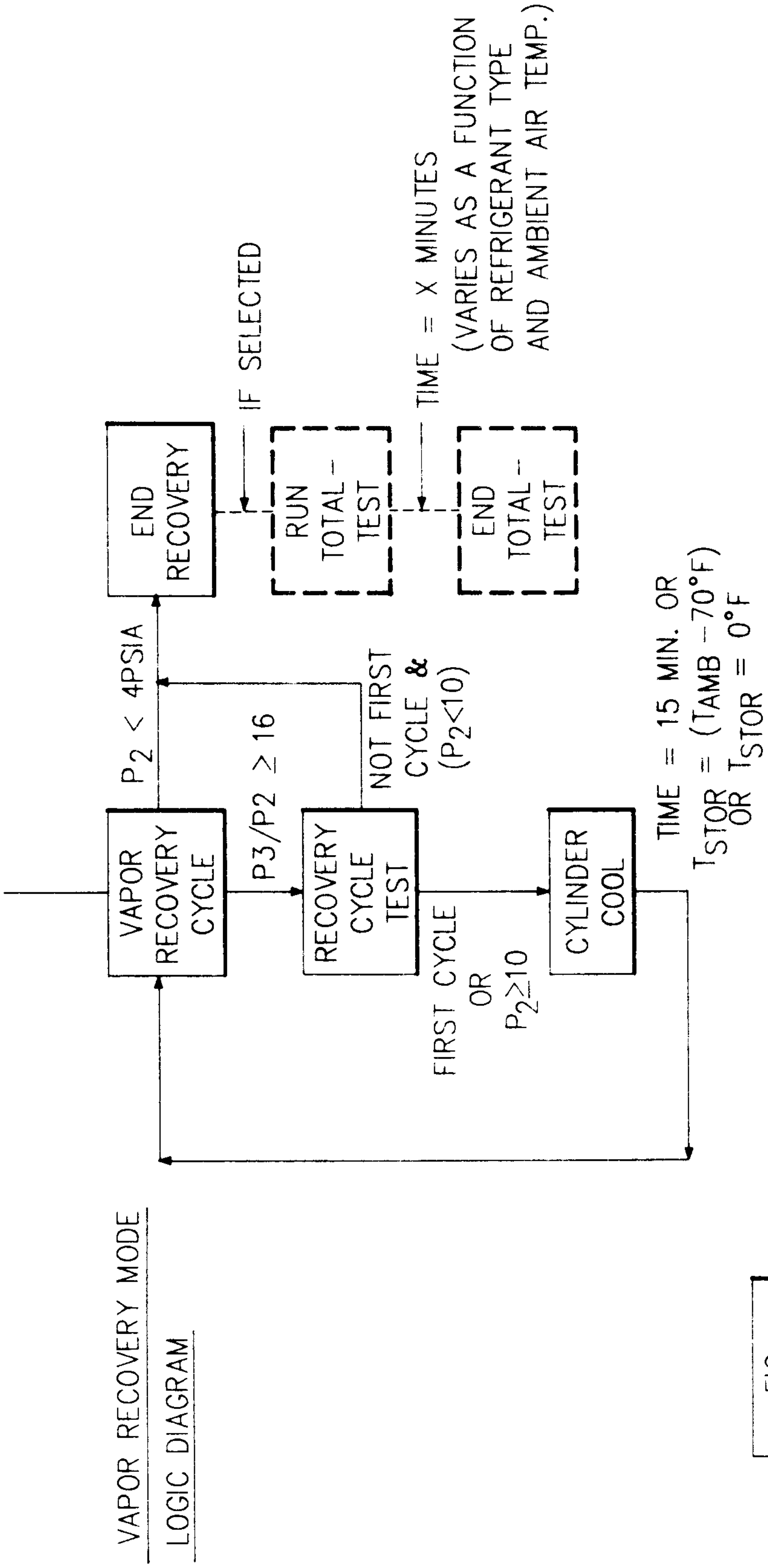

VAPOR RECOVERY MODE
LOGIC DIAGRAM
VAPOR RECOVERY CYCLE
P3/P2 ≥ 16
RECOVERY CYCLE TEST
FIRST CYCLE OR P2≥10
CYLINDER COOL
TIME = 15 MIN. OR
TSTOR = (TAMB −70°F)
OR TSTOR = 0°F
NOT FIRST CYCLE & (P2<10)
P2 < 4PSIA
END RECOVERY
IF SELECTED
RUN TOTAL-TEST
TIME = X MINUTES
(VARIES AS A FUNCTION
OF REFRIGERANT TYPE
AND AMBIENT AIR TEMP.)
END TOTAL-TEST

FIG.4B

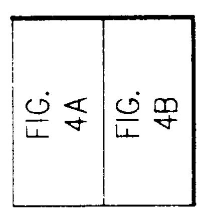


FIG. 4A
FIG. 4B

FIG.4

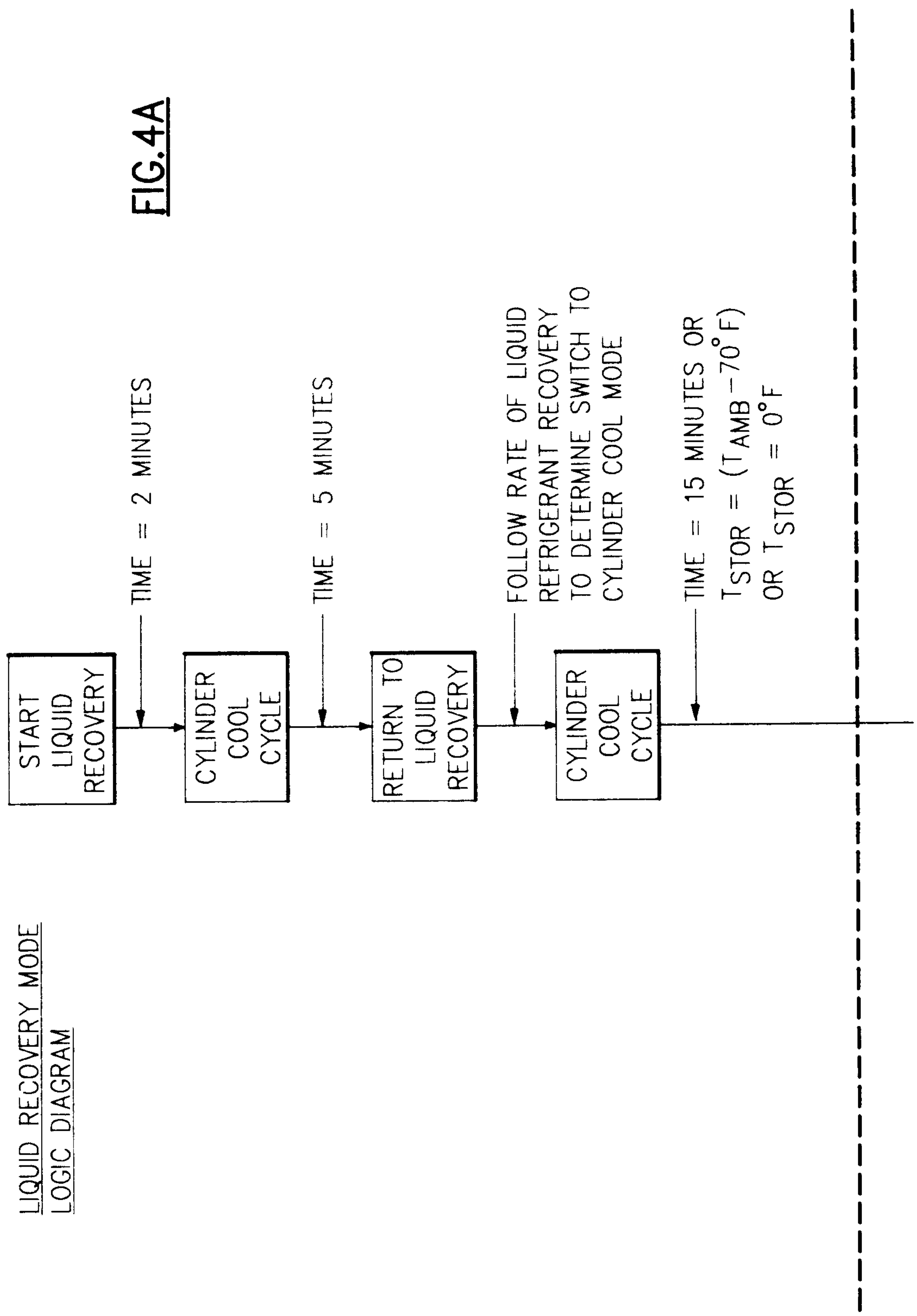
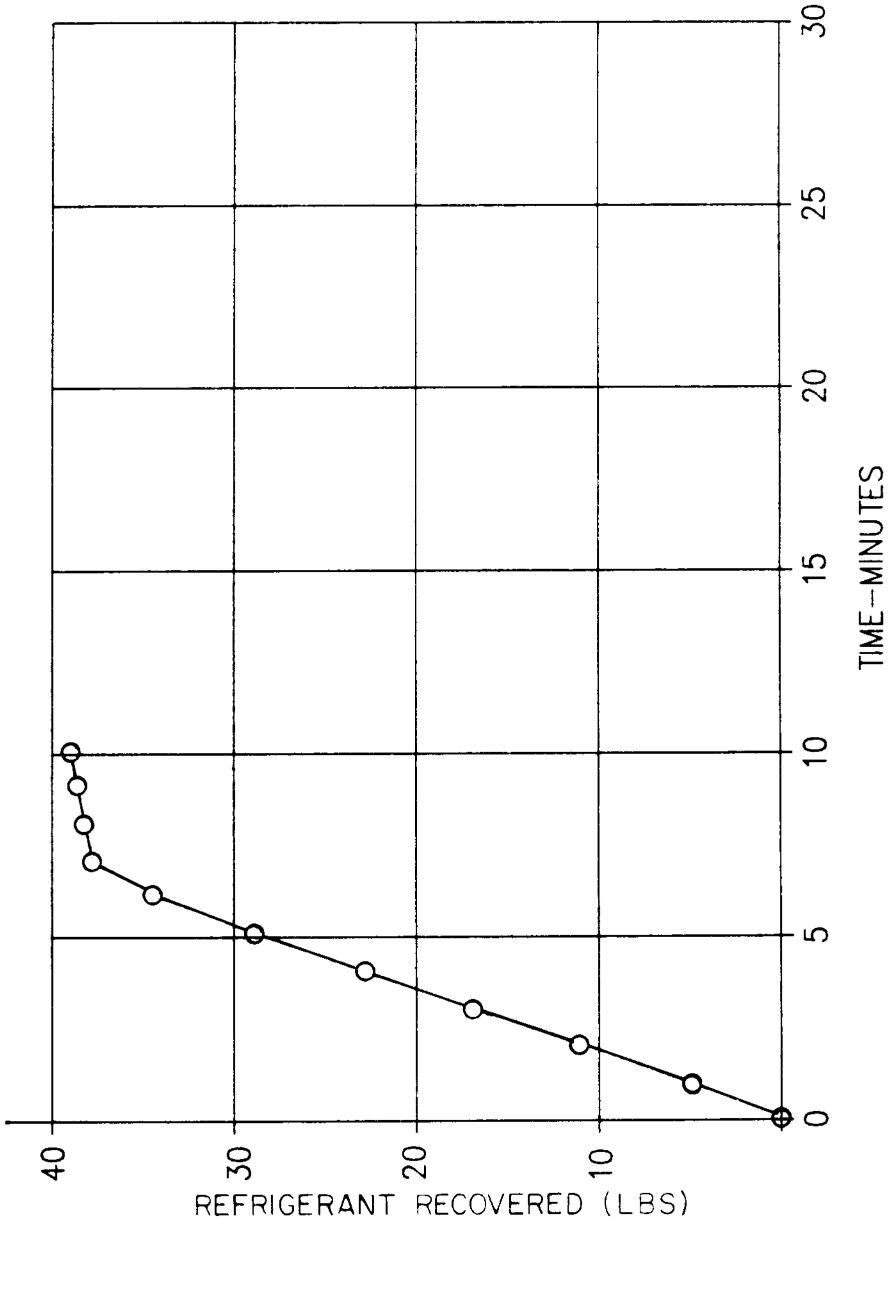

LIQUID RECOVERY MODE
LOGIC DIAGRAM
START LIQUID RECOVERY
TIME = 2 MINUTES
CYLINDER COOL CYCLE
TIME = 5 MINUTES
RETURN TO LIQUID RECOVERY
FOLLOW RATE OF LIQUID REFRIGERANT RECOVERY TO DETERMINE SWITCH TO CYLINDER COOL MODE
CYLINDER COOL CYCLE
TIME = 15 MINUTES OR
$T_{STOR} = (T_{AMB} - 70°F)$
OR $T_{STOR} = 0°F$

FIG.4A



TIME--MINUTES
0
5
10
15
20
25
30
REFRIGERANT RECOVERED (LBS)
10
20
30
40

FIG.5

VAPOR RECOVERY MODE

LOGIC DIAGRAM

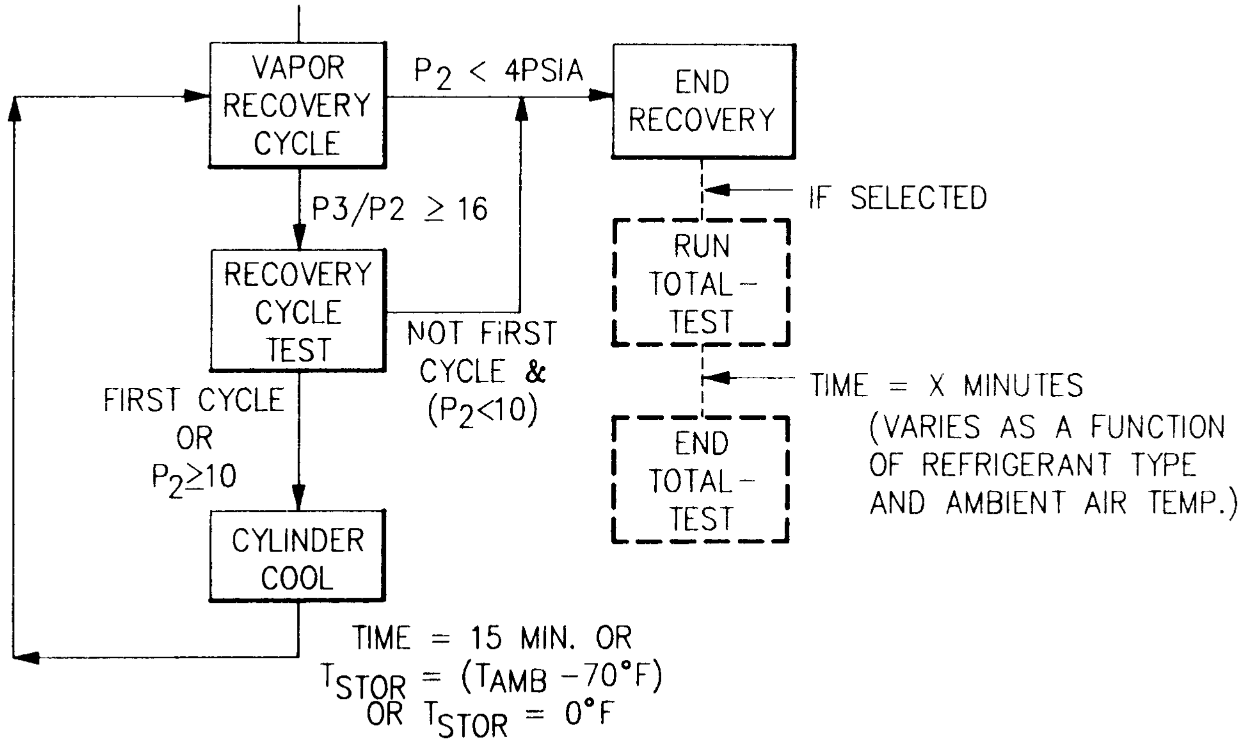


FIG.6B

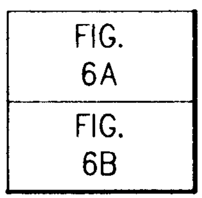


FIG.6

LIQUID RECOVERY MODE
LOGIC DIAGRAM

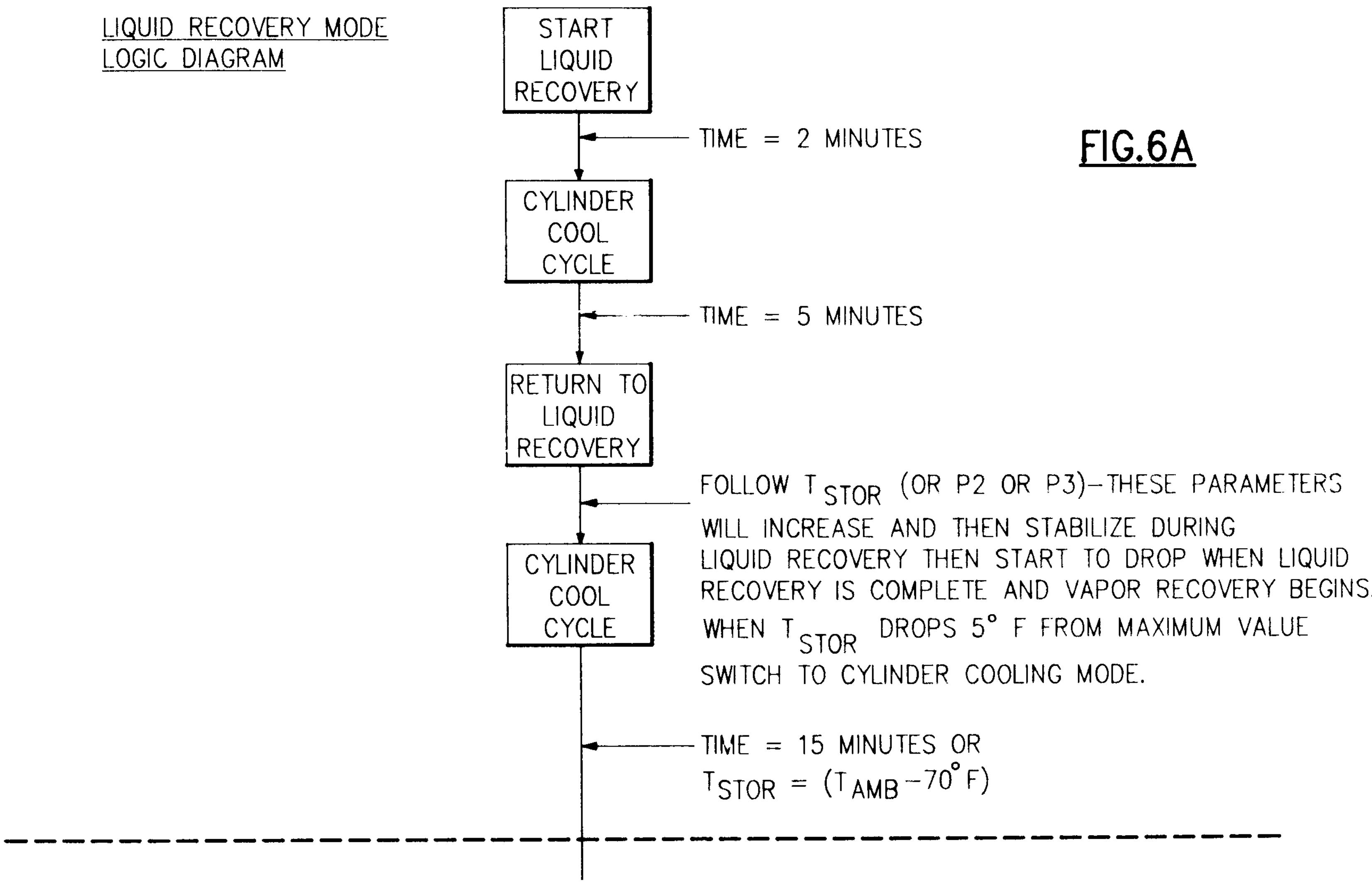


FIG.6A

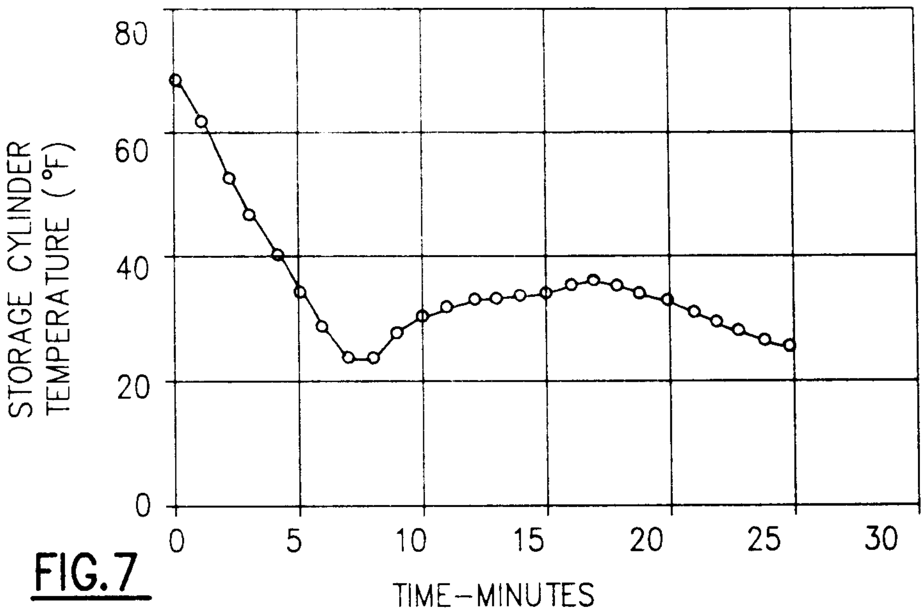


FIG.7

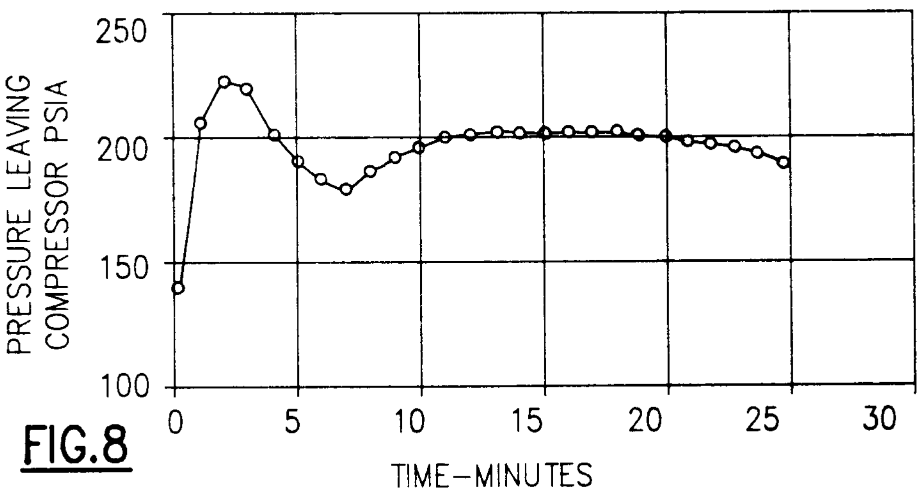


FIG.8

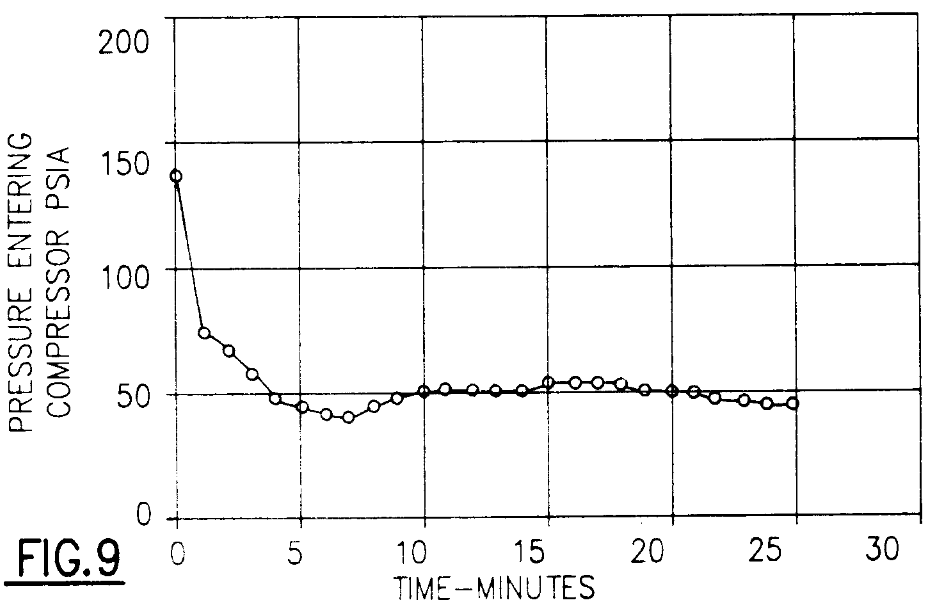


FIG.9

REFRIGERANT RECOVERY/RECYCLE
UNIT COMPONENT/MODE CHART

MODE	COMPONENT						
	SV1	SV2	SV3	SV4	SV5/SV6	SV7	COMPRESSOR/COND FAN
STANDBY	CL	CL	CL	CL	CL	CL	OFF
SERVICE	OP	OP	OP	OP	CL	OP	OFF
RECOVERY (LIQUID)	OP	OP	CL	CL	CL	OP	ON
RECOVERY (VAPOR)	CL	OP	OP	OP	CL	CL	ON
CYLINDER COOL	OP	OP	CL	CL	CL	CL	ON
RECYCLE	OP	OP	CL	OP	CL	CL	ON
TOTALEST	OP	OP	CL	OP	OP	CL	ON
RECHARGE	CL	CL	CL	CL	CL	OP	OFF

NOTES:

SOLENOID VALVES SV5 AND SV6 OPERATE TOGETHER AS A SINGLE OUTPUT FROM MICROPROCESSOR.

COMPRESSOR MOTOR/COND FAN MOTOR OPERATE TOGETHER AS A SINGLE OUTPUT FROM MICROPROCESSOR.

OP = OPEN (ENERGIZED)
CL = CLOSED (DE-ENERGIZED)
ON = ENERGIZED
OFF = DE-ENERGIZED

FIG.10

European Patent Office

EUROPEAN SEARCH REPORT

Application Number

EP 91 11 8875

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int. Cl.5)
A	WO-A-8 912 792 (MURRAY) * page 4, line 14 - page 18, line 32; figure *	1-6	F25B45/00
A	FR-A-2 645 948 (MATHIEU) * page 4, line 29 - page 9, line 2; figure *	1-6	
A	US-A-4 285 206 (KOSER) * column 3, line 48 - column 8, line 56; figures *	1-6	
D,A	US-A-4 923 806 (KLODOWSKI) * column 2, line 67 - column 11, line 6; figures *	3	

TECHNICAL FIELDS SEARCHED (Int. Cl.5)

F25B

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
THE HAGUE	19 DECEMBER 1991	BROMAN B.T.

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
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