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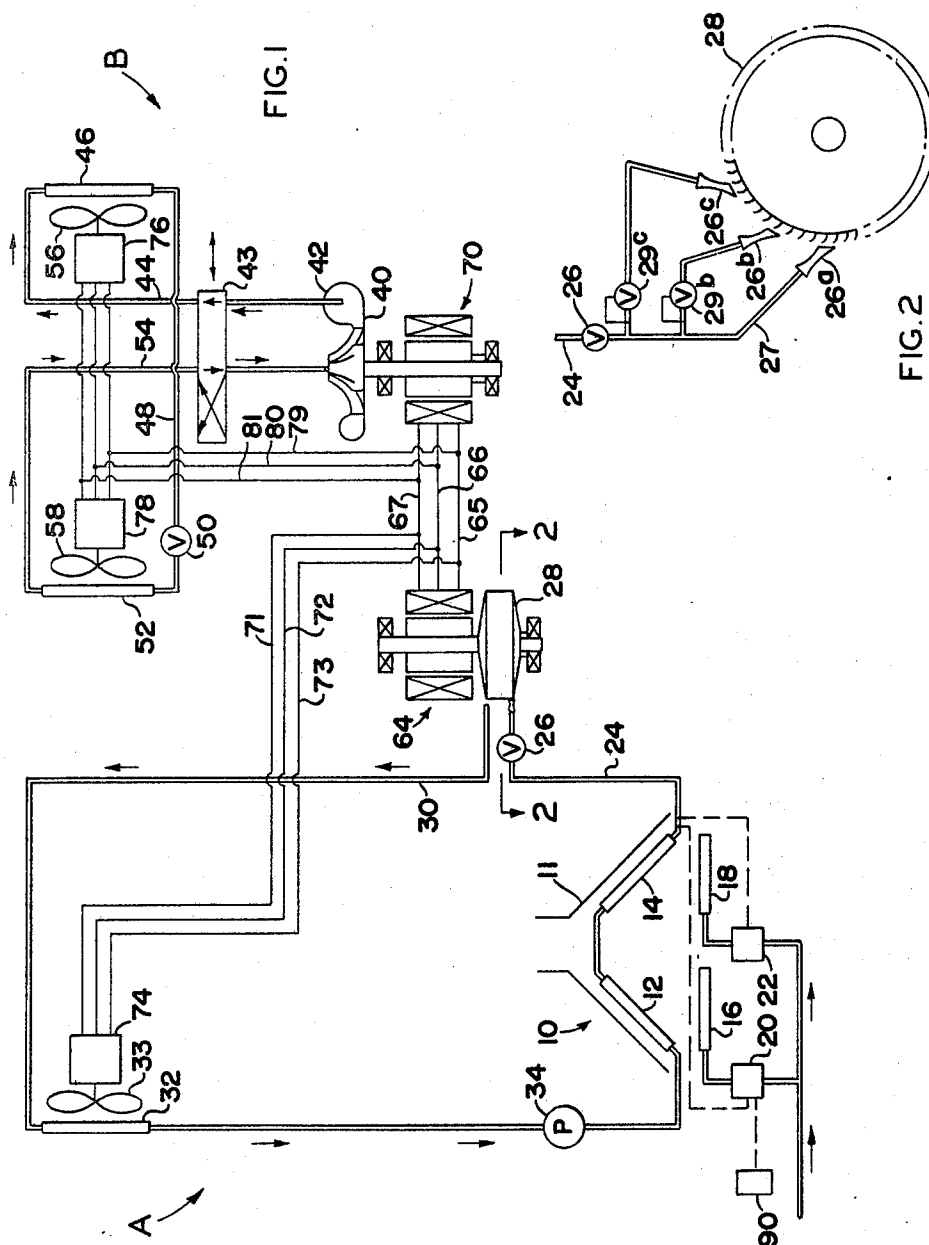
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HEAT-PUMP SYSTEM

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2 Sheets-Sheet 1



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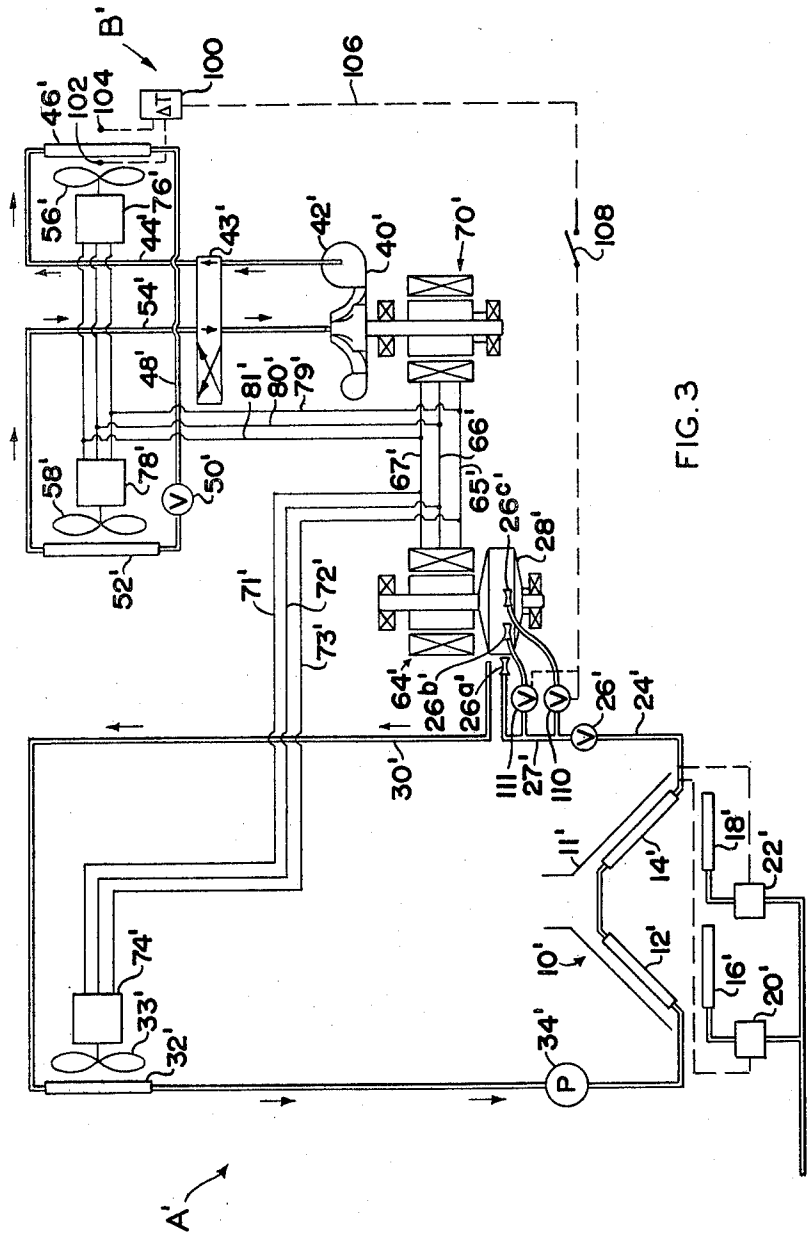


FIG. 3

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HEAT-PUMP SYSTEM

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10 Claims

ABSTRACT OF THE DISCLOSURE

An improved expansion-type heat-pump system which includes a vapor-turbine which is drivingly interconnected with the compressor and evaporator fan through an alternator-motor arrangement which permits control of cooling output to be accomplished by simple modulation of turbine speed.

The present invention is directed toward the art of heat-pumping and refrigeration and, more specifically, to an improved vapor turbine driven reversible-type heat-pump system.

The invention is particularly suited for constructing reversible heat-pump systems of a size for use in heating and cooling residential dwellings, and will be described with particular reference thereto; however, it will be appreciated the invention is capable of broader application and could be utilized for constructing systems of a variety of sizes and for cooling alone or reverse cycle heating and cooling operation.

As discussed in the commonly assigned, copending application Ser. No. 624,063, to Swenson et al., filed Mar. 17, 1967, now U.S. Patent No. 3,400,554 issued Sept. 10, 1968, many attempts have been made to provide a self-contained, vapor-turbine driven refrigeration system suitable for residential use. The problems involved in providing such a system are many, but one of the main problems has been in providing adequate sealing between the turbine and the compressor so that undesired mixing of the respective fluids in the vapor and refrigeration cycles does not take place. Additionally, problems have been encountered in controlling the two respective cycles to match their output to the demand, especially when modulated output is desired.

In the above-noted copending application, the sealing problems are overcome by drivingly interconnecting the turbine and compressor through a magnetic coupling. A stationary impervious membrane between the rotating components of the coupling permit complete sealing between the turbine and the compressor without the use of rotary mechanical seals, etc. This arrangement has constituted a marked step forward in the art of vapor-turbine driven compressor units; however, it has not been fully satisfactory for use in commercial units.

The noted system, like prior systems, for modulated operation, requires a somewhat complicated control set-up to assure matching of component output with system demand. Accordingly, the overall system has certain inherent limitations.

The present invention overcomes the above-noted problems of prior systems and provides a vapor-turbine driven reversible heat-pump system which permits response to less than design cooling loads by simple modulation of the turbine speed.

In accordance with the invention, there is provided an improved reversible heat-pump system which includes a closed-loop, expansion-type refrigeration cycle comprising a rotary compressor supplying compressed fluid

through an expansion valve to an evaporator across which air to be cooled is directed by a fan. An alternator driven by a vapor-turbine supplied with vapor from a closed cycle generating unit, has its output directly connected by ordinary electric conductors to electric motors which drive the compressor and the fan. Additionally, means are provided for varying the speed of the turbine in a modulated manner in response to changes in need for cooling or heating output.

The compressor's output in heat-pumping is very sensitive to its rotational speed. Because there is a substantially direct proportion between the speed of the turbine and the resultant speed of the fan and compressor, the cooling output of the system can be controlled simply by controlling the speed of the turbine. This produces a greatly simplified control arrangement. Additionally, for most systems, a relatively small range of turbine speeds (approximately 60–100% of design speed) covers the complete range of cooling or heating demands. This produces the following operational advantages:

(1) Increased overall, or total system performance energy ratio (i.e. heat pumped divided by energy supplied) because of a reduced compressor pressure ratio requirement and fan power load much of the time. For example, depending on system components and size, a minimum of 15% increase in total system performance energy ratio can be obtained in the range of 20 to 70% of the system's maximum cooling capacity, which corresponds to a major portion of the system's overall operation.

(2) Decreased system fan noise level at less than maximum demand operation since fan speed is directly proportional to system speed and therefore full speed fan operation takes place only when maximum system cooling or heating output is required. Modulation of fan speed along with the cooling or heating effect is very desirable, and difficult to achieve in other cooling or heating systems. A particular advantage in cooling is that reduced fan speed tends to keep the cooling (evaporator) coil at a lower temperature than would otherwise be the case at reduced compressor speed operation, and thus retain the system's ability to remove moisture from the dwelling air.

(3) Increased system service life since the key components are subjected to only a corresponding portion of maximum design service stress when operating at less than the design point.

(4) The system can modulate down to zero effective cooling capacity (i.e. 60–70% design speed for most applicable compressors) without additional control function and drive only the fans to provide continuous dwelling air circulation.

(5) The system can operate continuously in a speed range of from approximately 60% (no cooling) to 100% (design cooling load), thereby avoiding start-up and shut-down which is troublesome in low-load, high speed shaft-bearing systems of the type utilized in small turbines and compressors. In addition to the above-mentioned advantages, the use of the alternator-motor interconnection between the turbine and the compressor eliminates all need for rotating mechanical seals. Thus, there are no problems in using the most ideal fluids in both the vapor and reversible heat-pump cycles.

Accordingly, a primary object of the present invention is the provision of a vapor-turbine driven, expansion-type reversible heat-pump system wherein the turbine and compressor are drivingly interconnected through an alternator-motor arrangement.

A further object is the provision of a system of the type described wherein the modulated level of cooling or heating output is controlled simply by modulation of the turbine speed.

Another object is the provision of a reversible heat-pump or cooling system wherein compressor and fan speed are simultaneously controlled in response to the heat pumping or cooling requirement.

Yet another object is the provision of a vapor-turbine driven expansion-type reversible heat-pump system which eliminates the need for rotating seals between the turbine and the compressor.

A still further object is the provision of a system of the general type described wherein modulation of turbine speed throughout a limited range permits cooling or heating output to be varied from zero to 100% of design output.

These and other objects and advantages will become apparent from the following description when read in conjunction with the accompanying drawing wherein:

FIGURE 1 shows, somewhat diagrammatically, a reversible heat-pump system constructed in accordance with a preferred embodiment of the invention;

FIGURE 2 is a detail view of the steam supply arrangement; and

FIGURE 3 is a view similar to FIGURE 1 showing a second control set-up.

Referring specifically to FIGURE 1, it is seen that the improved reversible heat-pump system includes a closed loop vapor cycle A and a conventional, closed loop expansion-type reversible heat-pumping cycle B. In general, the closed loop vapor cycle A is constructed in basically the same manner as the vapor cycle shown in the commonly assigned copending application Ser. No. 624,063 filed Mar. 17, 1967. Specifically, as shown, it includes a fuel fired once through type boiler unit 10 comprising a housing 11 including serial connected evaporator and superheater section 12 and 14, respectively. Preferably, the boiler is gas fired and includes separate burners 16 and 18 arranged to heat the evaporator section 12 and the superheater section 14 respectively. The supply of fuel to burner 16 is preferably controlled by a pressure responsive valve 20 responsive to the pressure of the steam leaving superheater section 14 modified in a manner to hereafter be described. A temperature responsive valve 22 functions to control the supply of gas to burner 18 and is preferably controlled in response to the temperature of the steam leaving the superheater section 14.

The steam output from superheater section 14 is conducted through a line 24 and a shut-off valve 26 to the inlet of a conventional Terry-type steam turbine 28 through a valved nozzle arrangement 27 (see FIGURE 2) which will subsequently be described in detail. The exhaust steam from the turbine 28 passes through a line 30 to a coil-type, air cooled condenser 32 and thence through a condensate pump 34 which returns the condensed fluid back to the inlet of the evaporator section 12. A fan 33 functions to supply the necessary cooling air across condenser 32.

Preferably, the vapor circuit is hermetically sealed so as to eliminate the necessity for supplying makeup water to the system. This is desirable, especially in units designed in the size range suitable for residential use, since the addition of makeup water systems tend to unduly complicate the system and introduce the possibility of foreign material entering the system.

The closed-loop, expansion-type reversible heat-pump cycle B is shown as a relatively conventional arrangement comprised of a standard high speed centrifugal compressor 40 which has its outlet 42 connected through a two-position mode valve 43 with a pair of heat exchangers 46 and 52. A line 48 including an expansion valve 50 connects the heat exchangers 46, 52 in serial flow relationship. Lines 44, 54 connect the opposite sides of heat exchangers 46, 52 with mode valve 43. As is apparent, this arrangement allows the heat exchangers 46, 52 to function alternately as either condenser or evaporator depending upon the position of mode valve 43. With the valve 43 in the position shown, heat exchanger 46 functions as

the condenser and heat exchanger 52 functions as the evaporator, and, in the description of operation, as appears hereafter, the heat exchangers 46, 52 will be referred to as the condenser and evaporator respectively.

The specific fluid utilized in the refrigerant cycle B is not of particular importance, but is preferably one of the commonly available refrigerants, such as a member of the fluorocarbon family, selected in accordance with maximum "cycle B" loop efficiency.

As is standard, fans 56 and 58 are provided to direct the required air flow across the condenser coil 46 and the evaporator coil 52 respectively. The ducts necessary to supply the conditioned air to the area being cooled are not shown but could, for example, be of the type shown in the commonly assigned patent application Ser. No. 624,063, filed Mar. 17, 1967.

Of particular importance to the present invention are the means utilized to drivingly interconnect the turbine 28 and the compressor 40. As previously discussed, in the past it has been the practice to directly interconnect the output shaft of the turbine with the input shaft of the compressor and to utilize special rotating seals in an attempt to prevent mixing of the fluids in the vapor cycle and the refrigerant cycle. According to the present invention however, the turbine 28 has its output shaft directly connected to the input shaft of an alternator unit 64 of conventional design. The electrical output of alternator 64 is directly connected by conductors 65, 66 and 67 to a high frequency motor 70 which drives compressor 40. Preferably, the alternator 64 and the motor 70 operate in a range of up to approximately 1200 c.p.s. Additionally, the output of the alternator 64 is also directly connected through conductors 71, 72 and 73 with a conventional high frequency motor 74 which drives steam condenser fan 33. The "cycle B" loop condenser and evaporator fans 56 and 58 are likewise driven by high frequency motors 76 and 78 which are similarly supplied with current from alternator 64 through conductors 79, 80 and 81.

As is well known, the output of a rotary or centrifugal compressor increases substantially with its speed of rotation. Likewise, the speed of rotation of motor 70 is, of course, directly dependent on the speed of rotation of the alternator 64 which is in turn governed by the speed of rotation of the turbine 28. Accordingly, by regulating the speed of rotation of turbine 28 the cooling or heating output of the cycle B is controlled. A variety of different controls could be utilized for modifying the speed of rotation of turbine 28; however, according to the preferred embodiment the means utilized comprise three sequentially actuated nozzles 26a-c directing steam to turbine 28. During operation, valve 26 is always open and continuously supplies steam through line 27 to nozzle 26a to maintain turbine 28 rotating at a predetermined minimum speed. Nozzles 26b and 26c are respectively supplied with conventional steam pressure responsive valves 29b and 29c. The pressures at which valves 29b and 29c open are selected so that they will sequentially open upon preselected rises in pressure in line 27. For example, valve 29b is arranged to open and supply sufficient steam to drive turbine 28 at an intermediate speed whereas, thereafter, upon a further increase in system demand as evidenced by further pressure increase, valve 29c opens to supply steam sufficient to drive the turbine at its maximum design speed.

In order to vary the output steam pressure of the vapor generator 10 in response to system demand, a thermostat 90 responsive to dwelling air is provided. Thermostat 90 is a conventional proportioning thermostat and transmits a signal proportional to the difference between the dwelling air temperature and its set point temperature. The signal is utilized in conventional manner to modify the setting of valve 20 to provide the required pressure output from the vapor generator thereby providing automatic regulation of the speed of turbine 28. No attempt need be made to sense the speed of the turbine directly, as the

speed will simply increase or decrease in response to the number of nozzles activated and the increasing or decreasing pressure applied to each nozzle until the dwelling air temperature has stabilized with respect to the thermostat 90 set-point temperature. It should be pointed out however, that in some applications, some signal other than the dwelling thermostat 90 difference signal would be preferable, such as "cycle B" loop fluid temperature at some point. Likewise, the turbine shaft speed need not be controlled by the method described. A simple throttle valve upstream of a single nozzle could be utilized, but would not be as efficient in operation as the sequencing method described.

In addition to the regulation of the output of compressor 40 in response to variations in the speed of turbine 28, the various fans 33, 58 and 56 are simultaneously regulated producing a varying air flow across the various heat exchangers in response to the cooling demand as indicated by thermostat 90. That is, the output of the fan motors varies in accordance with the turbine speed.

It should be noted that when the thermostat 90 senses a dwelling air temperature set point temperature difference of zero, the system components still rotate under the action of the single nozzle 26a. If this rotative speed is selected at a level below that which the compressor contributes heat-pumping or cooling effect (60-70% or less of maximum operating speed for most applicable compressors), continuous fan operation for the purpose of providing continuous circulation of dwelling air is achieved with a minimum of energy input to the system.

As can be appreciated, the described arrangement eliminates the need for separate controls for the various fans and the compressor. This results in a substantial savings in control circuitry and it additionally provides a system which is fully self-regulating and modulating and simple in control circuitry and it additionally provides a system components, the system's output can vary from 0 to 100% of design with a speed change in the turbine of from 60-100% of design speed.

FIGURE 3 illustrates a second embodiment of a system formed in accordance with the invention. The system of FIGURE 3 is primarily the same as the FIGURE 2 system with the exception of the control set-up. Accordingly, in FIGURE 3, the same reference numerals differentiated by the addition of a prime (') suffix have been utilized to denote the elements which correspond to the FIGURE 2 system. Basically, the FIGURE 3 system makes the system speed responsive to the imposed temperature gradient upon the heat exchanger 46', i.e. the condenser when the system is operated on the cooling mode.

As is well known, because of differences in location and radiation effects, demand is not necessarily proportional to the outside or inside ambient temperature/inside set point differential. Additionally, it is desirable to regulate the evaporator temperature to a narrow range while cooling so that adequate dehumidification is obtained. For this reason, the FIGURE 3 system is arranged to control the centrifugal compressor 40' so that it runs at a speed commensurate with the pressure ratio dictated by the most ideal temperature gradient across heat exchange 46', in effect, to optimize heat rejection.

Specifically, as shown in FIGURE 3, a differential temperature sensor 100 including sensors 102 and 104 is arranged to generate a signal indicative of the temperature change across heat exchanger 46'. The output from sensor 100 is transmitted through a line 106, including a reversible room thermostat 108, to valves 110 and 111 which respectively control steam supply to the turbine nozzles 26c' and 26b'. Although the output from sensor 100 is preferably electrical, it could, of course, be pneumatic, in which case thermostat 108 and valves 110, 111 would be of the pneumatically operated type.

In operation, 26' is arranged to provide a continuously open steam flow through nozzle 26a' sufficient to maintain turbine 28' rotating at the minimum system speed i.e. zero output from compressor 40', and fans 56', 58' and 74' maintaining air flow at minimum flow requirements. Upon a change in room temperature indicating a need for cooling (assuming the system is in cooling mode) thermostat 108 closes. Consequently, the output of sensor 100 is thus conducted to valves 110 and 111. Assuming that the differential temperature sensed across condenser 46' is low, indicating a need for greater turbine speed, both valves 110 and 111 are opened. Thereafter, as the differential temperature increases indicating turbine speed beyond that required, one of the valves, for example, valve 110, is arranged to close. Should the differential temperature increase still further the second valve is arranged to close and air circulation only operation provided.

In the reversed cycle (heating) mode operation the inside coil 52' (functioning as the condenser) is regulated. As can be appreciated, the object is to optimize heat absorption at the outside coil 46', which functions as the evaporator. This is achieved by the same control set-up and measuring the temperature change across the coil 46' and controlling steam admission to the turbine cause the compressor to run at the speed which produces the pressure ratio which maximizes the heat transfer capacity of the evaporator coil 46'.

Having thus described our invention, we claim:

1. A refrigeration system including: a rotary compressor; means for conducting compressed refrigerant from said compressor through an expansion valve and a heat exchange coil adjusted to function as an evaporator and back to said compressor, a fan for directing air across coil; a vapor turbine; an alternator directly driven by said turbine; a first electric motor drivingly connected to said compressor; a second electric motor drivingly connected to said fan; conductors for directly connecting the electrical output of said alternator with said electric motors; and, control means, including thermostat means for controlling the supply of vapor to said turbine in accordance with the cooling output required from said system.

2. A refrigeration system as defined in claim 1 wherein said control means includes a plurality of sequentially operable nozzles supplying vapor to said turbine.

3. A refrigeration system as defined in claim 1 wherein said control means includes a plurality of valves positioned in parallel and sequentially opened in response to increase in the pressure of the vapor being supplied to said turbine.

4. A refrigeration system as defined in claim 1 wherein said thermostat means controls the pressure of the vapor supplied to said turbine.

5. A refrigeration system as defined in claim 1 wherein said refrigeration system further includes a once through, gas fired vapor generator for supplying vapor to said turbine.

6. A refrigeration system as defined in claim 5 wherein said thermostat means controls the quantity of gas supplied to said vapor generator.

7. The refrigeration system as defined in claim 1 wherein said control means includes means responsive to the change in temperature of the air flowing across said coil.

8. The refrigeration system as defined in claim 5 wherein said control means includes means for controlling the vapor output from said vapor generator.

9. The refrigeration system as defined in claim 1 including a second coil connected in series with said first coil and means for reversing the flow of refrigerant from said compressor whereby said system can operate as a reverse cycle heating or cooling system.

10. A refrigeration system including: a rotary compressor; means for conducting compressed refrigerant

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from said compressor through an expansion valve and
a heat exchange coil adjusted to function as an evapo-
rator and back to said compressor, a fan for directing
air across said coil; a vapor turbine; an alternator directly
driven by said turbine; a first electric motor drivingly
connected to said compressor; a second electric motor
drivingly connected to said fan; conductors for directly
connecting the electrical output of said alternator with
said electric motors; and control means for controlling the
speed of said turbine in accordance with the output level
required from said system.

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