Jan. 4, 1966 E. A. GAMACHE 3,226,949

MULTI-ZONE REFRIGERATION SYSTEM AND APPARATUS

Filed May 5, 1964 3 Sheets-Sheet 1

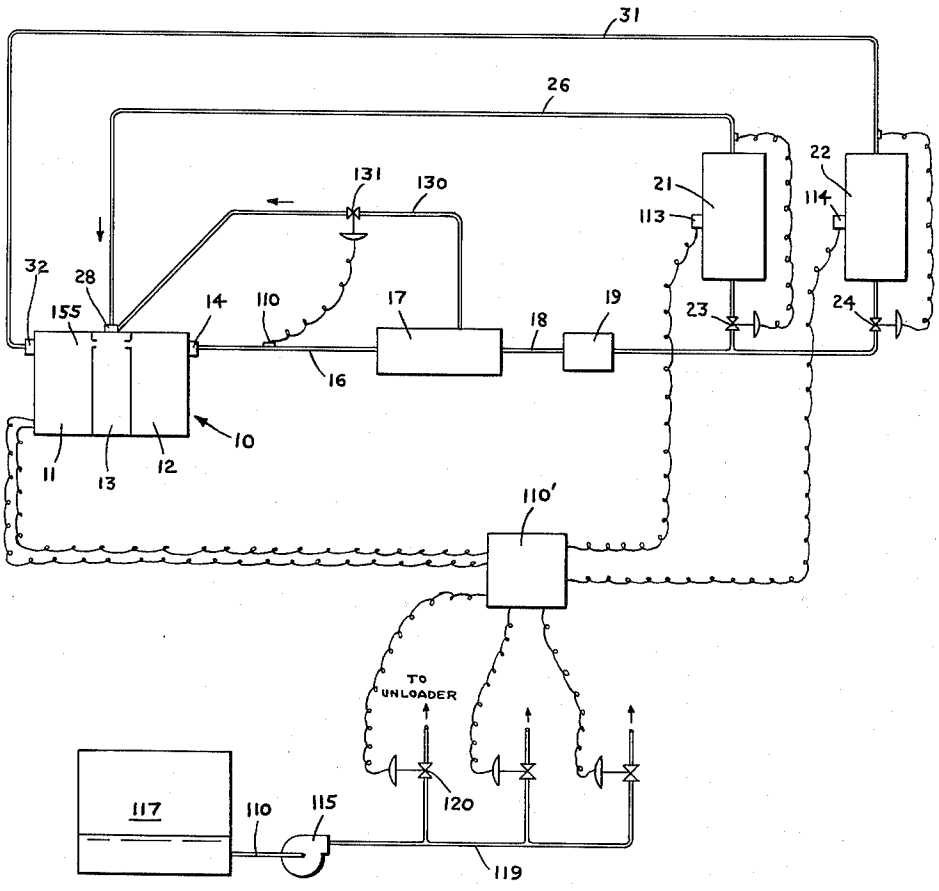


FIG. 1

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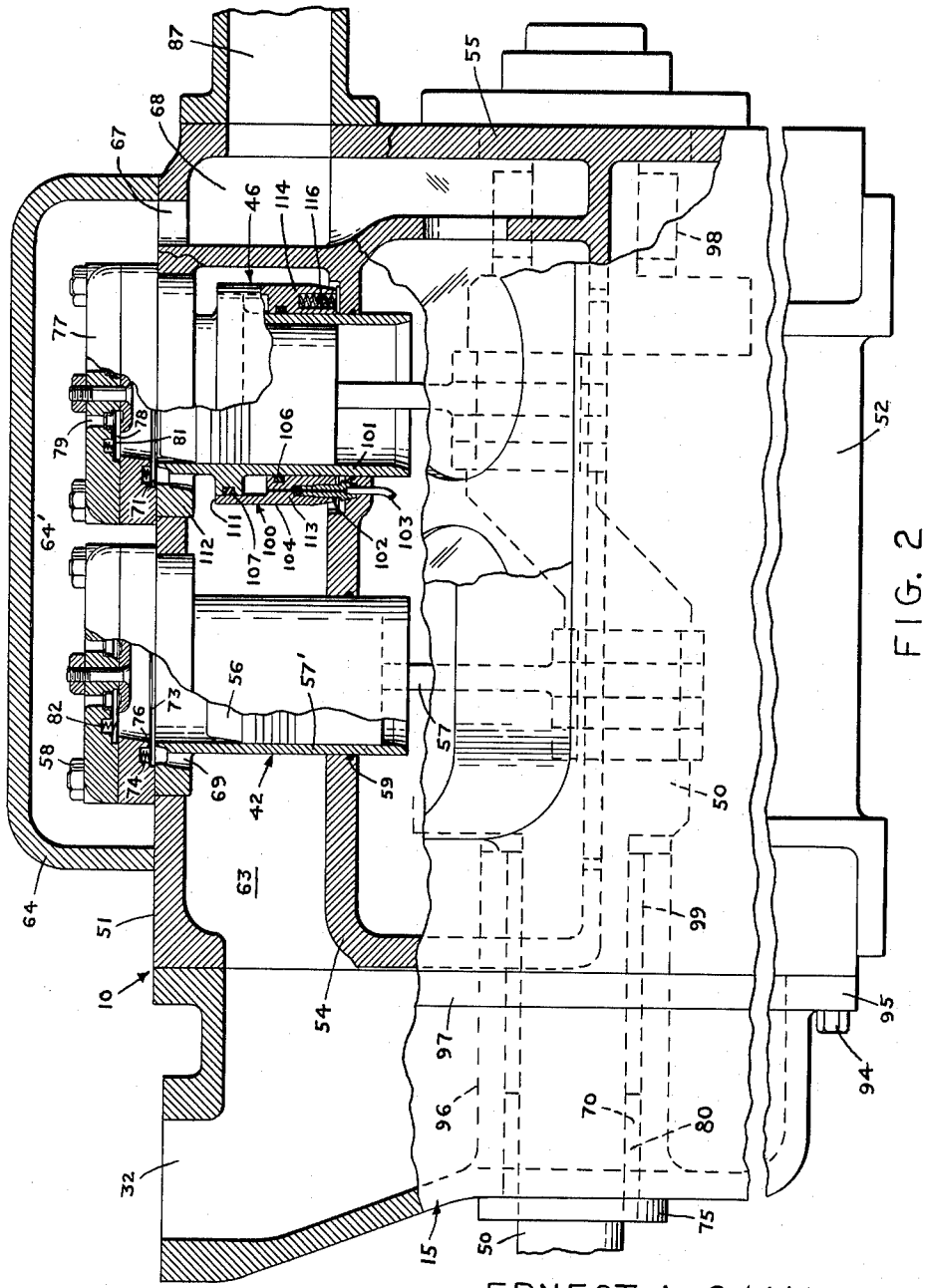


FIG. 2

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MULTI-ZONE REFRIGERATION SYSTEM AND APPARATUS

Filed May 5, 1964 3 Sheets-Sheet 3

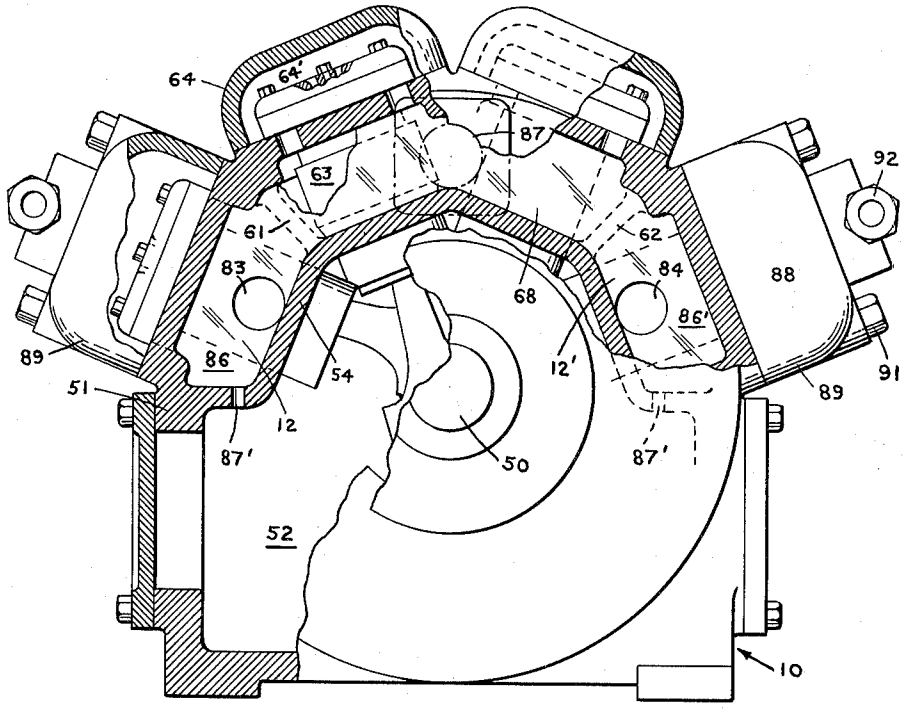


FIG. 4

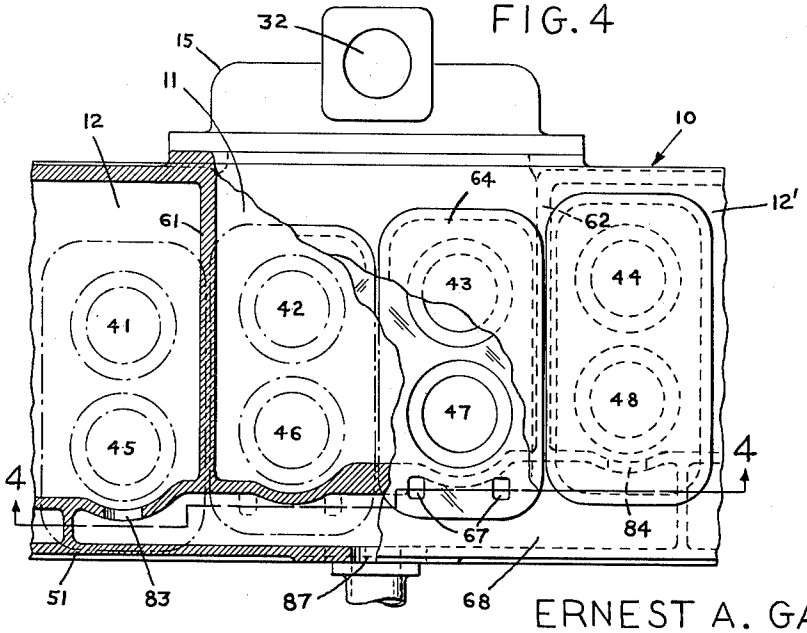


FIG. 3

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Patented Jan. 4, 1966

3,226,949

MULTI-ZONE REFRIGERATION SYSTEM AND APPARATUS

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6 Claims. (Cl. 62—510)

This invention relates to a refrigeration system operable to simultaneously service a plurality of cooling zones, exhibiting widely varying temperatures. It relates in particular to a multi-stage compressor adapted to provide hot compressed refrigerant to said system.

In a normal commercial refrigeration system, a refrigerant is circulated in liquid and vaporized phases through elements making up the system. At the system evaporator outlet, refrigerant gas at a predetermined pressure is directed to the compressor suction inlet for recirculation.

In the instance of a multi-zone installation where cooling zones are maintained at widely divergent temperatures one from the other, returning refrigerant gas streams will exhibit a comparable divergent pressure differential. For example, in a supermarket or similar installation handling food stuffs, at least one refrigerated section may be required to operate at a temperature of about 15° F. Other sections holding foods in cold storage, may require a constant temperature of as low as minus 40° F. depending on the character of the food being stored.

It is known to be economically inefficient to equalize the pressure of refrigerant gas streams returning to a single compressor from evaporating zones operating at divergent pressures. To accommodate such conditions it is customary to employ separate compressors, or compressors connected in tandem to receive refrigerant gas at a single low pressure, and to discharge the same as a hot stream at a higher pressure.

The present invention provides an improved, closed multi-zone refrigeration system of the type contemplated in which at least two different temperature cooling zones are simultaneously maintained. A multi-stage compressor incorporated into the system feeds hot compressed gas to a condenser, said compressor being adapted to receive streams of refrigerant gas at pressures substantially equal to the pressures of the system's respective evaporators. A first compressor stage receives a vaporous stream from the low temperature and pressure zone for compressing the same to an intermediate pressure. A higher second compression stage receives hot, partially compressed gas, together with a second stream of refrigerant gas from the higher pressure evaporators. The second stream mixes with and cools the first stage discharge gas and also supplements the gas feed to the second stage.

Unloader means cooperative with the respective compressor stages serve to adjust the capacity of each stage in accordance with the loading conditions at the respective cooling zones. Also provided in the compressor is means for introducing liquid refrigerant from the system directly into the compressor intermediate stage, supplementing and cooling the second refrigerant stream as required.

It is therefore, an object of the invention to provide a refrigeration system having a plurality of evaporator units adapted to pass streams of refrigerant at varying pressures to provide varying degrees of cooling at different cooling zones.

Another object is to provide a refrigeration system operable at a plurality of pressure levels and having a refrigerant compressor applicable in the system.

A still further object is to provide a refrigeration system simultaneously operable at a plurality of cooling levels and delivering vaporized refrigerant streams at a

first pressure and at a second higher pressure, said system including a single multi-stage compressor for receiving vaporized refrigerant at these varying pressures for compressing the same.

Still another object is to provide a multi-stage compressor having a plurality of inlet means being maintained at different refrigerant pressures.

Another object is to provide a reciprocating compressor adapted for multi-stage operation, in a refrigeration system, and including a first stage operable at a low pressure, and being communicated with a second stage operable at higher pressure, each of said stages being furnished with separate flows of refrigerant drawn from the system.

Another object of the invention is to provide a multi-stage compressor embodying interconnected first and second stages having substantially equal volumetric capacity, said first stage discharging into an intermediate chamber adapted to receive another stream of vaporized refrigerant to mix with vapors passing from the first low pressure stages to said second stage.

These and other objects of the invention will become clear to those skilled in the art from the following description of both the system and the apparatus, made in conjunction with the appended drawings.

FIGURE 1 illustrates schematically a refrigerant system and unloader circuit including a plurality of evaporators operable at different pressure levels to provide cooling zones of divergent temperatures.

FIGURE 2 is a side elevation in partial cross section of a multi-stage compressor adapted for use in the system shown in FIGURE 1.

FIGURE 3 is a segmentary view of a portion of the top of the compressor shown in FIGURE 2 with a portion of the wall broken away to show internal structure.

FIGURE 4 is a segmentary view partially in cross section taken along line 4—4 of FIGURE 3.

FIGURE 1 depicts schematically a refrigeration system of the type presently contemplated which includes a common condenser and a plurality of evaporators operable at different pressures and temperatures. While the figure illustrates two evaporators, it is understood that added units may be employed in the system and be so connected to operate in banks or other suitable arrangements of pressures and temperatures. The closed system includes a compressor of the multi-stage type having a first compression stage connected to an intermediate chamber serving as suction for a second compression stage. Thus, a plurality of inlet openings communicated with the first stage form the intermediate chamber whereby the compressor may be simultaneously fed at least two streams of refrigerant passed thereto from the respective evaporators.

The discharge side of the high compression stage is communicated with the inlet of the condenser, after which refrigerant is passed in a normal manner to a receiver and through an expansion means communicated with the inlet side of the evaporators, thus forming the closed system.

Referring to FIGURE 1, the system shown diagrammatically includes a reciprocating compressor **10** having a first low pressure stage **11** connected to a higher pressure stage **12** through an intermediate chamber **13**. Discharge outlet **14** communicates through line **16** to the inlet of condenser **17**. The downstream side of condenser **17** is connected through line **18** to a liquid receiver **19** holding a supply of liquid refrigerant at condenser pressure. A first evaporator unit **21** is positioned in a cooling zone to operate at a higher temperature of approximately 15° F. A second evaporator **22** is positioned in a second cooling zone and adapted to operate at a lower temperature of approximately minus 40° F.

It should be appreciated that the temperatures and

pressures hereinafter referred to are designated for the purpose of facilitating the description of the invention. However, such temperatures will be adequate to supply cooling at relatively different cooling levels.

In accordance with conventional practice, the liquid holding portion of receiver 19 is communicated through expansion valve 23 to the upstream or inlet side of evaporator 21. A second expansion valve 24 likewise communicates the liquid holding portion of receiver 19 to the inlet of evaporator 22.

A return line 26 connected to the downstream side of evaporator 21 carries vaporized refrigerant from the high pressure cooling zone to inlet 28 for directing refrigerant to intermediate chamber 13. Similarly, line 31 connected to the downstream side of evaporator 22 conducts a second returning stream of vaporized refrigerant from evaporator 22 to suction inlet 32, feeding the low pressure first stage.

The closed system may circulate a refrigerant such as R–12 or R–22, or mixtures thereof. To facilitate the following description, it is presumed that the refrigerating medium is R–22. At required cooling conditions, the low pressure evaporator 22 is controlled to operate at a temperature of approximately minus 40° F., and the high pressure evaporator 21 functions at a temperature of approximately 15° F.

The system, and the operating temperatures herein designated, exemplify requirements of a refrigeration installation adapted to provide the necessary cooling to a multi-zone installation such as a supermarket or food processing plant. In the instance of the latter, meat or other perishable commodities are normally stored at a sufficiently low temperature to avoid deterioration of the product over a period of time. Thus, a low temperature of minus 40° F. is deemed adequate for maintaining the quality and condition of the frozen food. Evaporator 21 is disposed in a section of the installation where a higher temperature is required. This temperature can be approximately 15° F., which is normal for a cutting or wrapping room where the product is only temporarily stored and sufficiently cooled to permit handling.

After initial compression in the first stage, refrigerant at an intermediate pressure and at a compressive temperature of approximately 157° F. is delivered to intermediate chamber 13. It should be noted that according to the invention, the intermediate chamber is essentially a connection or communicating passage from the discharge of the low pressure stage to the suction of the high pressure stage.

Inlet 28 to intermediate chamber 13 receives vaporous refrigerant from evaporator 21 at a temperature of approximately 15° F. Mixing of the secondary or side load of refrigerant gas passed to the intermediate chamber 13 with partially compressed refrigerant from the first stage 11, achieves two effects. First, it cools hot compressed refrigerant passing from the first stage. Secondly, it supplements the flow of refrigerant to the second stage in sufficient amount to permit the volumetric displacement of both first and second stages to be substantially equal.

At the compressor discharge 14, hot refrigerant is delivered to air or water cooled condenser 17, which is pre-set and controlled to operate at a temperature of approximately 105° F.

Description of the compressor

Referring to FIGURE 2, a reciprocating compressor of the type presently contemplated includes a plurality of compression cylinders mounted within a casing, and operably connected to a single crankshaft. While the following description will be drawn to the shown construction of a compression machine, it is appreciated that preferably the number of cylinders actually required can be any even number, each being of substantially equal volumetric capacity such that the first and second compression stages contain equal numbers of pistons.

As shown in FIGURE 3, the respective compressor cylinders are numbered 41 through 48 inclusive. Cylinders 42, 43, 46 and 47 constitute the first low pressure stage. Cylinders 41, 44, 45 and 48 comprise the higher pressure stage. The cylinders comprising the respective high and low pressure stages are physically separated and are radially mounted in banks within the compressor casing 51.

Referring to FIGURE 2, compressor 10 includes a casing or shaped housing 51 having opposed end openings. A wall 55 at one end of housing 51 includes a recess, positioning a main shaft bearing 98. End bell 15 is removably fastened to the casing opposite opening by bolts 94 positioned in flange 95, thereby forming a sealed enclosure to crankcase 52.

End bell 15 includes a center hub 96 extending into crankcase 52 forming a fluid tight seal at panel 97. Hub 96 positions a second main bearing 99 disposed in alignment with bearing 98.

Crankcase 50 is rotatably disposed in crankcase 52 and journaled in main bearings 98 and 99, having a keyed portion of the shaft extending through the end bell 15 to engage a suitable drive member. A central recess 70 in end bell 15, together with cover plate 75, defines a chamber 80 about crankshaft 50.

Crankshaft 50 includes a plurality of spaced throws, two being here shown for engaging the lower end of piston rods associated with the respective compression cylinders.

Referring to FIGURES 2 and 4, casing 51 includes means forming a partition 54 which separates crankcase 52 from intake and discharge portion of the respective high and low pressure stages.

Assembly 42 is illustrative of the remaining cylinders and includes a piston 56 pinned to rod 57, the latter being journaled to throw of crankshaft 52. A cylinder liner assembly 42 is positioned in housing 51, and partition 54, respectively, and fastened thereto by a plurality of bolts 58. Low stage 57′ extends downwardly into crankcase 52, being sealably received at an opening formed in partition 54, by a seal ring or gasket 59.

Referring to FIGURES 3 and 4, partitions 61 and 62 extend longitudinally of compressor casing, defining low pressure stage 11 and high pressure stages 12 and 12′.

End bell 15 carried on casing 51 includes a low pressure suction inlet 32 connected through means forming a passage, in communication with the low pressure stage 11, for delivering vaporized refrigerant to cylinder assemblies 42, 46, 43 and 47.

As seen in FIGURE 2, each compression cylinder is provided with concentric ring type suction and discharge valves for controlling flow of gas to and from the cylinder.

Casing 64 fastened to the upper surface of housing 51 forms a discharge manifold 64′ for cylinders 42 and 46. Passage means 67 in the upper wall of casing 52, directs refrigerant leaving the low pressure stage 11 into the intermediate chamber 68.

Referring again to FIGURE 2, the annular valve arrangement includes refrigerant forming a passage 69 communicated with suction passage 63 for introducing refrigerant to the cylinder assembly 42. The valve means includes the annular member 71 having a channel 72 formed therein and communicated with passage 69. A movable element 73 is normally urged by springs 74 to close passage 69. Annular protruding lip 76 positioned adjacent element 73, limits upward movement of the latter when piston 56 is on the intake stroke thereby drawing refrigerant through passage 69 and into the cylinder liner.

Cap 77 forms a head member to the cylinder liner assembly and includes a peripheral groove 78 in communication with the discharge opening 79 for directing compressed refrigerant gas from the first stage, into the first

stage manifold 64'. A diaphragm 81 retained in groove 78 is biased to regulate discharge of refrigerant.

On the upstroke of piston 56, diaphragm 81 is displaced against spring 82 a predetermined pressure thereby discharging hot refrigerant gas at first stage pressure.

Referring to FIGURE 3, hot refrigerant gas flows from the first low pressure stage into intermediate chamber 68, and is re-distributed therefrom to compartments 12 and 12' making up high pressure stages, both of which compartments are disposed outboard of the low pressure stages. Passageways 83 and 84 are communicated with the high pressure stages 12 and 12' holding compression cylinder 41, 45, 44 and 48.

A shown in FIGURE 4, second stage inlet chambers 86 and 86' are communicated through restrictive passage means 87' to crankcase 52, to effect the proper pressure balance across the high stage pistons by bleeding vapor from the high stage suction to the crankcase.

Referring to FIGURE 2, means forming a second suction inlet 87 is formed in housing 51 providing communication with intermediate chamber 68. Suction opening 87 is provided with a suitable connection receiving a line connected to the downstream side of evaporator 21 whereby refrigerant gas may be introduced into intermediate chamber 68 for mixing with hot compressed refrigerant discharged from the first compression stage 11.

High pressure stages 12 and 12' are defined by cylinders 41, 45, 44 and 48, each of which is fixedly mounted in casing 51 and extending through partition 54 to crankcase 52. The structural arrangement of the respective cylinders follows substantially the construction of cylinder assembly 42 previously referred to, with respect to the low pressure stage.

In like manner, valve mean associated with high pressure cylinders 41, 45, 44 and 48 is identical in construction and arrangement to the valve means regulating flow to and from the low pressure cylinder. As shown in FIGURE 4, the high pressure cylinders discharge hot compressed refrigerant into discharge manifold 88 formed by casing 89, suitably carried in place on casing 51 by bolts 91. A passage means 92 formed in casing 89 carries refrigerant at discharge temperature and pressure through discharge opening 14 for passage to condenser 17.

Unloader arrangement

Referring to FIGURE 2, low pressure cylinder 46 is shown provided with a shut off means for unloading certain cylinders, thereby varying the capacity of the compressor. Although the unloader device is shown only in respect to cylinder 46, it is understood that one or more of the remaining low compression cylinders and one or more of the high pressure cylinders, may similarly be provided with unloading means communicated with a central control for regulating the respective unloading units.

Unloader assembly 100 is shown in FIGURE 2, and includes annular ring 104 carried on the outer wall of liner 101 and sealably fastened thereto as a permanent peripheral seal or by means of resilient O-rings. A nozzle 102 is threadably carried in partition 54 and connected through conduit 103 to a source of oil or other pressurizing medium for actuating the unloader. Nozzle 102 is held in a substantially upright position and includes a central opening extending therethrough communicating with conduit 103. A ring member 104 includes an inner sealing surface having a ring 106 disposed in rubbing contact with the outer wall of liner 57 conforming a fluid type seal therewith. A cavity formed in the upper face of ring member 104 defines an annular opening disposed in sliding contact with seal ring 107 carried on the outer surface of annular member 101.

As shown in FIGURE 2, ring member 104 is provided with one or more longitudinal passages adapted to receive one or more nozzle members 102 which form a seal tight joint therewith. For this purpose, a plurality of rings may be carried on the outer surface of said nozzle member and disposed in contact with the adjacent wall. Upper face 111 of the ring member is adapted to engage the lower surface of valve ring 112 having passages 69 disposed therein. The lower face of ring 104 is provided with a plurality of bores 114 adapted to receive and hold springs 116 for normally biasing ring 104 upward into engagement with the lower surface of ring valve 112.

Referring to FIGURE 1, a compressor unloading system adapted to the instant invention incorporates a pressurizing source including a pump 115 having a suction inlet communicated with an oil reservoir 117. Solenoid valve means 120 is connected in line 119 to pump 115 discharge for regulating the flow of pressurized oil to the respective nozzles 102 at the individual unloaders.

Referring to FIGURE 1, following a normal unloading system, valve 120 is representative of similar valve arrangements connected to remaining high or low pressure cylinders.

In the unloaded position, oil is redirected from the unloader assembly at compressor cylinder 46 to the oil reservoir, thereby reducing the oil pressures permitting the latter to be spring biased to the cut off position, blocking the flow to said cylinder. Thus, when one or more cylinders in the first stages are to be unloaded, removal of pressure from conduit 103 will permit springs 116 to urge the valve ring member 113 upwardly to engagement at the lower surface of valve ring 112.

When the load on the system requires that a cooling capacity be reduced, unloading is achieved by cutting off flow of gas from suction chamber 63 to passage 69. In like manner, each of the unloading assemblies may serve to reduce the volumetric capacity of either or both the first and second compression stages.

As seen in FIGURE 1, individual cylinder unloader assemblies 100 are connected to a control center 110'. The control center is in turn communicated by sensing means to evaporators 21 and 22 to receive a signal. The control means may be of a type known to the art and so programmed to establish a compressor capacity pattern in accordance with a predetermined load pattern.

Operation

The preferred start-up procedure for the compressor follows substantially the steps recommended for similar refrigerant system compressors. For example, to facilitate starting up, the compression cylinders are at maximum unloaded condition to lessen the starting burden, and maintained thus until the compressor reaches rated speed.

For full or rated operation conditions, all cylinders are originally loaded. Referring to FIGURE 1, a hypothetical load as heretofore noted, consists of evaporator 21 being operated at a high pressure to maintain a temperature of approximately 15° F. Evaporator 22 is simultaneously operable to maintain a temperature of approximately minus 40° F. Refrigerant vapor from the downstream side of evaporator 21 is conducted through line 26 to inlet 28 of the compressor intermediate chamber. Simultaneously, gas from evaporator 22 is conducted by conduit 31 to inlet 32 of the first compressor stage. In the intermediate chamber 13, partially compressed refrigerant gas at a temperature of about 155° F. from the first compression stage mixes with refrigerant gas received from evaporator 21.

Thus, the total output of the first stage compressor cylinders is supplemented by a secondary flow of refrigerant from the high pressure portion of the system, the latter flow being hereinafter referred to as the side load on the compressor.

Since both first and second compression stages may be of substantially equal volumetric displacement the side load injected into intermediate chamber 13 effects two functions. First, it serves to cool heated first stage discharge gas. Secondly, it provides the additional volume of gas flow to the second compression stage.

Partial loading

At part load operation, return vapor flow from the evaporators to the compressor 10 may result from a reduction in the cooling load at either or both evaporators 21 and 22. When the vapor feed to the first compression stage 11 is reduced, one or more cylinders in said stage may be unloaded and the output of the remaining loaded cylinders fed into the second compression stage.

Simultaneously, the volumetric displacement of high pressure cylinders is balanced to receive the intermediate stage discharge of partially compressed refrigerant gas. Thus, one or more cylinders in the high pressure second stage are unloaded to maintain the compressor discharge conditions.

When the cooling load on high pressure evaporator 21 is reduced, the intermediate or normal side load to compressor 10 is reduced accordingly. To supplement the side load to the cooling effect first compression stage discharge, regardless of the evaporator output, means is provided for introducing liquid refrigerant directly into the intermediate chamber.

As shown in the schematic FIGURE 1, conduit 130 is connected to the liquid holding portion of condenser 17 or receiver 19. Valve means 131 interposed in line 30 is solenoid operated to adjust flow of liquid passing therethrough. Thus, refrigerant on entering intermediate stage 13 is immediately vaporized at the higher intermediate stage temperature to form a gaseous mixture for combining with the discharge from the first compression stage.

Unloader control

Referring to FIGURE 1, evaporators 21 and 22 again operate at a temperature of 15° F. and minus 40° F., respectively. Line 31 communicates the outlet of the evaporator 21 to the downstream side of the first compression stage. Similarly, line 24 communicates the outlet of the higher pressure evaporator 21 to the inlet 32 communicated with the intermediate stage of the compressor. As has been previously described, hot compressed refrigerant gas leaving the compressor at line 16 is directed to the upstream side of condenser 17 at a temperature of about 215° F.

Valve means 131 is disposed in line 130 and includes a solenoid to control passage of liquid therethrough. Temperature sensing means 110 is positioned in line 16 immediately downstream of compressor discharge 14 to monitor the temperature of hot compressed gaseous refrigerant passing to condenser 17. Thus, when the output of one or more of the evaporators decreases thereby indicating a lesser temperature at the compressor downstream side, valve 131 will automatically open to meter a controlled flow of liquid refrigerant from the high pressure side of the system to the compressor intermediate stage 13.

Conversely, in the event the discharge temperature of hot compressed refrigerant at the compressor rises above the desired 216° F., valve 131 is automatically responsive to the change, to throttle the rate of flow of liquid entering the intermediate stage from the liquid part of the system.

Sensing means 113 is normally disposed in the evaporator 21 liquid holding portion and may include an arrangement normally adapted for such purpose comprising a pressure sensitive liquid containing bulb disposed within the evaporator and being responsive to pressure changes therein. Similar sensing means 114 is disposed in evaporator 22 and performs a like function of monitoring the pressure therein.

From the foregoing it is seen that the presentation provides a novel and improved compressor structure for use in a multi-zone refrigeration system not heretofore known. It is understood, however, that the disclosed embodiment represents a preferred form of the device which may be modified and adjusted without departing from the spirit and scope of the invention as defined in the appended claims.

What is claimed is:

1. A refrigeration system comprising:
(a) a compressor operatively connected to a source of motive power,
(b) a condenser receiving gaseous refrigerant from the compressor to condense the same,
(c) a first evaporator receiving liquid refrigerant from the condenser,
(d) a second evaporator receiving liquid refrigerant from the condenser and operating at a temperature substantially higher than the operating temperature of the first evaporator,
(e) expansion means disposed between the condenser and each of the evaporators to expand the liquid refrigerant selectively responsive to the operating temperature of the evaporators,
(f) The compressor having a first stage and a second stage,
(g) the first stage of the compressor connected to draw refrigerant from the first evaporator and compress the same,
(h) the second stage of the compressor connected to draw refrigerant from both the first stage and the second evaporator in a combined flow for compressing the refrigerant prior to discharge thereof into the condenser,
(i) a first unloader means operatively connected with the first stage of the compressor,
(j) a second unloader means operatively connected with the second stage of the compressor, and
(k) a control means to sense the operating temperature conditions of the refrigeration system and to selectively operate the first unloader means and the second unloader means to maintain the volumetric displacement of the compressor responsive to the operating load conditions.

2. A refrigeration system comprising:
(a) a compressor operatively connected to a source of motive power,
(b) a condenser receiving gaseous refrigerant from the compressor to condense the same,
(c) a first evaporator receiving liquid refrigerant from the condenser,
(d) a second evaporator receiving liquid refrigerant from the condenser and operating at a temperature substantially higher than the operating temperature of the first evaporator,
(e) expansion means disposed between the condenser and each of the evaporators to expand the liquid refrigerant selectively responsive to the operating temperature of the evaporators,
(f) the compressor having a first stage and a second stage,
(g) the first stage of the compressor connected to draw refrigerant from the first evaporator and compress the same,
(h) the second stage of the compressor connected to draw refrigerant from both the first stage and the second evaporator in a combined flow for compressing the refrigerant prior to discharge thereof into the condenser, and
(i) the first stage's discharge of refrigerant at a substantially higher temperature that the temperature of the refrigerant combining therewith from the second evaporator whereby the first stage flow is cooled and added to so that the combined flow of the second stage is of substantially the same volumetric displacement as that of the first stage.

3. A refrigeration system comprising:
(a) a compressor operatively connected to a source of motive power,
(b) a condenser receiving gasous refrigerant from the compressor to condense the same,

(c) a first evaporator receiving liquid refrigerant from the condenser,

(d) a second evaporator receiving liquid refrigerant from the condenser and operating at a temperature substantially higher than the operating temperature of the first evaporator,

(e) expansion means disposed between the condenser and each of the evaporators to expand the liquid refrigerant selectively responsive to the operating temperature of the evaporators,

(f) the compressor having a first stage and a second stage,

(g) the first stage of the compressor connected to draw refrigerant from the first evaporator and compress the same,

(h) the second stage of the compressor connected to draw refrigerant from both the first stage and the second evaporator in a combined flow for compressing the refrigerant prior to discharge thereof into the condenser,

(i) an intermediate chamber means disposed between the first stage and the second stage,

(j) the first stage discharging its refrigerant into the intermediate chamber means, and

(k) the second evaporator discharging a predetermined quantity of refrigerant into the intermediate chamber means to cool and mix with the refrigerant from the first stage whereby the volumetric displacement of the second stage is substantially equal to that of the first stage.

4. The combination claimed in claim 3 wherein:

(a) the condenser discharging a predetermined quantity of liquid refrigerant into the intermediate chamber wherein it will flash into gaseous refrigerant to supplement the flow of gaseous refrigerant from the second evaporator whenever the flow therefrom is below the predetermined quantity required to build up the volumetric displacement of the second stage to a substantially equal amount to that of the first stage.

5. A refrigeration system comprising:

(a) a compressor operatively connected to a source of motive power,

(b) a condenser receiving gaseous refrigerant from the compressor to condense the same,

(c) a first evaporator receiving liquid refrigerant from the condenser,

(d) a second evaporator receiving liquid refrigerant from the condenser and operating at a temperature substantially higher than the operating temperature of the first evaporator,

(e) expansion means disposed between the condenser and each of the evaporators to expand the liquid refrigerant selectively responsive to the operating temperature of the evaporators,

(f) the compressor having a first stage, a second stage

and an intermediate chamber means connecting said first and second stages,

(g) the first stage of the compressor connected to draw refrigerant from the first evaporator,

(h) the first stage of the compressor to discharge the refrigerant into the intermediate chamber means at a temperature much higher than the temperature of the refrigerant in the second evaporator,

(i) the second evaporator to pass refrigerant into the intermediate chamber means at a pressure substantially equal to the pressure of the refrigerant from the first stage, and

(j) the second stage of the compressor to draw refrigerant from the intermediate chamber means including the combined flow of the first stage and the second evaporator for compressing the refrigerant prior to its discharge therefrom into the condenser.

6. A multi-stage compressor for use in a refrigeration system having at least two evaporators, one of which operates at a substantially higher temperature and pressure than the other, the compressor comprising:

(a) a casing having a plurality of chamber means formed therein,

(b) a first compression stage disposed in one of the chamber means and communicating with the evaporator of lower temperature and pressure,

(c) a second compression stage disposed in another of the chamber means remote from the first compression stage,

(d) an intermediate chamber means communicating the first compression stage and the second compression stage,

(e) the intermediate chamber means receiving gaseous refrigerant from the evaporator of higher temperature and pressure to increase the quantity of refrigerant flow thereof whereby the volumetric displacement of the second compression stage is substantially equal to that of the first compression stage.

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