June 11, 1957 E. L. KUMM 2,795,115

ABSORPTION REFRIGERATION

Filed May 14, 1954 3 Sheets-Sheet 1

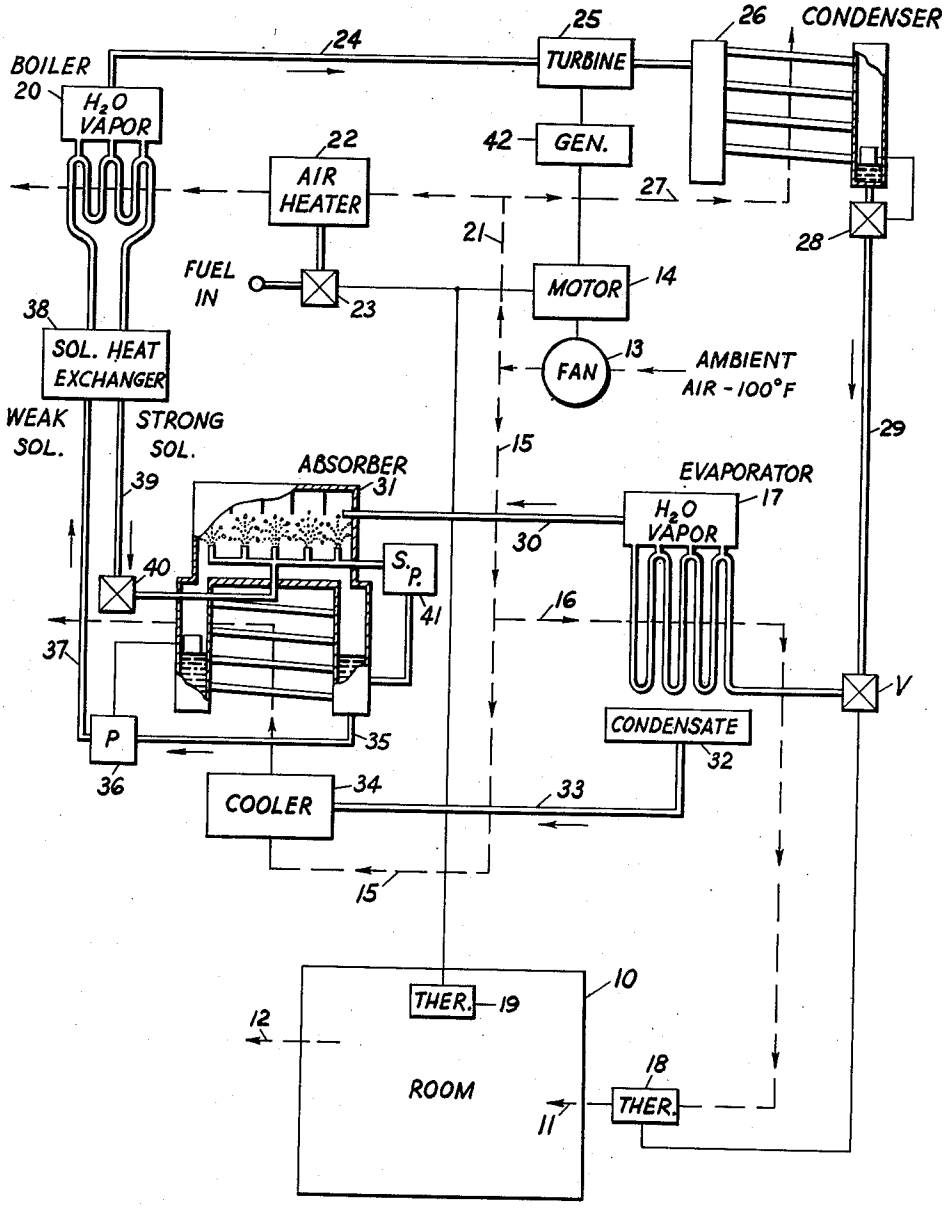

FIGURE 1

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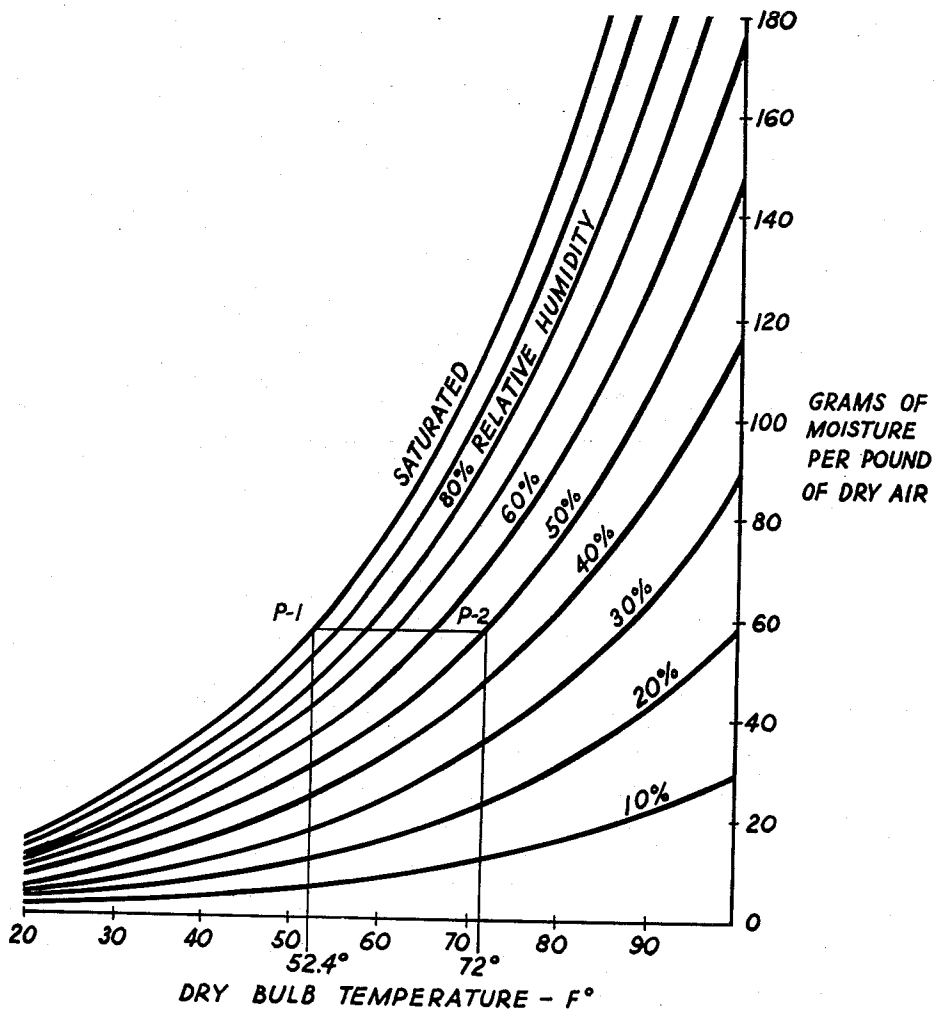

FIGURE 2

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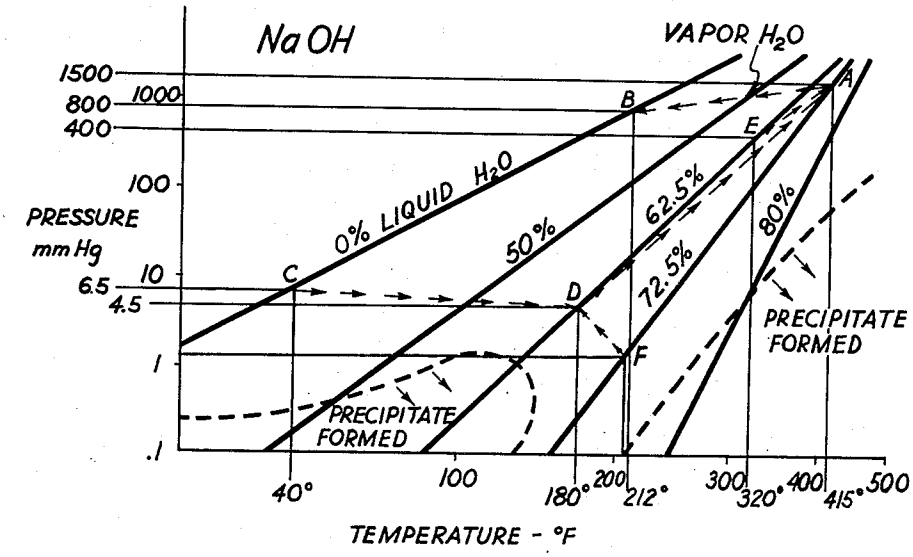

FIGURE 3

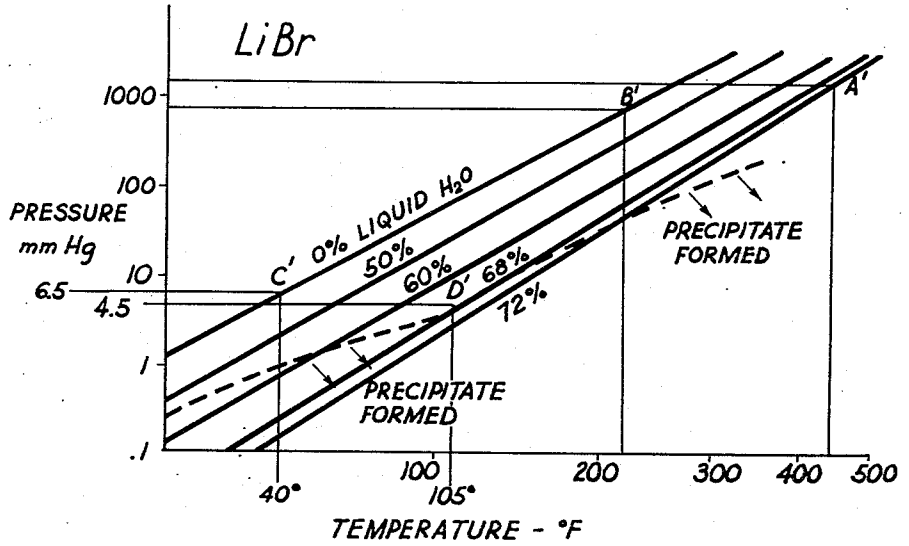

FIGURE 4

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2,795,115

ABSORPTION REFRIGERATION

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4 Claims. (Cl. 62—5)

This invention relates to refrigeration and more particularly, to an improved water vapor absorption refrigeration cycle.

Absorption refrigeration has many advantages over vapor compression refrigeration systems, principally among which are: substantially uniform efficiency under variable loads, fewer moving parts resulting in longer life and quieter operation, and the elimination of bulky driving equipment and electrical power sources.

In the case of small air conditioning units for use in the home, automobile, and in particular portable installations for military uses in the field, the above noted advantages are of special significance. On the other hand, if present-day units of this type are to be kept small, a relatively large source of cooling water is required. Therefore, unless water is substantially free, or water cooling towers are installed, the cost of operating such units is prohibitive.

In instances where cooling water is not available, as is frequently the case in military bases located in deserts or other outlying areas, fan blown air must be relied upon for cooling certain of the components in the system. To provide effective cooling in this manner necessarily requires relatively large heat exchangers in the various components. This requirement not only increases the cost of such units, but materially reduces their portability.

The cooling water problem or the alternative of large and bulky equipment, render military and commercial portable air conditioners now available relatively expensive to manufacture and costly to operate. Probably the most significant single factor in deterring the sales of air conditioning equipment to the public at large in this cost factor.

Bearing the above in mind, it is a primary object of the present invention to provide an improved method and apparatus for conditioning a gas in which the equipment costs and operating expenses are vastly reduced as compared to present day units of equal capacity.

More particularly, an object of the invention is to provide an improved water vapor absorption refrigeration cycle in which no external source of cooling water nor electrical power is required, yet one in which the various components are conveniently small in size so as to provide a readily transportable unit.

Another object is to provide a self contained unit of the above type which, in addition to space cooling, may be adapted for space heating of a desired location.

Still another object of the invention is to provide in an air-conditioning unit, improved means for simultaneously controlling both the temperature and relative humidity of the air.

These and further objects and advantages of the invention are attained by means of a water vapor absorption refrigeration cycle employing as a medium for vapor absorption, solutions of one or more of the soluble basic hydroxides; for example, sodium hydroxide and potassium hydroxide. Certain substances may be present in addition to the soluble basic hydroxide in any percent which does not materially detract from the properties of the basic hydroxide solutions.

Aside from the advantages of greater drying power at higher temperatures, more favorable specific heats, and greater stability, the basic hydroxide solutions are much more soluble at higher temperatures as compared to salt solutions previously used. This high degree of solubility permits greater temperature differences to exist between the ambient air and the various components of the refrigeration apparatus, whereby fan-blown ambient air is sufficient for cooling purposes notwithstanding the relatively small size of the components. In other words, the salts employed in prior art absorption solutions precipitate at relatively low maximum temperatures whereby temperature differences in the refrigeration components must be restricted. Accordingly, the required amount of cooling medium, which is an inverse function of the temperature difference between the cooling medium and the component to be cooled, is relatively large.

In accordance with further features of the invention, a preferred form of refrigeration apparatus employing the basic hydroxide solutions, incorporates a turbine capable of generating sufficient power to circulate both the absorbing solution and the gas to be conditioned. Sufficient energy is present in the system for driving such turbine in view of the high temperature differences between the unit components permitted by the use of the basic hydroxide solutions.

In a preferred embodiment of the present invention, two thermostats coupled to the refrigeration aparatus are respectively positioned in the gas flow path and in the location containing the conditioned gas, for maintaining the temperature and relative humidity of the gas at desired values.

A better understanding of the invention will be had by referring to the accompanying drawings, in which:

Figure 1 is a schematic drawing of a typical water vapor absorption refrigeration system incorporating the various features of the present invention;

Figure 2 is a conventional psychometric chart for air well known in the air-conditioning industry;

Figure 3 is a chart exhibiting the water absorption and solubility characteristics of a sodium hydroxide solution, typical of the basic hydroxide solutions of the present invention; and

Figure 4 is a chart similar to Figure 3, but giving the properties of a lithium bromide solution typical of absorption media employed heretofore.

Referring to Figure 1, there is schematically shown in the lower portion of the drawing, a room or desired location **10** to be air-conditioned in accordance with the invention. Treated air is passed into the room at **11** and withdrawn or exhausted at **12**. This exhaust air may be expelled to the atmosphere or re-cycled through the conditioning apparatus.

As a specific example of conditions under which the air conditioning unit of the present invention operates, assume that the ambient or outside air temperature is 100° F. and that it is desired to maintain the temperature in room **10** at 72° F. and at a relative humidity of 50%.

Referring to the central portion of Figure 1, the ambient or environmental air is circulated through the various air conditioning components to room **10** by means of one or more fans or blowers such as the fan **13** operated by motor **14**. As shown schematically by the dashed lines, this ambient air flows down an air passage **15**, branch passage **16**, and past the coils of an evaporator **17** to the room **10**. The fan blown air does not directly contact the solutions employed in the refrigeration aparatus, nor the liquid water condensed from these

solutions. Heat is exchanged from the fan-blown air to the liquids concerned through pressure-tight metal walls forming heat exchangers in the various components of the refrigeration apparatus. The fan-blown air passing the coils of the evaporator **17** is normally cooled below its dew point with the resulting condensation of moisture from the air resulting in cooled saturated air entering the room **10** at **11**.

In accordance with a feature of the present invention, a first thermostat **18** is disposed in the entering air path **11** and maintains the temperature of this cooled saturated air at a predetermined value depending on the relative humidity desired in the room **10**. A second thermostat **19** maintains the air in room **10** at the desired temperature. The manner in which these thermostats maintain these temperatures is through control of the water vapor absorption cycle of refrigeration and will be explained subsequently. In normal use, the cold saturated air from the evaporator **17** entering room **10** absorbs heat from the room and its occupants resulting in an increase in its temperature and a decrease in its relative humidity, so that the final mean air temperature and humidity when mixed with the rest of the air in the room is comfortable to the occupants.

Referring to the conventional psychometric chart of Figure 2, it will be seen that if the saturated cooled air entering the room is maintained at a temperature of 52.4° F. by the thermostat **18** (point P–1 in Figure 2) warming of this air to 72° F. results in a unique relative humidity of 50%. (Point P–2 in Figure 2.) Since the absolute moisture content or grams of moisture per pound of air remains substantially constant upon warming the air, the point P–2 is at the same horizontal level as the point P–1, and inasmuch as thermostat **19** maintains the temperature at 72° F., it is clear that both the temperature and relative humidity in the room **10** may be controlled, within limits, by only the two thermostats **18** and **19**. Changing the setting of thermostat **18** to a new predetermined temperature value enables a different desired relative humidity to be had in room **10**, as will be readily perceived by an inspection of the chart in Figure 2.

The water vapor absorption refrigeration cycle for contioning the air passed into the room **10** in accordance with the invention will now be described. For the sake of concreteness, specific values of temperatures and pressures will be assumed. It is to be understood of course that such values are merely illustrative.

Referring to the upper left hand portion of the refrigeration uint shown in Figure 1, water vapor is boiled from an absorbing solution in a boiler **20**. This boiler is heated by means of air passed from the fan **13** through an air passage **21** and an air heater **22**. Fuel for the air heater **22** is introduced through an ignitor and inlet valve **23** and is sufficient to heat the fan blown ambient air from 100° F. up to temperatures in the neighborhood of 1375° F. This heated air, on passing through the heat exchanger tubes of the boiler **20**, will be expelled at a temperature of, for example, 680° F., and may be used, if desired, for space heating purposes. The heat exchange within the boiler **20** is such as to heat the solution to a temperature of about 415° F. at a vapor pressure of approximately 1500 mm. of mercury.

Figure 3 is a chart exhibiting the characteristics of the absorbing hydroxide solution of the present invention, in this instance the solution being sodium hydroxide. The ordinates of Figure 3 indicate the water vapor pressure in equilibrium with varying solution concentrations at certain temperatures. The abscissae indicate the temperature at which a varying solution concentration is in equilibrium with a constant water vapor pressure. The various graph lines in the chart represent varying concentrations of the sodium hydroxide salt solutions from 0% representing liquid water, to 80%. The curved broken lines define the upper temperature and lower pressure limitations at which the salt solution will precipitate.

Referring to both Figures 1 and 3, the point A in Figure 3 represents the conditions of the solution in the boiler **20**. At this point on the curve, it will be seen that the concentration of the solution is approximately 72.5% and is at a temperature of 415° F. and a pressure of 1500 mm. of mercury.

The water vapor is passed from the boiler **20** of Figure 1 along a conduit **24** through a turbine **25** and to a condenser **26**. Condenser **26** is cooled by the ambient air from fan **13** guided through a suitable branch air passage **27** as shown. By operating the condenser **26** at 212° F., the temperature difference between the condenser and the 100° F. ambient air is large, enabling the heat exchanging surfaces in the condenser **26** to be minimized. As the water vapor condenses at a temperature of approximately 212° F. in the condenser **26**, the conditions in the condenser will be represented by the point B in Figure 3. At this point, the solution is liquid water at a vapor pressure of about 800 mm. of mercury.

The liquid level in the condenser **26** is maintained at a constant value by a float controlled valve **28**. This liquid condensate passes through a conduit **29** and flow regulator valve V into the coils of the evaporator **17** wherein evaporation reduces the temperature of the water to approximately 40° F. at a pressure of 6.5 mm. of mercury. This condition is indicated by the point C in Figure 3.

At the temperature of 40° F., the minimum temperature differential allowable for cooling the air passing through the coils of the evaporator **17** in order that the exit air from the evaporator have a temperature of 52.4° F., is then 52.4—40=12.4° F. The required size of the heat exchanger in the evaporator is proportional directly to the allowable minimum temperature differential in a manner well known to the designers of heat transfer equipment. Consequently, the water evaporation temperature must be as low as possible considering the limitations of ice formation at 32° F. and the equilibrium vapor pressure over the weak solution.

Water vapor from the evaporator **17** passes through a conduit **30** into an absorber **31**. The conditions in the absorber **31** are given by point D in Figure 3 wherein an equilibrium water vapor pressure of 4.5 mm. of mercury for a weak solution of 62.5% concentration exists at the relatively high temperature of 180° F. Thus a water vapor pressure differential of 6.5—4.5=2 mm. of mercury exists between the water in the evaporator **17** and the weak solution in the absorber **31**. This pressure differential causes the evaporation of the water in the evaporator and its subsequent absorption in the absorber.

Cooling air is blown through the heat exchanger portion of the absorber **31** by way of the air passage **15** from fan **13** and serves to remove the heat in the absorber resulting from the absorption of the water vapor. Of major importance is the consideration that the temperature differential between the weak solution in the absorber **31** and the fan-blown air is quite large, even for high ambient air temperatures, which enables a considerable reduction in the size of the required heat exchanger. If the ambient air temperature is 100° F., as assumed, an initial temperature differential of 80° F. results across the heat exchanger of the absorber **31**. This differential could be increased further by the use of more concentrated solutions in the apparatus. Also, a further reduction in size of the heat exchanger in the absorber **31** may be realized by pre-cooling the fan-blown ambient air. This pre-cooling may be conveniently accomplished by collecting the water condensate from the coils of the evaporator **17** by means of a suitable collection pan **32**, and passing the same through the conduit **33** to a humidifying cooler **34** in the air passage **15** just prior to entry of the air to the absorber heat exchanger. With this arrangement it is possible to pre-cool the ambient air significantly. The tempera-

ture differentials that exist between the operating point D and the points at which precipitates would form, as indicated by the dashed lines in Figure 3, assures operational feasibility.

The weak solution from the absorber **31** passes through a lower conduit **35**, solution pump **36**, conduit **37**, and through a solution heat exchanger **38** to the boiler **20**. As previously stated, the hydroxide concentration of the solution in the boiler of 72.5% at a temperature of 415° F., yields an equilibrium water vapor pressure of about 1500 mm. of mercury. Therefore, the weak solution from the absorber **31** must be pumped from the pressure of 4.5 mm. of mercury to 1500 mm. of mercury, whereas the strong solution resulting from the boiling and returning to the absorber by way of a conduit **39**, may merely be throttled as by valve **40** to maintain the proper rate. To increase the rate of absorption, it is desirable to spray the returning strong solution from conduit **39** by means of a header pipe and spray chamber in the absorber **31** as shown. If such spraying resulting from the throttling action of the discharge valve **40** is not sufficient, an additional spray pump **41** may be employed for circulating the solution within the absorber.

The counterflow heat exchanger **38** results in discharge temperatures represented for the weak and strong solutions, respectively, by the points E and F in Figure 3. The chief purpose of the solution heat exchanger **38** is to conserve heat in the system.

The water vapor generated in the boiler **20** and passing through the turbine **25**, results in a pressure loss through the turbine which is dependent on the work loading of the turbine. With the use of the hydroxide solutions, sufficient pressure and temperature differences between the boiler and condenser exist to enable this turbine to drive a generator **42** providing sufficient electrical power to drive the fan motor **14** and pump **36**.

As mentioned previously, the temperature and relative humidity in the room **10** are controlled by both the thermostats **18** and **19**. As shown in Figure 1, the thermostat **18** in the incoming air path **11**, is operatively coupled to the flow regulator valve V in conduit **29**. This valve will regulate the flow of liquid water from the condenser **26** to the coils of the evaporator **17** and thereby control the degree of cooling of the evaporator coils resulting from evaporation of the liquid water. Accordingly, the temperature of the air passing through the evaporator coils along passage **16** and entering the room at **11** may be controlled. For example, if the temperature of the air increases above the predetermined value of 52.4° F., the thermostat **18** will operate to increase the opening of the regulator valve V to permit a greater water flow to the evaporator and thereby increase the cooling rate. On the other hand, if the temperature of the air from the evaporator drops below 52.4° F., the thermostat **18** will operate to decrease the opening of the regulator valve V thereby decreasing the liquid flow to the evaporator with a consequent decrease in the rate of cooling.

The thermostat **19** in the room **10** for maintaining the mean temperature in the room at 72° F. is operatively coupled, as shown in Figure 1, to the fuel inlet valve and ignitor **23** and the fan motor **14**. If the temperature in room **10** exceeds 72° F., the thermostat **19** will operate to speed up the fan motor **14** to increase the flow rate of cooled air to the room. On the other hand, if the temperature drops below 72° F., the thermostat **19** will operate to decrease the speed of the fan motor **14** to slow down the flow rate of cooled air to the room. Also the fuel inlet rate to the heater may be increased or decreased should the room temperature increase or decrease above or below 72° F. If the ambient or environmental air temperature is below 72° F. whereby the temperature in room **10** will tend to be lower, the thermostat **19** may be adjusted to shut down the air conditioning apparatus by closing the fuel valve **23** and turning off the fan motor **14**.

On the other hand, the apparatus of the present invention may also be used for space heating purposes. For example, the heated air passing from the boiler **20** and/or the air passing from the condenser **26** may be directed by suitable air passages (not shown) to the room **10**, and the cold air from the evaporator exhausted to the environment.

It is to be emphasized in the water vapor absorption refrigeration cycle described in connection with Figure 1, that the sole use of fan blown air and the avoidance of water cooling systems and water towers is possible in view of the unique properties of the absorption medium exhibited in Figure 3. In fact, it is the limitations of prior art absorption media that have heretofore prevented such a system as shown in Figure 1 from being operationally practicable.

To emphasize the advance in the refrigeration art afforded by the present invention, Figure 4 discloses the limitations of a lithium bromide solution, typical of prior art absorption media. In this chart, wherein the ordinates and abscissae are the same as shown in Figure 3, it will be noted that the dashed lines, indicating the pressure and temperature limitations at which precipitation occurs, are far more restricting than in the case of the sodium hydroxide solutions shown in Figure 3. The corresponding operational points in the various apparatus components labeled A, B, C, and D in Figure 3, are correspondingly labeled A′, B′, C′, and D′ in Figure 4. Note in Figure 4 that the water vapor at 40° F. passing from the evaporator into the absorber results in an absorber operation temperature of 105° F. at a corresponding vapor pressure of 4.5 mm. as indicated at the point D′. If the absorber were operated at a higher temperature at the vapor pressure of 4.5 mm., it will be seen at once that the salt would precipitate.

With the absorber at a temperature of 105° F. and fan blown air at 100° F., the size of the heat exchanger in the absorber would be economically impractical. The only manner of increasing the absorber temperature without precipitation would be to increase the evaporation temperatures in the coils of the evaporator **17** for a given line pressure loss between the points C′ and D′ of Figure 4. However, this temperature increase would materially increase the evaporator heat exchanger size required to cool a given quantity of air to 52.4° F. as given before. Also, the air and power requirements for the absorber, using lithium bromide solution, would be significantly increased. With the absorber at 105° F. and an ambient cooling air of 100° F., the lithium bromide absorber heat exchanger would have an initial temperature differential of 5° F. whereas the basic hydroxide absorber would have an initial temperature differential of 80° F. A decrease in temperature differential across a heat exchanger not only increases the heat exchanger surface required to transfer a given amount of heat, but also increases the flow and power requirements in a manner well known to designers of heat exchanger equipment.

On the other hand, it is relevant in this connection to point out that a refrigeration and dehumidification apparatus using lithium bromide solutions is mechanically practical if water or air at 80° F. or lower are used as a cooling medium in the heat exchanger of the absorber **31**. However, the new and novel arrangement using the basic hydroxide solution as the absorber, permits the new and practical application of using only ambient air, even at relatively high temperatures, as the medium and results in the design of compact apparatus not requiring any water as the coolant medium.

From the preceding comparison, the refrigeration and dehumidification apparatus using the ambient air for cooling is shown to be practical only in case sodium hydroxide or other basic hydroxide solutions of similar absorption

characteristics to the sodium hydroxide solution displayed in Figure 3, are used in the water vapor absorption system.

While the present invention has been described in connection with a portable air-conditioning unit, it is to be understood that the principles thereof are applicable to any type of gas conditioning apparatus, whether for comfort cooling or other purposes. The invention therefore, is not to be thought of as limited to the particular embodiment and operational values disclosed for illustrative purposes.

I claim:

1. A water vapor absorption refrigeration apparatus including: boiler, condenser, evaporator, and absorber components; fan means for circulating ambient air to said components, said apparatus employing basic hydroxide solutions as the absorption medium whereby sufficient temperature differences between said absorber and environmental air exist for said fan blown air solely to cool said absorber; means for heating air passed to said boiler; and means for cooling air passed to said absorber.

2. An apparatus according to claim 1, including a turbine operatively connected between said boiler and condenser, and means for transferring energy from said turbine to said fan means.

3. A water vapor absorption refrigeration apparatus including: boiler, condenser, evaporator, and absorber components; means for circulating ambient air to said components; first means including an air heater disposed in the path of said circulating ambient air for heating the ambient air passed to said boiler; and second means disposed in the path of the ambient air circulated to said absorber including means for cooling said ambient air and passing the cooled ambient air to said absorber whereby said absorber is cooled, said refrigerating apparatus employing basic hydroxide solutions as the absorption medium.

4. An apparatus according to claim 3, in which ambient air circulated to said evaporator is cooled by said evaporator to a predetermined temperature at which water vapor will condense and said air will be saturated; said refrigeration apparatus including a first thermal responsive element disposed in the path of the air leaving said evaporator; means responsive to said thermal responsive element for controlling the operation of said evaporator whereby said air leaving said evaporator is maintained substantially at said predetermined temperature; means for passing said air to a desired location; a second thermal responsive element disposed in said desired location; means responsive to said second thermal responsive element for controlling said means for circulating ambient air for maintaining said air in said location at a desired temperature different from said predetermined temperature, whereby a desired relative humidity of said air exists at said desired temperature.

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