March 16, 1971 H. J. HENRY ET AL 3,570,265

COMPRESSOR COOLING SYSTEM

Filed April 21, 1969 2 Sheets-Sheet 1

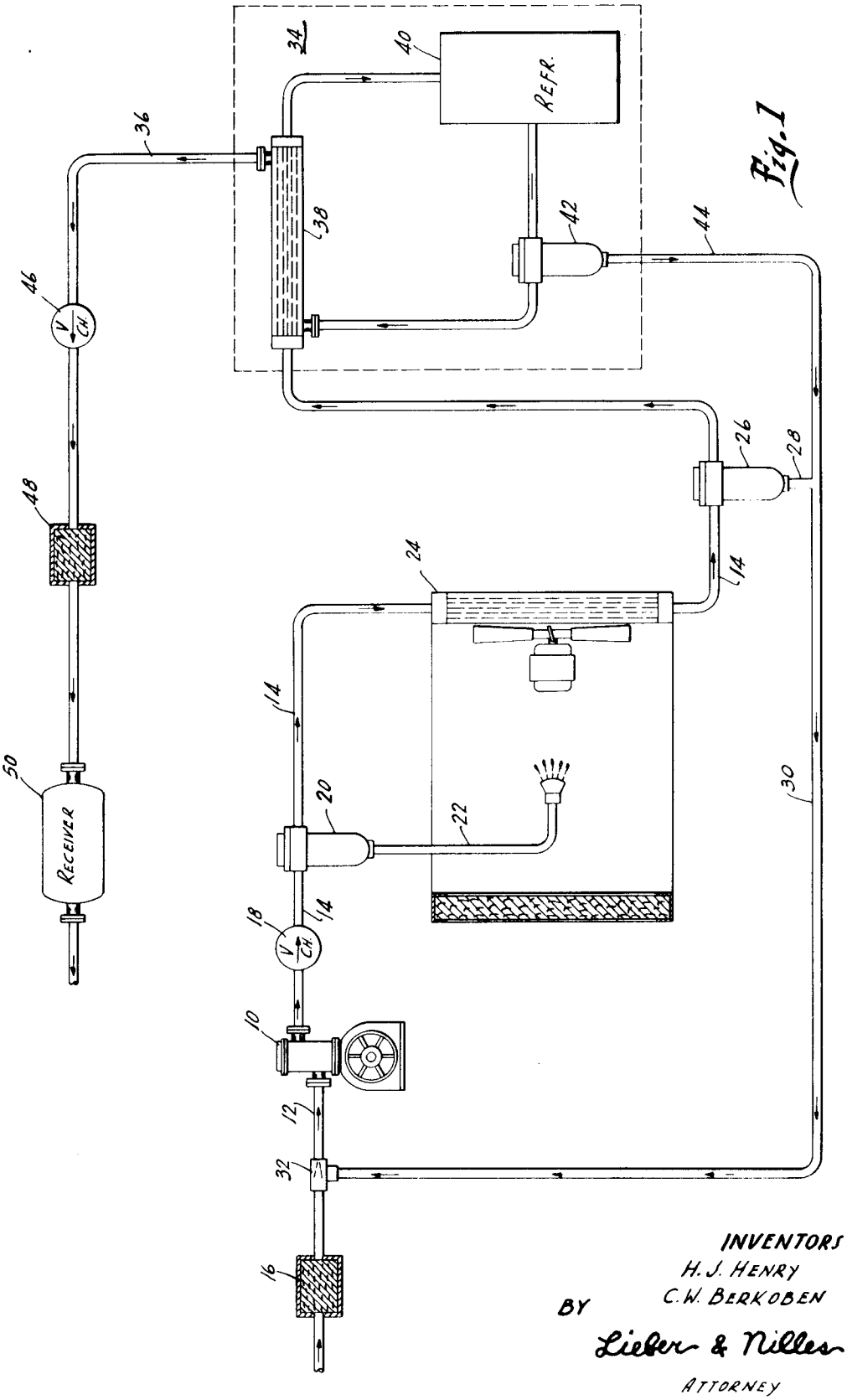


Fig. 1

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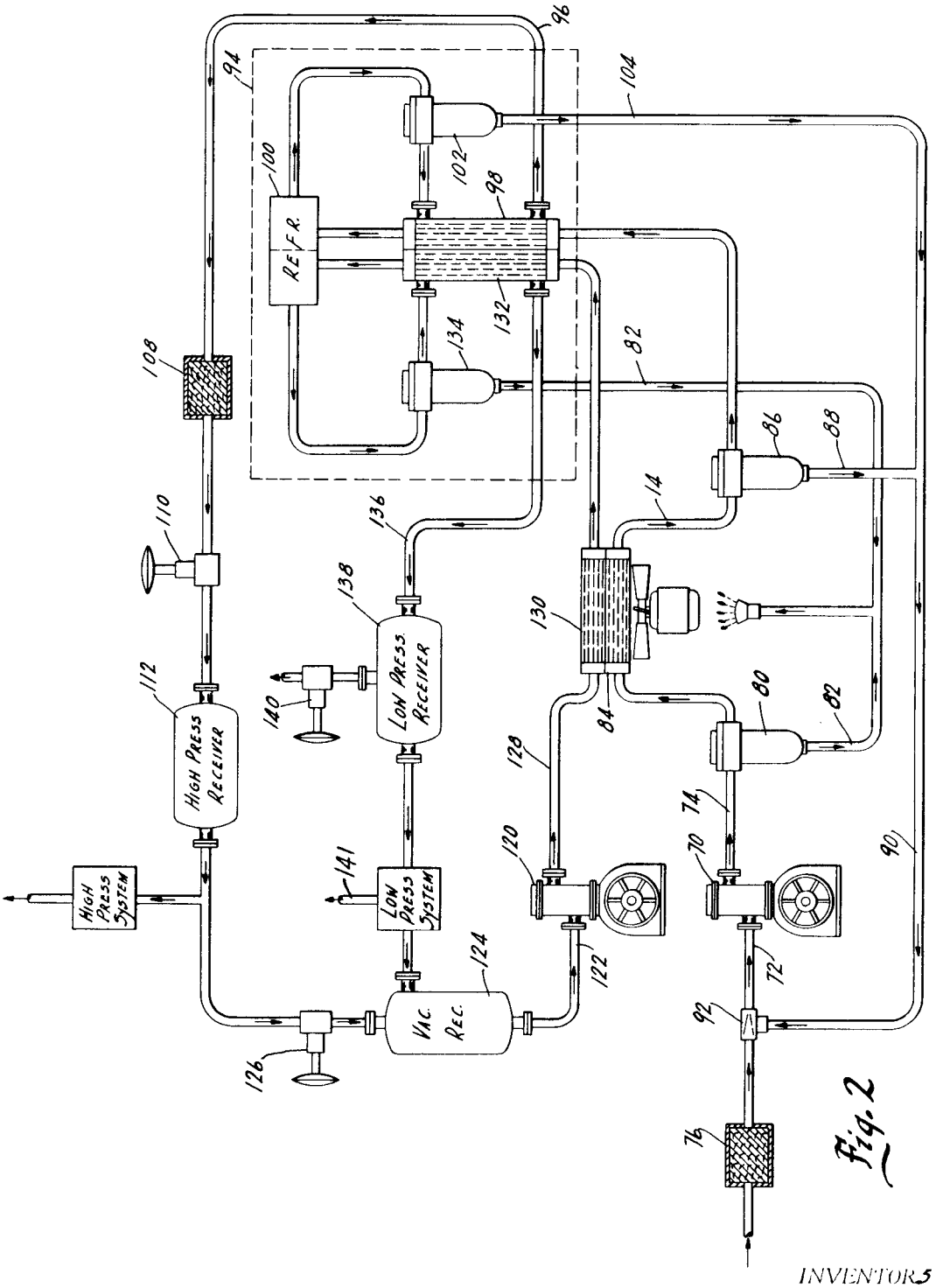


Fig. 2

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COMPRESSOR COOLING SYSTEM

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ABSTRACT OF THE DISCLOSURE

A self-stabilizing cooling system for air compressors embodying evaporative cooling, and in which means are provided for automatically obtaining condensed water vapor from the atmosphere in required amounts for cooling purposes, means also being provided for recirculating the coolant in the system and for injecting the same into the low pressure inlet.

BACKGROUND

One of the difficulties present in the compression of air and gas is that considerable heat is generated during the compressing operations. As a result, the compressor parts are subjected to heat which frequently causes, or at least contributes to, rapid deterioration, breakage and compressor malfunction.

In effects to minimize the damage attendant heat generation in compressors, numerous arrangements and/or variants in the lubrication of compressors have heretofore been proposed. Also, efforts have been made to reduce the heat generated by introducing a cooling medium of one type or another directly into the system and especially at or near those parts of the compressor which are most susceptible to heat.

However, these prior proposals for compressor lubrication and/or injection of coolant into the system have, for the most part, presented a number of objections. For example, the use of excessive lubricant often presents the need for bulky and expensive separators for removing oil, reservoirs for storing the same, pumps for circulating the lubricant, etc. As for the injection of coolant directly into the system, these same objections are frequently present, plus the fact that adequate supplies of the coolant must be provided and prior systems have required rather complicated controls and have certainly not been self-stabilizing.

SUMMARY

It is therefore an object of the present invention to provide an improved self-stabilizing cooling system for air and/or gas compressors which obviates the aforesaid disadvantages and objections attendant prior systems.

Another object of this invention is to provide an improved system for cooling air or gas compressors and for maintaining the same at an efficient operating temperature which utilizes the water vapor present in the atmosphere for cooling purposes.

Still another object of the invention is to provide an improved system of cooling air or gas compressors to maintain the same at an efficient operating temperature which provides for the recirculation of condensed vapor entrained in the system and a relatively vapor-free final compressed air product.

A further object of the present invention is to provide a cooling system for compressors, which comprises, means interposed in the high pressure discharge line for cooling the high pressure fluid passing therethrough to cause condensation of moisture entrained therein, and means for separating the condensate from the high pressure air and returning the same to the low pressure inlet.

THE DRAWINGS

A clear conception of the several features constituting the present invention and of the construction and mode of operation of a typical air compressor system embodying the improvements, may be had by referring to the drawings accompanying and forming a part of this specification.

FIG. 1 of the drawings is a more-or-less schematic view diagrammatically illustrating one simple typical air compressing system embodying the evaporative cooling concept of the present invention; and

FIG. 2 is another schematic view diagrammatically illustrating a somewhat more sophiscated system also embodying the invention.

DETAILED DESCRIPTION

Referring to the drawings, it is to be understood that the typical systems shown therein exemplify specific types of air compressing systems and certain specific operating conditions. It should be understood that the present improved evaporative cooling system may be used to like advantage with any type of compressor such as a reciprocating compressor or a screw compressor operating on air or gas containing a condensable vapor, and the word "fluid" as used herein is intended to designate either air or gas or a combination thereof. Also, the cooling system may be advantageously used in a multi-stage compressor system embodying a plurality of the same or different types of low and high pressure compressors. In addition, the intake and discharge conditions may vary, and the system may remain self-stabilizing within finite limits or inlet conditions.

Referring particularly to FIG. 1 of the drawings, the air compressor **10** has a low pressure inlet **12** and a high pressure discharge line **14**. In the specific example, 160 c.f.m. inlet air at 94° F. and 85% relative humidity is admitted to the low pressure inlet line **12** past filter **16**. Under the conditions specified, the inlet 160 c.f.m. air will contain 10.9 lbs. per minute dry air and .326 lb. per minute water vapor. With the compressor operating at a discharge pressure of 57 p.s.i.g., water vapor in the inlet air is condensed and recirculated as coolant and the system will stabilize at a compressor discharge temperature of approximately 180° F. Under stabilized condtions, the compressor discharge air will saturate with .910 lb. per minute vapor and will carry .326 lb. per minute liquid.

As shown, a one-way check valve **18** is interposed in the high pressure discharge line **14**, and a moisture separator **20** is interposed in the conduit **14** beyond the check valve. The moisture separator **20** is assumed to be approximately 90% efficient and will therefore remove and eject 90% of the liquid in the compressor discharge line, for example, .293 lb. per minute of liquid, which is then evaporated into room atmosphere or wasted to a drain. In the example shown, the liquid removed by the moisture separator **20** is conducted through a conduit **22** and is discharged into the cooling air stream over the aftercooler **24**. The compressed air in the high pressure discharge line **14** beyond the moisture separator **20** is cooled through the aftercooler or condenser **24** to 115° F. with a 1 p.s.i.g. pressure drop. At the aftercooler inlet, there is .910 lb. per minute vapor and .033 lb. per minute liquid in the 10.9 lbs. per minute air at 185° F. At the outlet of the aftercooler **24** with a temperature of 115° F., the air will saturate at .145 lb. per minute vapor, and the additional condensation results in a total of .798 lb. per minute liquid. The dicharge from the aftercooler **24** is then passed through a second moisture separator **26** interposed in the high pressure discharge line **14**, and this moisture separator **26** is again 90% efficient and separates .718 lb. per minute of liquid. This liquid as sep-

arated by the separtor **26** is then returned through a conduit **28**, **30** back to the compressor inlet as at nozzle **32**, means being provided in the moisture separators to insure no air loss will take place into the conduit with the condensed moisture, and that pressure control means provide and intermediate pressure in conduit **44** and **30** lower than the pressure existing in moisture separators **42** and **26** and higher pressure than the pressure existing in conduit **12** at the compressor intake.

Following the moisture separator **26**, the air in high pressure line **14** is passed through a refrigerated dryer generally designated by numeral **34**. The refrigerated dryer **34**, as shown, consists of an outlet **36**, an air-to-air heat exchanger **38**, a refrigerator unit **40** which includes an evaporator and a condenser, and finally a moisture separator **42**. The air-to-air heat exchanger, refrigerator unit and moisture separator are connected in series as shown, and conditions at the inlet to the dryer are 115° F. with the air containing .145 lb. per minute vapor and .080 lb. per minute liquid. With the evaporator of refrigerator unit **40** at a temperature of 35° F., the condition of the air within the dryer after it has passed over the evaporator and cooled to 35° F., will be such as to contain .011 lb. per minute vapor, .214 lb. per minute liquid and 10.9 lbs. per minute dry air. Again, the moisture separator is 90% efficient, and .192 lb. per minute of the liquid is separated and connected into conduit **44** for return via line **30** to the compressor inlet. Following the moisture separator, the air contains .011 lb. per minute vapor and .022 lb. per minute liquid. As this air is passed through the dryer heat exchanger **38**, it is reheated to exit from the unit via outlet **36** at approximately 70° F., the liquid now having been revaporized to result in a discharge air content of .033 lb. per minute vapor. It is obvious to one skilled in the art that temperature control of the air exiting from the dryer into conduit **36** is readily obtainable through, for example, a control means such as to bypass a portion of the cold air entering heat exchanger **38** and/or bypass a portion of the hot air entering heat exchanger **38**.

From the outlet line **36**, the high pressure air passes check valve **46** and is conducted through filter **48** to a receiver **50** for subsequent use as required. At the receiver, the 10.9 lbs. per minute dry air at 50 p.s.i.g. has a temperature of 70° F. and contains .033 lb. per minute vapor. This corresponds to an approximate 66° F. dewpoint at 50 p.s.i.g. or approximately 27° F. dewpoint at atmospheric pressure.

With reference to the liquid ejected at the second and third moisture separators, it will be noted that the total .910 lb. per minute is injected into the compressor inlet. In turn, .584 lb. per minute is evaporated within the compressor during the compression cycle thus maintaining cooling balance of the compressor discharge temperature at a given compression ratio, the total liquid discharged to the atmosphere at the first moisture separator being the difference between the amount of vapor contained at the compressor atmosphere inlet and that in the discharge air from the system as indicated at the receiver **50**. The self-sustaining evaporative cooling system is thereby operational where the compressor atmosphere inlet contains at least an equal amount of water vapor as is being discharged from the compressor system at the receiver with any excess being ejected as liquid into the atmosphere, and the system is thus self-stabilizing for varying conditions of compressor inlet.

Referring now to FIG. 2 of the drawings, the system shown therein illustrates a further adaptation of the invention as applied to a particular system. For example, in the system of FIG. 2, an auxiliary low pressure compressor and closed system is used having a parallel but separate path through the aftercooler condenser and the refrigerated dryer. It will, however, be noted that evaporative cooling principle is still only applied in the case of the so-called high pressure compressor, and the low pressure compressor and its closed system is operative while taking partial advantage of the cooling and condensing elements of the main system.

As shown, the pressure air compressor **70** of FIG. 2 again has a low pressure inlet **72** and a high pressure discharge line **74**. In the specific example, 160 c.f.m. inlet air at 94° F. and 85% relative humidity is admitted to the low pressure inlet line **72** past filter **76**. Under the conditions specified, the inlet 160 c.f.m. air will contain 10.9 lbs. per minute dry air and .326 lb. per minute vapor. With the compressor operating at a discharge pressure of 57 p.s.i.g., water vapor in the inlet air is condensed and recirculated as a coolant, and the system will stabilize at a compressor discharge temperature of approximately 180° F. Under stabilized conditions, the compressor discharge air will saturate with .910 lb. per minute vapor and will carry .326 lb. per minute liquid.

Again, a moisture separator **80** is interposed in the high pressure conduit **74**. This moisture separator **80** is again assumed to be approximately 90% efficient so as to remove and eject 90% of the liquid in the compressor discharge line which again is shown as being conducted through a conduit **82** for discharge to the atmosphere adjacent an aftercooler **84**. The compressed air remaining in the high pressure discharge line **74** beyond the moisture separator **80** is cooled through the aftercooler condenser **84** to 115° F. with 1 p.s.i.g. pressure drop as in the first system.

At the aftercooler inlet, there is .910 lb. per minute vapor and .033 lb. per minute liquid in the 10.9 lbs. per minute air at 185° F. At the outlet of the aftercooler **84**, with a temperature of 115° F., the air will saturate at .145 lb. per minute vapor, and the additional condensation results in a total of .798 lb. per minute liquid. The discharge from the aftercooler **84** is then passed through a second moisture separator **86** interposed in the high pressure line **74**, and this moisture separator **86** is again assumed to be 90% efficient so as to separate .718 lb. per minute of liquid. The liquid separated by the separator **86** is then returned via conduits **88**, **90** back to the compressor inlet at nozzle **92**.

From the moisture separator **86**, the air in the high pressure line **74** is then passed through one section of the refrigerated dryer generally designated by numeral **94**. The refrigerated dryer **94** again consists of an outlet **96**, an air-to-air heat exchanger **98**, a refrigerator unit **100**, and finally a moisture separator **102**. Conditions at the inlet to the dryer **94** are 115° F. with the air containing .145 lb. per minute vapor and .080 lb. per minute liquid. With the evaporator of the refrigerator unit **100** at a temperature of 35° F., the condition of the air within the dryer after passing over the evaporator will be .011 lb. per minute vapor and .214 lb. per minute liquid. At the moisture separator **102**, .192 lb. per minute liquid is separated and connected into conduit **104** for return via line **90** to the compressor inlet **72**. Beyond the moisture separator **102**, the air contains .011 lb. per minute vapor and .022 lb. per minute liquid. As this air is passed through the dryer heat exchanger **98**, it exits from the unit by way of outlet **96** at approximately 70° F., and the liquid has now been revaporized to result in a discharge air content of .033 lb. per minue vapor.

From the outlet **96**, the high pressure air is passed through a filter **108** past a pressure regulator **110** and then to the high pressure receiver **112**. At the receiver, the 10.9 lb. per minute air at 50 p.s.i.g. has a temperature of 70° F. with a 66° F. dewpoint, the air containing .033 lb. per minute vapor and it is mostly dispensed for use in the high pressure system.

In the system illustrated, a combination low pressure and vacuum compressor **120** is provided which has an inlet **122** connected through a vacuum receiver **124** and a pressure regulator **126** with the pressure system. This is essentially a recirculating system and after initial charging it stabilizes so that the relatively low losses through con-

duit **141** are automatically replenished with conditioned air from the high pressure system as controlled through regulator **126**. With the compressor **120** operating at inlet conditions of 70° F. and approximately 12 p.s.i.a. (2.7 p.s.i. vacuum) and a discharge pressure of 4 p.s.i.g. the discharge temperature is approximately 140° F. without evaporative cooling. The output is conducted via discharge conduit **128** to the auxiliary air-to-air aftercooler condenser **130** and then to the air-to-air auxiliary heat exchanger **132** of the refrigerator dryer unit **94** at 105° F. The air is then passed over the evaporator of the refrigerator dryer unit, cooled to 35° F. and 90% of the entrained moisture is separated therefrom by a separator **134** and ejected to atmosphere as by way of conduit **82**. From the air-to-air heat exchanger of the dryer unit, the air at 4 p.s.i.g. exits at 70° F., being temperature controlled, and is conducted via line **136** to a low pressure reeciver **138** with pressure control obtained by regulator **140**, the air being then conducted from the low pressure receiver for use of a low pressure system and then back to a vacuum receiver **124**.

As previously described, relatively small losses occur through conduit **141**. The small losses which occur from the low pressure system through conduit **141** and with make-up air available from the high pressure source permit regulators **126** and **140** to maintain accurate control of pressures in both vacuum receiver **124** and low pressure receiver **138** while utilizing a fixed volume (capacity) compressor **120**.

It will thus be seen that the low pressure system operates mostly in a closed circuit while receiving filtered, cooled dry air from the high pressure system as needed to replenish losses through conduit **141**. The high pressure system is self-stabilizing and self-suporting, and the low pressure system has no other effect on the function and operation of the high pressure system. In each instance, the same components are utilized but different conditions prevail as indicated.

We claim:

1. A cooling system for gas compressors having a low pressure inlet **12** and a high pressure discharge **14**: said cooling system comprising sprayed coil means **24** interposed in the high pressure discharge for cooling the high pressure fluid passing therethrough to cause condensation of vapors entrained therein; means **20**, **26** for separating condensate from the high pressure gas; means **30** for returning a portion of the separated condensate back to the inlet **12** for entrainment with the supply gas to cool same; and means **22** for delivering a portion of the separated condensate to the spray system for the cooling means **24**, whereby to promote evaporative cooling on the coil.

2. The system of claim **1** wherein separator means **20**, **26** and delivery means **22** are sized to substantially prevent pressure loss from the high pressure discharge **14** to atmosphere or to low pressure inlet **12**.

3. The system of claim **1** wherein the separator means comprises two individual separators, one located between the compressor and coil means, and the other located downstream from the coil.

4. The system of claim **1** wherein the inlet **12** includes a venturi nozzle mixing device for assimilating the returning condensate into the supply gas stream.

5. The system of claim **1** and further comprising refrigerated drying means **40** for cooling the gas discharged from cooling means **24**; means **42** for separating condensate from the cooled gas; and means **44** for returning separated condensate back to inlet **12** for entrainment with the supply gas.

6. The system of claim **1** wherein the system is sized so that return means **30** delivers relatively large quantities of condensate to inlet **12** and return means **22** delivers relatively small quantities of condensate to the spray system, whereby the mass flow of fluid through the compressor is increased.

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