Jan. 31, 1967 N. HOLBAY 3,301,000
COMBINATION VAPOR COMPRESSION AND ABSORPTION
REFRIGERATION SYSTEM
Filed Feb. 15, 1965 2 Sheets-Sheet 1

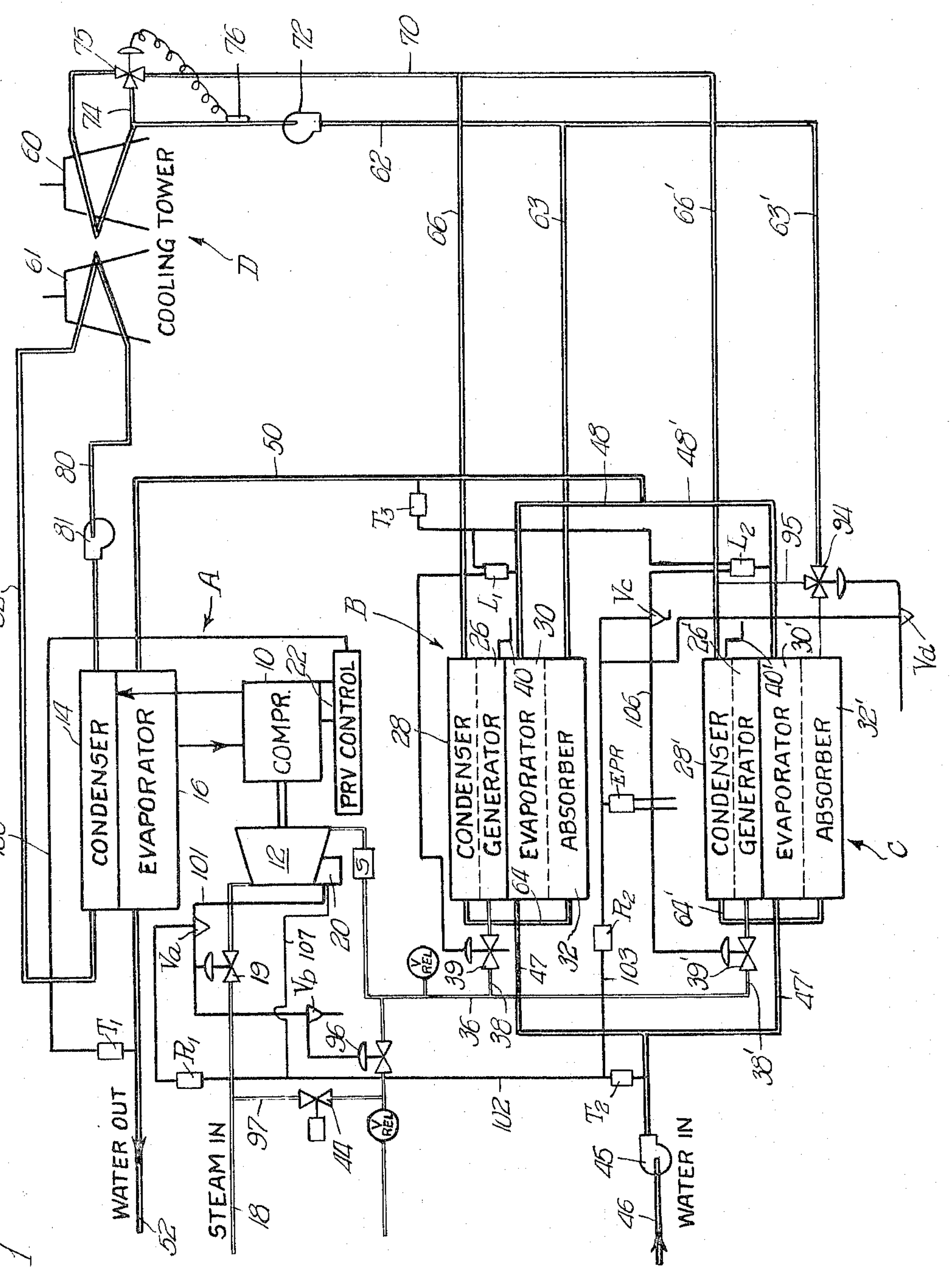

Fig. 1

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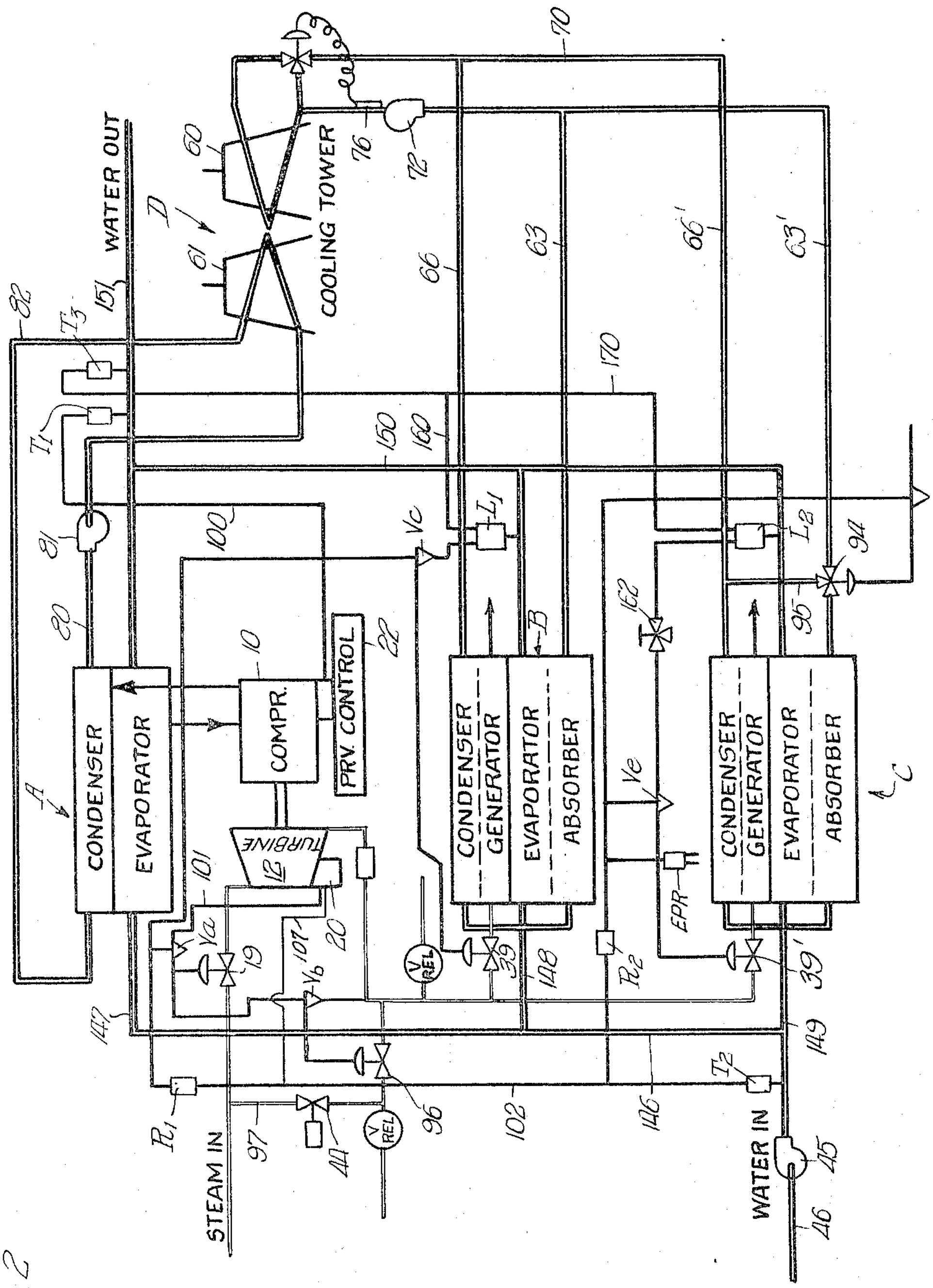

Fig. 2

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3,301,000
Patented Jan. 31, 1967

3,301,000

COMBINATION VAPOR COMPRESSION AND ABSORPTION REFRIGERATION SYSTEM

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Filed Feb. 15, 1965, Ser. No. 432,658
11 Claims. (Cl. 62—141)

This invention relates generally to refrigeration systems and more particularly to improvements in control systems for combined vapor compression-absorption refrigeration units.

In refrigeration systems of the type wherein the sole refrigeration producer comprises a conventional compressor-condenser-evaporator circuit including a steam turbine as the prime mover driving the compressor, only a fraction of the energy present in the steam supplied to the turbine is converted into cooling capacity. Unless the exhaust steam is required for some specific application, such as for example a process or other load requiring heating, it is delivered to a steam condenser where its potential for producing additional refrigeration capacity is wasted.

The present invention is directed to combination systems using a turbine driven compressor and one or more absorption refrigeration units which are arranged to make use of this otherwise wasted heat energy by passing the exhaust steam from the turbine into the absorption units. The absorption unit thus serves as a condenser for the exhaust steam from the turbine, and substantially all of the thermal energy of the exhaust steam is efficiently utilized.

While other types of compressors may be used in combination with a steam turbine, the description herein will be directed to a compressor of the centrifugal type, since centrifugal compressors are the ones most commonly used in refrigeration systems having a total refrigeration capacity within the range for which the present invention is best suited. Therefore, the term "centrifugal-absorption system" will be used hereinafter to refer to the combined refrigeration producing system.

Combined centrifugal-absorption systems have a number of advantages over centrifugal systems or absorption systems per se. Inasmuch as most refrigeration systems, and especially those used in air conditioning units, operate on partial load most of the time, the operating economy of a combination centrifugal-absorption system, due to the fact that it makes use of the energy of the exhaust steam, is superior to that of a condensing type, turbine driven, centrifugal system. Combined systems also offer an advantage over the use of absorption systems alone because their steam consumption rate over substantially the entire range of operation is less than the steam consumption rate of an absorption unit or units having the same capacity.

Notwithstanding these aforementioned advantages, the control of refrigeration capacity over a wide range of varying loads has presented many problems. For example, in order to insure economical operation, there must be a correct balance between the portion of the load carried by the absorption unit and portion of the load carried by the centrifugal unit. In addition, there are a number of practical limitations on the manner of controlling capacity. In this regard, the temperature of the chilled water leaving the system for circulation to the cooling load should be kept constant regardless of variations in said load. In most air conditioning systems, the latent to sensible load ratio increases substantially at partial load. This requires an increased dehumidifying capacity from the cooling coils, and since the dehumidifying capacity of the cooling coil is related to the difference between the

temperature of the entering chilled water and the dewpoint temperature of the entering air, at higher chilled water temperatures, the dehumidifying effect of the cooling will decrease, resulting in higher relative humidities in the air conditioned space. Consequently, although it may result in some savings in operation, it is not recommended that the chilled water temperature be increased as the load decreases.

Maximum operating economy, particularly of the absorption units, is reached at relatively high capacity, and the operating costs per ton of capacity begin to increase as the load factor is reduced. In systems using multiple units, it is therefore desirable that an absorption unit be shut down as the load falls below a predetermined value in order to enable the units which remain in operation to function at a higher load factor. It is also expedient to provide means for discontinuing the supply of cooling water to the absorption units which are inoperative. The latter may take the form of a bypass means around the absorption units in combination with a control valve responsive to the condition of the unit.

It is therefore a principal object of the invention to provide an improved refrigeration capacity control system which is adapted to achieve more economical operation at partial loads.

It is another object of the invention to provide a refrigeration system with dual capacity control means for better economy at partial load operation.

It is another object of the invention to provide an improved control system for a combination centrifugal-absorption system which includes means for automatically discontinuing the operation of certain components in a predetermined sequence as the load decreases.

It is another object of the invention to provide a refrigeration control system, in accordance with the foregoing objects, which is adapted to maintain a substantially constant exiting chilled water temperature.

It is still another object of the invention to provide a refrigeration control system, in accordance with the foregoing objects, which includes means for preventing cooling water from passing through any inoperative absorption units.

Additional objects and advantages will be apparent from reading the following detailed description taken in conjunction with the drawings wherein:

FIGURE 1 is a schematic representation of a combined centrifugal-absorption refrigeration system constructed in accordance with the principles of the present invention;

FIGURE 2 is a schematic representation of a second embodiment of the invention.

There are two principal arrangements for combined systems which may be classified according to the type of chilled water circuit associated therewith. If chilled water is flowing in parallel paths through the centrifugal unit and the absorption unit(s), it is commonly referred to as a "parallel system", which term will be used to designate this type of system in the present specification. If the chilled water flows first through the absorption unit(s) and then through the centrifugal unit, the arrangement is referred to as a "series system".

In a parallel system, the temperature drop of the chilled water as it passes through each unit is approximately the same. In a series system, the temperature drop is cumulative and consists of the sum of the temperature drops in the respective units. Consequently, series systems are more adaptable for systems which require larger overall chilled water temperature differences. Series systems are, in many respects, easier to control. Since all of the chilled water flows through the centrifugal unit, the temperature of the leaving chilled water can be controlled by modulating the capacity of the compressor.

The first type of system to be described is the series system which is shown schematically in FIGURE 1. While the two embodiments differ with respect to their overall configuration and operation, many of the individual components may be identical. Therefore, the same reference characters and numerals will be applied to corresponding components in both modifications, if the use of such common characters and numerals does not create any ambiguity.

The series system shown in FIGURE 1 comprises a turbine driven centrifugal unit A, a pair of absorption units B and C, and a cooling tower D, or other source of cooling water. It will be seen that insofar as the chilled liquid circuits is concerned, the details of which will be described below, the absorption units are connected in parallel with respect to each other; and both absorption units are connected in series with respect to the centrifugal unit A.

THE CENTRIFUGAL UNIT

Centrifugal unit A comprises a compressor **10**, driven by a steam turbine **12**, a condenser **14**, and an evaporator **16**. The compressor, condenser, and evaporator are connected to provide a closed refrigeration circuit in a manner understood by those skilled in the art. The compressor **10** may be of any conventional type, but for purposes of illustration, it is described as a centrifugal compressor. As pointed out above, this is because centrifugal units are most commonly used in systems having a refrigeration capacity in the range for which the invention is best suited. Steam is supplied to turbine **12** through steam inlet line **18**, said line including a control valve **19**.

The turbine is provided with a variable speed regulator means or governor which broadly includes steam valve **19** and speed responsive control means **20**, said means being adapted to be set for any desired speed by the capacity control system described in more detail below.

Turbine speed responsive control means **20** may take the form of a centrifugally actuated speed governor on some other conventional arrangement which is effective to operate the steam valve **19**, through a pneumatic control, in response to changes in compressor rotor (and turbine output shaft) speed caused by variations in the cooling load and corresponding torque load on the turbine.

THE ABSORPTION UNITS

The absorption units B and C are essentially identical, so in describing unit B, it will be understood that unit C includes corresponding components which will be designated by the same reference numerals, but distinguished by the use of a prime. For purposes of illustration, it will be assumed that the absorption units are of the type using a hygroscopic brine, such as Li Br as the absorbent, and water as the refrigerant. Unit B comprises a generator **26**, a condenser **28**, an evaporator **30**, and an absorber **32** all connected to provide a closed circuit, continuous cycle refrigeration system. Since the general operation of absorption systems of this type are so well known, no detailed description is given herein; although reference may be made to U.S. Patent 2,986,906 for a more complete description of a typical absorption unit.

The steam exhausted from the turbine is supplied to a heat exchanger (not shown) forming a part of generators **26**, **26'** for heating the dilute absorbent solution. Under the pressure (approximately 75 mm. Hg absolute) existing within the generator and condenser, the refrigerant (water) boils off and flows to a heat exchanger (not shown) in the condenser where the refrigerant vapor is condensed, collected, and then forwarded to the evaporator. There, the refrigerant is passed over a heat exchanger (not shown), referred to as the chilled water coil, where it boils at the reduced pressure existing in the evaporator-absorber shell, extracting heat from the chilled water flowing therethrough. The refrigerant vapor evaporated in the evaporator section is absorbed by the absorbent solution in the absorption section, and heat of absorption is removed by cooling water flowing through a heat exchanger (not shown) in the absorber section. The details of the steam supply, chilled water, and cooling circuits are described herein below.

STEAM SUPPLY

High pressure steam is supplied through line **18** to the turbine. The exhausted steam, having a pressure which has been reduced to approximately 12 p.s.i.g., is conducted through line **36**, containing a steam separator S, and a pair of parallel branch supply lines **38**, **38'** containing valves **39**, **39'** to the generators of absorption units B and C respectively. After passing through the generator heat exchangers, the condensate is discharged through lines **40**, **40'**.

THE CHILLED WATER CIRCUIT (SERIES)

In a series circuit, the chilled water flows first through the absorption units and then through the centrifugal unit, with all of the absorption units being connected in parallel with one another. The inlet line for the chilled water from the cooling load is designated at **46**, said water being circulated by pump means **45** to a pair of parallel branch conduits **47**, **47'** which pass through the chilled water coils in the evaporators **30**, **30'** respectively. The water leaving the absorption units is combined at the juncture of conduits **48**, **48'** and forwarded to the heat exchanger in the chiller or evaporator **16** through line **50**. The line for conducting the chilled water back to the cooling load is indicated at **52**.

While the operating temperatures are not especially critical, in the typical air conditioning system, the chilled water would normally enter the combined centrifugal absorption unit at a temperature of approximately 57° F. The temperature drop of the chilled water as it passes through the absorption units is approximatley 10° F. so that the chilled water forwarded to the centrifugal unit evaporator is at a temperature of approximately 47° F. The centrifugal unit provides an additional temperature drop of about 5° F. Therefore, the water forwarded to the cooling load is at approximately 42° F.

COOLING WATER CIRCUIT

Cooling tower D, which in a preferred embodiment, comprises two separate units to supply cooling water independently to the centrifugal A unit and the absorption units B and C, provides a convenient source of cooling water, although it should be understood that some other water source may also be used. Each of the absorption units requires cooling water not only for the condenser, but also for the absorber to remove the heat generated by the absorption of water vapor into the absorbent solution Referring first to the section of cooling tower D utilized to provide water for the absorption units, said section being designated **60**, the water flows through the supply header **62** to branch conduits **63**, **63'** to the heat exchangers in absorbers **32**, **32'**, and is then passed through lines **64**, **64'** to the condenser heat exchanger. The water is returned to the cooling tower through lines **66**, **66'** and header **70**. A pump **72** circulates water throughout the cooling system.

The cooling water supplied to the absorption units is preferably maintained at a constant temperature (approximately 85° F.) by means of a bypass line **74** around the cooling tower which cooperates with a three-way valve **75** operated by a temperature responsive bulb **76** sensing the temperature of the water supplied to the absorption unit. The cooling tower section **61** used to supply water to the centrifugal unit condenser **14** is connected to a circuit including a supply line **80**, pump **81**, and return line **82**.

THE CONTROL SYSTEM

While it will be appreciated that various types of control systems may be employed, in a preferred embodiment, the control system associated with the combined centrifugal absorption units is pneumatically actuated. Capacity control is achieved in two ways. First of all, that which may be regarded as the "primary" capacity control is effected by means for altering the flow of suction gas from the evaporator to the compressor. While a valve for simply throttling this flow could be used, in a preferred embodiment, this means takes the form of a pre-rotation vane control mechanism (PRV) which is adapted to vary the relative angle of the refrigeration gas as it is directed into the compressor rotor. This includes a plurality of circumferentially spaced vanes which direct the gas into the turbine rotor inlet either in a direction generally opposed to the rotation of the rotor or in a direction generally coincident therewith. PRV control systems are a common method of capacity control so that a detailed description would appear to be unnecessary.

A "secondary" capacity control is responsive to the chilled water being returned from the load and is adapted to re-set the speed regulating mechanism of the turbine such that it will operate at and drive the compressor at a lower speed during reduced load conditions. This arrangement permits greater operating economy in systems which operate at partial load most of the time. The main disadvantage of using speed control alone is that surging can exist at a higher percentage of load than with units equipped with PRV control. This surging problem can be minimized when speed control and PRV control are combined.

The primary capacity control system comprises a temperature responsive element T_1 sensing the temperature of the chilled water being forwarded to the load through line **52**. Changes in this temperature produce corresponding changes in the position of the pre-rotation vanes on the compressor inlet as indicated by variations in the pressure of control air supplied to the PRV mechanism through pneumatic control line **100**. At maximum load, the PRV unit **22** sets the vanes so that the centrifugal compressor is operating at maximum capacity. Because of the maximum load on the compressor, the turbine speed regulator **20** will supply control air through line **101** to cause steam supply valve **19** to be wide open. Three-way air valve Va in line **101**, which is controlled by the two-position pneumatic relay R_1, is also open to the turbine speed regulator S at this time. If the load decreases, the temperature of the leaving chilled water will drop causing the PRV control to move the vanes to a position which produces a reduction in compressor capacity. This will reduce the torque load on the turbine and, to maintain constant speed, the output from the turbine speed responsive control **20** will begin to throttle the main steam valve **19** which results in a decrease of the steam flow through the system and a reduction of the pressure of the steam supplied to the absorption unit generators **26, 26'**. It will thus be seen that the final chilled water temperature leaving the system will be controlled by thermostat T_1 and the PRV control unit **22**. The reverse of the above sequence of operations will occur if the chilled water temperature rises due to increased cooling load.

Steam valve **19** not only controls the speed of the turbine, but also the capacity of the absorption units, although indirectly. If the steam pressure at the absorption units rises, the cooling capacity of these units will increase, resulting in a lower leaving chilled water temperature. This lower water temperature will require a reduction in the capacity of the centrifugal unit and also in the amount of steam flow. The net result will be a decrease in the steam pressure at the inlet of the absorption units, and the corresponding reduction of the absorption units capacity will restore a balanced condition. The absorption unit steam valves **39, 39'** are fully open during normal operation and are actuated only under certain conditions which will be explained below. If only one absorption machine is used in the combination, no control valve is needed in the steam line **36** between the turbine and the absorption machine.

The secondary capacity control is responsive to the chilled water being returned from the load through line **46**. Temperature responsive element T_2 sensing this temperature is adapted to vary the pressure of control air through line **102**. On decreasing load, the control air pressure in line **102** is lowered by T_2, and line **102**, being in direct contact with the turbine speed regulator **20** through line **107**, will be effective to reset the control point of speed regulator **20** so that the turbine will operate at a lower speed at partial load, thereby resulting in better operating characteristics.

Another important feature in the control system is the automatic shut-down sequence which permits more economical operation at reduced loads. The advantages of shutting down one or more of the units when the load reaches a particular pre-determined value have been mentioned above. Means are provided in the present system to automatically effect this shut-down sequence in response to the temperature of the chilled water being returned from the load.

Temperature sensor T_2, located in chilled water return line **46**, will gradually reduce its branch pressure as supplied to pneumatic relay R_2 through line **103**. When this pressure reaches the trip point of relay R_2, it will simultaneously actuate three-way air valve Vc and electropneumatic relay EPR. This will exhaust air from the pneumatic control line **106** connected to steam valve **39'** causing it to close, and will also disconnect, by means of electropneumatic relay EPR, the electrical control circuit of absorption machine C causing it to become inoperative. At the same time, three-way air valve V*d* will exhaust air from the control line of three-way cooling water valve **94** to divert all of the water through bypass line **95** around the absorber and condenser of absorption machine C and return it to the cooling tower via line **66'**.

In combination centrifugal-absorption systems, it is advantageous to keep the centrifugal unit in operation as long as possible. Were the centrifugal unit not in operation, high pressure steam would have to be throttled by the pressure regulating valve before entering the absorption units. The steam valve and the turbine of the centrifugal unit will throttle the high pressure steam as long as it is in operation, and besides the throttling effect, it will always produce some amount of cooling capacity.

As the load further decreases, the pressure of control air supplied to relay R_1 through line **102** will drop to the trip point of the relay R_1 causing it to exhaust air through air valve V*a* from line **101**. This will completely close steam valve **19** and discontinue operation of centrifugal unit A. At the same time, steam valve **96** will open through the operation of three-way valve V*b*, and steam will now be supplied to the absorption units through bypass line **97**, pressure reduction valve **44**, and steam valve **96**.

By setting the control point of temperature controller T_3 which is responsive to the temperature of the chilled water flowing to the centrifugal unit, at a temperature of approximately 42° F., this controller will take over the operation of steam valve **39** in the event the leaving chilled water temperature drops further.

Each of the absorption units is provided with low limit controllers L_1, L_2 which are adapted to shut off the steam supplied to the generators in the event the temperatures of the chilled water leaving the evaporator drops below a predetermined value. Under normal conditions, these are set for approximately 40° F.

THE PARALLEL CHILLED WATER SYSTEM

Referring now to FIGURE 2, there is illustrated in schematic form, a combination centrifugal-absorption refrigeration system in which the chilled water circuit is connected in parallel. The pneumatic control system is substantially the same as that associated with the series system shown in FIGURE 1. However, some modification of this control system is necessary and will now be described in detail. The respective components of the parallel system are designated by the same reference numerals as the series system shown in FIGURE 1, with the exception of certain of the chilled water lines which are indicated by different numerals.

The chilled water returning from the load through line **46** is circulated by pump **45** to a supply conduit or header **146**, which in turn, is connected to a plurality of parallel branch conduits **147**, **148**, and **149** respectively leading to the evaporator in centrifugal unit A and the evaporators in absorption unit B and C. The liquid to be chilled flows in parallel through the respective chilled water coils in these evaporators and then into a common return conduit **150** connected to a line **151** leading back to the load.

Capacity control is, like the series system, achieved by primary and secondary control circuits. The primary capacity control comprises a temperature sensing element T_1 sensing the leaving water temperature through line **151**, said temperature responsive means being connected through control line **100** to the PRV control mechanism **22** associated with compressor **10**. The secondary capacity control is operated by temperature responsive element T_2 sensing the incoming chilled water temperature, said temperature responsive element operating through line **107** to vary the effective set point of the speed control **20**.

As the system load decreases further, the capacity of the centrifugal unit will decrease at a much faster rate than that of the absorption units. The result is that, at partial load, the temperature of the chilled water leaving the centrifugal unit will be higher than the temperature that must be maintained in the common chilled water outlet line **151**. To compensate for this, and to maintain a constant system leaving chilled water temperature, the water passing through the absorption units will have to be cooled to a temperature lower than the temperature of the chilled water leaving the system. This will require the absorption units to operate at a somewhat lower evaporator temperature.

If the load reaches a predetermined low point, returning chilled water thermostat T_2 will first shut down absorption unit C through relay R_2, electropneumatic relay EPR, and three-way valve V*e*. On further decrease of the load, the turbine is shut down through the operation of pneumatic relay R_1 which actuates valve V*a*, closing steam valve **19** and simultaneously opening valve **96** through the operation of three-way valve V*b*. At the same time, the three-way air valve V*c* will enable the thermostat T_3 to assume control of the leaving chilled water temperature by the actuation of steam valve **39** associated with absorption unit B. Thermostat T_3 is operable to control steam valve **39** by varying the supply of control air through line **160** to limit controller L_1 and three-way valve V*c*. During this period of operation, steam is supplied through bypass line **97** and valve **96** which opens up when the centrifugal unit is shut down. During periods when the cooling load is increasing, the aforementioned sequence is reversed.

Thermostat T_3 can also control the two absorption units when these units are operating alone without the centrifugal unit. Manually operated valve **162** will then connect valve **39'** to the control line of thermostat T_3 through low limit controller L_2 via line **170**.

The three-way pneumatic valves V*a*, V*b*, etc., two position pneumatic relays R_1, R_2, electropneumatic relay EPR, and proportional temperature controllers T_1, T_2, etc., are all standard items available from most manufacturers of control apparatus. However, in order to complete the disclosure reference is made herein to specific units manufactured by Johnson Service Co., Milwaukee, Wisconsin.

Unit:	Johnson Service Model No.
Proportional controller, T_1, T_2, T_3	T–800
Two-position relays, R_1, R_2	R–350
Electro-pneumatic relay EPR	G–152
Three-way valve, V*a*, V*b*, etc.	V–128

While this invention has been described in connection with certain specific embodiments thereof, it is to be understood that this is by way of illustration and not by way of limitation; and the scope of this invention is defined solely by the appended claims which should be construed as broadly as the prior art will permit.

What is claimed is:

1. A refrigeration system comprising a compressor, a condenser, and an evaporator connected to provide a refrigeration circuit; a first heat exchanger in heat transfer relation with said evaporator; an absorption refrigeration machine having a generator, a condenser, an evaporator, and an absorber connected to provide a closed, continuous cycle refrigeration circuit; a second heat exchanger in heat transfer relation with refrigerant in said absorption machine evaporator; means for circulating a heat exchange medium to and from a cooling load through said first and second heat exchangers; a steam turbine driving said compressor; means for conducting steam in series flow to said turbine and in heat exchange relation with dilute absorbent solution in said generator; first capacity control means including means for altering the flow of refrigerant gas from said evaporator to said compressor; and second capacity control means including valve means responsive to the temperature of the heat exchange medium returning from said load for varying the flow of steam to said turbine, whereby the rotational speed of said turbine is regulated in response to changes in the cooling load.

2. Apparatus as defined in claim **1** wherein said first and second heat exchangers are connected in series flow relation.

3. Apparatus as defined in claim **1** wherein said heat exchange medium upon returning from said load is passed in series from said second heat exchanger to said first heat exchanger.

4. Apparatus as defined in claim **1** wherein said first and second heat exchangers are connected in parallel flow relation.

5. Apparatus as defined in claim **1** wherein said first and second heat exchangers are connected in series flow relation.

6. Apparatus as defined in claim **5** wherein said heat exchange medium upon returning from said load is passed in series from said second heat exchanger to said first heat exchanger.

7. Apparatus as defined in claim **6** wherein said first and second heat exchangers are connected in parallel flow relation.

8. Apparatus as defined in claim **1** wherein said refrigeration system includes at least two absorption refrigeration machines of the type defined in claim **1**; means connecting each of the heat exchangers associated with the absorption machine evaporators in parallel flow relation; and means responsive to the temperature of heat exchange medium returning from said load for sequentially discontinuing operation of all but one of said absorption machines prior to discontinuing operation of said centrifugal compressor.

9. Apparatus as defined in claim **1** including steam supply by-pass means around said turbine to said absorption machine generator, said means including a reducing valve; and means operated in response to heat exchange medium returning from said load for closing said turbine steam supply valve and opening said by-pass means.

10. Apparatus as defined in claim **9**, including means responsive to heat exchange medium leaving said absorption unit evaporator heat exchanger for controlling the steam supplied through said by-pass means.

11. Apparatus as defined in claim **10** including means for supplying cooling water to each of said absorption unit absorbers and condensers; and means actuated in response to the discontinued operation of any absorption machine to bypass said cooling water around said absorption machine.

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