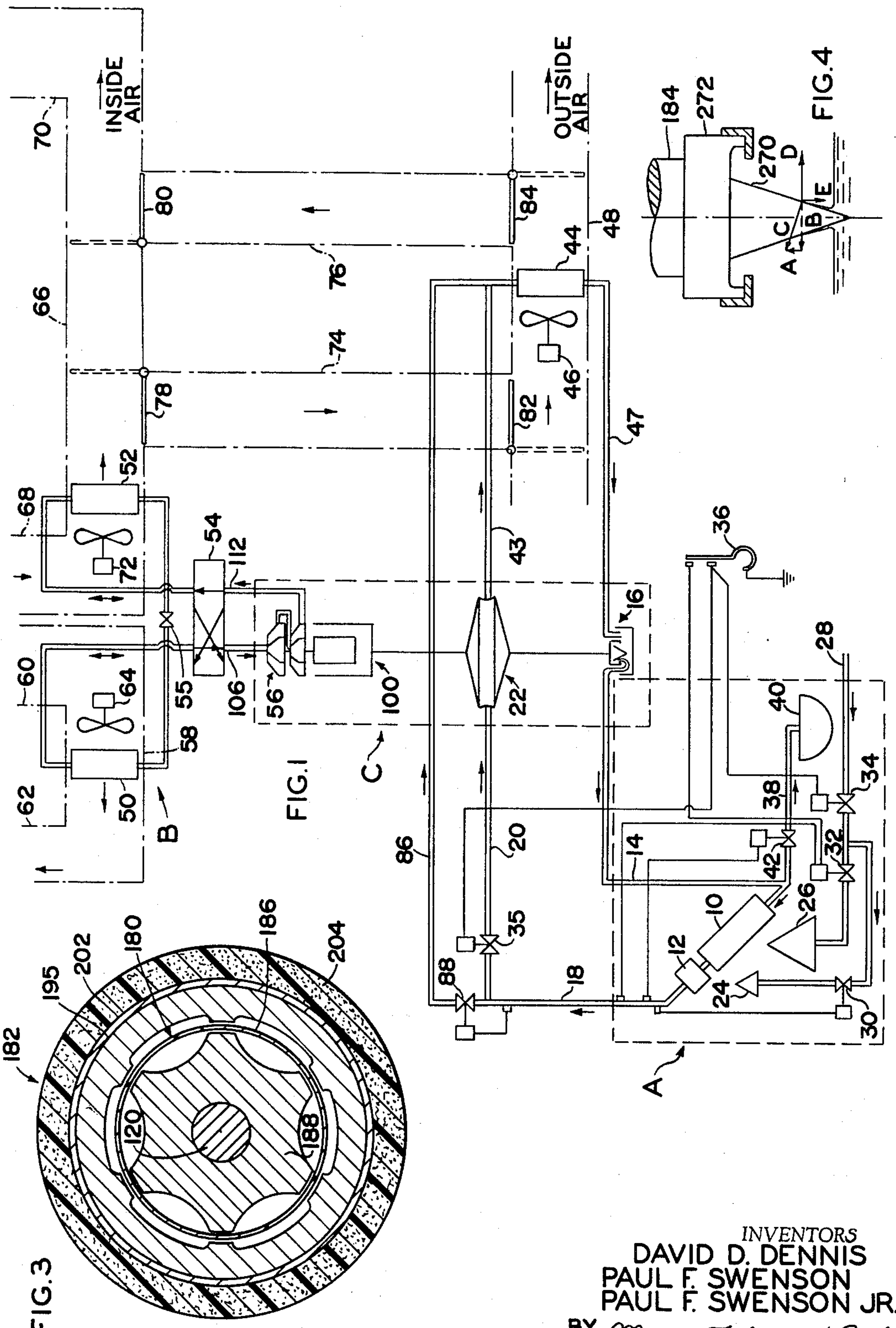
Sept. 10, 1968 D. D. DENNIS ETAL 3,400,554

FUEL-FIRED HEAT PUMP SYSTEM

Filed March 17, 1967 2 Sheets-Sheet 1



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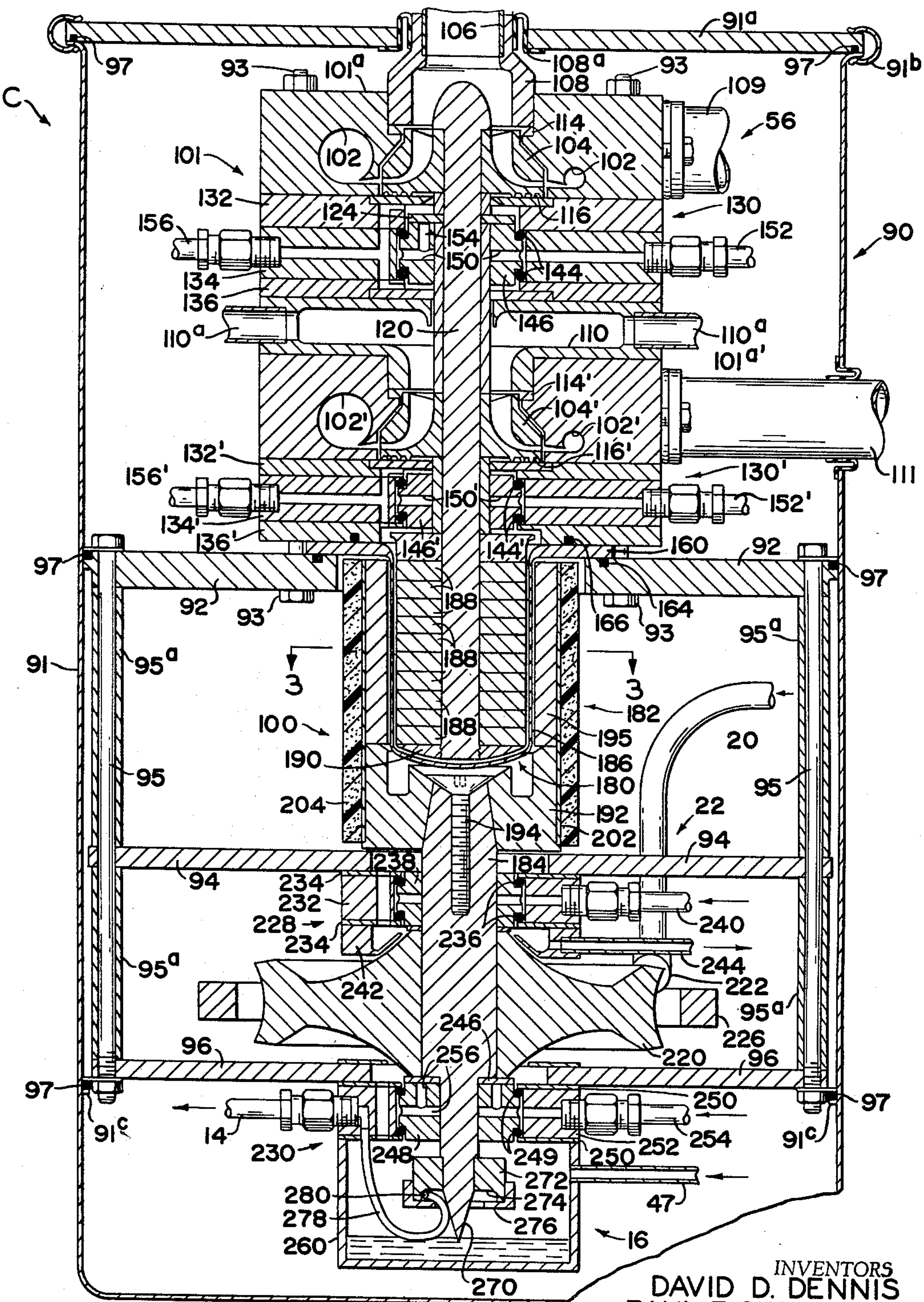

FIG. 2

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United States Patent Office

3,400,554

Patented Sept. 10, 1968

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3,400,554

FUEL-FIRED HEAT PUMP SYSTEM

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Filed Mar. 17, 1967, Ser. No. 624,063
5 Claims. (Cl. 62—238)

ABSTRACT OF THE DISCLOSURE

A closed-loop, reversible hermetically sealed refrigeration cycle the compressor of which is driven by a vapor turbine supplied with vapor from a once-through type vapor generator of a hermetically sealed closed-loop vapor cycle. The turbine and compressor are constructed as a unit and are drivingly interconnected through a magnetic coupling which is comprised of a pair of spaced but interfitted magnet members which are carried on the ends of the turbine and compressor shafts respectively. A fluid impervious non-magnetic barrier or partition member extends between the magnet members to seal between the turbine and compressor. A feed pump in the form of an inverted cone is carried on the lower end of the turbine shaft and extends down into a hot well or condensate receiving container and functions to pump the condensate back to the vapor generator. The system also includes an air duct arrangement which, when the system is operating on the heating mode, causes the air to be heated by both the Freon cycle and the vapor cycle apparatus. Additionally a vapor by-pass line is provided around the turbine to increase the heat available from the vapor cycle when needed.

The present invention is directed to the heating and cooling art, and, more particularly, to an improved fuel-fired reverse cycle heat pump system.

The invention is particularly applicable for constructing heating and cooling units of a size for use in residential dwellings and will be described with particular reference thereto; however, it is appreciated the invention is capable of broader application and could be used for constructing units for a variety of environmental control applications.

In general, most heat pump systems utilize a prime mover in the form of an electric motor for driving the compressor of a reversible fluorocarbon refrigeration cycle. The reason for this is, of course, the convenience of electricity as the energy source for the prime mover and the highly desirable characteristics of fluorocarbons as the working medium in refrigeration cycles. However, electricity is expensive as an energy source, especially for the residential user. Based on an energy cost per unit refrigeration comparison, the efficient electric motor is often not as economical as prime movers using fuels directly. In addition, the rotative speeds obtainable from commercially available electric motors using normal line power necessitate a reciprocating type compressor for refrigeration systems of capacity applicable to single residences and somewhat larger. Wear, noise and vibration are usually attendant to the operation of reciprocating machinery.

The use of a fuel such as gas, fuel oil or coal instead of electricity as the main power source has been considered desirable from the standpoint of lower energy costs, as well as potentially lower operating costs. Generally, previous attempts to design systems using these fuels as the power source have not been satisfactory.

In general, these fuel burning systems comprise a vapor turbine driven fluorocarbon compressor. The fuel is

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burned in a vapor generator to heat and vaporize a working fluid necessary for driving the turbine. Attempts have been made to use both fluorocarbons and water as the working fluid. However, problems encountered with both, have prevented the design of an economically sound system. Attempts to use fluorocarbons as the working fluid have been unsatisfactory since fluorocarbons generally have a low decomposition temperature and require that the vapor supplied to the turbine be of relatively low temperature. Consequently, the efficiency of the vapor cycle portion of the system is relatively low. Likewise, attempts to use water as the working fluid have been unsatisfactory. Although ideally, water vapor can yield a much higher cycle efficiency when used for driving the turbine, the incompatibility of water and fluorocarbons, and the difficulty of sealing between the compressor and the turbine has prevented the use of water as the working fluid. Those systems which have attempted to use water have required the use of complicated distillation or liquid separation apparatus such as shown in U.S. patent to Klaben et al. 3,250,082.

Various other problems are also encountered in attempting to design small size vapor turbine driven heat pump systems. For example, as is well known, vapor turbines operate most efficiently under vacuum conditions. After the vapor passes through the turbine and is condensed, a pump functions to return the working fluid to the vapor generator. In small units, the static head available is necessarily limited and the pump must pump from a relatively high vacuum. Small capacity pumps which will satisfactorily pump from high vacuum conditions are generally not available. Consequently, in previous vapor turbine driven heat pump systems it has not been economically feasible to operate the turbines under high vacuum conditions.

An additional problem present in reverse cycle heat pump systems generally, is their inefficiency when operating in their heating mode under high heating requirements.

The present invention provides a fuel-fire reverse cycle heat pump system which overcomes the above problems and permits such systems to be constructed in small sizes suitable for residential use.

In accordance with one aspect of the present invention a vapor turbine driven reverse cycle heat pump system is provided which includes: a closed vapor cycle comprising a vapor generator supplying vapor to a vapor turbine, a first heat exchanger for receiving and condensing said vapor after it has passed through said turbine, and means for returning the condensed vapor to the vapor generator; a closed circuit reversible refrigerant cycle including a compressor drivingly connected to said turbine and a second heat exchanger and means to selectively cause said second heat exchanger to function as the condenser or evaporator for said refrigerant cycle; air duct means interconnecting said first and second heat exchangers and means for causing air to flow through said duct means first over said second heat exchanger and thence over said first heat exchanger when said second heat exchanger is functioning as a condenser for said refrigerant cycle.

The above arrangement serves to greatly increase the overall efficiency of a vapor turbine driven reverse cycle heat pump system because when the system is being used for heating, the available heat in the vapor cycle condenser is used to increase the quantity of usable heat output of the system.

In accordance with a more limited aspect of the present invention, a pump particularly suited for use in a high vacuum environment is provided which comprises a container means for holding a fluid to be pumped, an elongated member having a longitudinal axis and extending downwardly into said container means, said member having a lower end adapted to be positioned to extend into the fluid in said container and having a circular cross-sectional configuration as defined by planes perpendicular to said longitudinal axis which gradually increases upwardly from said lower end, means defining a circumferentially extending fluid receiving recess upwardly spaced from said lower end, fluid pick-up means positioned within said recess for receiving fluid received therein, and means for rotating said member about its longitudinal axis.

In accordance with another aspect of the present invention an improved vapor turbine driven compressor unit is provided which includes: a housing and partition means dividing said housing into first and second chambers; a vapor turbine positioned in said first chamber and having an output shaft; a compressor positioned in said second chamber and having an input shaft; said turbine and said compressor being positioned so as to have their respective shafts in general alignment; coupling means for drivingly interconnecting said shafts in fluid sealed relationship, said coupling means comprising a first magnet member positively connected to said output shaft and a second magnet member positively connected to said input shaft, said first and second magnet members being in spaced mated relationship; and said partition means including a fluid impervious non-magnetic barrier member positioned between said first and second magnet members.

By magnetically interconnecting the turbine and compressor the problems previously encountered with regard to sealing are eliminated and the turbine and compressor can operate on incompatible fluids such as water and Freon.

A primary object of the present invention is the provision of a fuel-fire reverse cycle heat pump system.

An additional object of the present invention is the provision of a heat pump system which is highly efficient and overcomes problems previously encountered in heat pump systems.

A further object is the provision of an improved pump especially suited for use in a high vacuum environment.

A still further object is the provision of a vapor turbine driven compressor unit which overcomes problems previously encountered in sealing between the turbine and compressor.

Another object is the provision of a magnetic coupling which is especially suited for use as the driving connection between a compressor and a vapor turbine.

Yet another object is the provision of a turbine driven heat pump system which is especially suited for residential heating and cooling.

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

FIGURE 1 is a somewhat diagrammatic showing of a reversible heat pump system constructed in accordance with the present invention;

FIGURE 2 is a longitudinal cross-sectional view through the turbo-compressor unit of the system;

FIGURE 3 is a cross-sectional view taken on plane 3—3 of FIGURE 2; and

FIGURE 4 is an enlarged view of the feed water or condensate pump showing the forces acting on the fluid being pumped.

Referring now to the drawings wherein the showings are for the purpose of illustrating a preferred embodiment of the invention only, and not for the purpose of limiting same, FIGURE 1 shows the overall arrangement of the system comprising a fuel-fired steam generator unit A, and a reversible refrigeration cycle heat pump system B interconnected through a turbo-compressor unit C.

Steam generator A

The fuel-fired steam generator unit A is of the once-through type and includes an evaporator section **10** and

a serially connected superheater section **12**. Feedwater is supplied to the unit through a line **14** connected with feedwater or condensate pump **16** driven by the turbine of the turbo-compressor unit C. Steam at a pressure and temperature dictated by system requirements is conducted from the outlet of the superheater **12** by lines **18** and **20** to the turbine **22**. The steam temperature and pressure can be as high as necessary, within the limit of steam power plant practice, to achieve good cycle efficiency.

The steam-generator is fired by a pair of burners **24** and **26** supplied with fuel gas through line **28**. Burner **24** serves to heat superheater section **12** of the steam-generator, and is controlled by a conventional temperature responsive modulating valve **30** in accordance with the temperature of the steam leaving the superheater. Burner **26**, on the other hand, is utilized to heat the evaporator section **10**, and is controlled in accordance with the pressure of the steam leaving the steam-generator by a pressure responsive valve **32**. Main gas flow to the burners, and steam flow to the turbine, are controlled by conventional on-off solenoid valves **34** and **35** respectively, controlled by a two-step room thermostat **36**. The thermostat is of the type that can be shifted from direct to reverse acting, so as to be capable of controlling the system when it is functioning in either the heating or cooling mode.

A novel feature of the design of the steam-generator unit A is the means utilized to prevent damage to the water circuits in the event that burner operation should be disrupted, such as by lack of fuel, during cold weather. These means comprise a drain line **38** connected between the low point of the steam-generator circuitry and a reservoir tank **40**. Line **38** is controlled by a conventional temperature responsive valve **42** which is adjusted to open and drain the steam-generator in response to temperature in the range of freezing. As shown in FIGURE 1, the valve is connected to open in response to low temperatures sensed at the outlet of the steam-generator. However, it is apparent that other points in the steam-generator circuitry or points adjacent to the burners could equally well be utilized.

Although drain line **38** could, of course, be connected to a sewer, etc., by utilizing the reservoir tank **40**, the highly purified water utilized in the steam-generator is not lost and can be returned to the steam-generator when operation is restored.

As can be seen from FIGURE 1, the steam-water portion of the system is of the closed circuit type and comprises steam lines **18** and **20**, turbine **22**, turbine exhaust line **43**, condenser **44**, line **47**, feed pump **16**, and feedwater line **14** connected back to the steam-generator. The circuit is hermetically sealed so as to eliminate the necessity of supplying make-up water. This is important, especially in units designed for residential use, since the addition of a make-up water system would unduly complicate the unit.

Steam condenser **44** could be of many types; however, as shown, it comprises a conventional heat exchange coil over which air is conducted by a fan **46**. Fan **46** and condenser **44** are positioned in a duct **48**, the ends of which are connected to outside air.

Reversible refrigeration cycle system B

Although a variety of different heat pump systems could be used, system B is a conventional reversible refrigeration cycle comprising an outside heat exchanger **50** and an inside heat exchanger **52** connected with each other through a conventional expansion valve **55**, and through a two-position four-way mode valve **54** with compressor **56** of the turbo-compressor unit C.

Mode valve **54** contitutes means for permitting the heat exchangers **50**, **52** to selectively function as either the evaporator or condenser. In this manner inside heat exchanger **52** can perform as either a heating or cooling coil.

Heat exchanger **50** is positioned in a duct **58** which is communicated with the exterior of the building through an inlet duct **60** and an outlet duct **62**. An electric motor driven fan **64** is provided to assure the necessary air flow across the heat exchanger.

A similar duct system is provided for heat exchanger **52**, and includes a duct **66** communicated with the interior of the building through a return air duct **68**, and a supply air duct **70**. An electric motor driven fan **72** provides air circulation through the interior of the building and across heat exchanger **52**.

Although no controls are shown for fans **46**, **64** and **72** they would, of course, be provided with means to assure their operation during the periods when thermostat **36** is indicating a need for heating or cooling. The particular controls used for this purpose are not important to the present invention and could, for example, be merely switches operated by a solenoid or solenoids energized through thermostat **36**.

Heating mode operation

An important feature of the present invention is the arrangement provided for heating mode operation. During the periods when heating is required in the building, mode valve **54** will, of course, be shifted so that heat exchanger **52** is functioning as the condenser coil and the heat extracted therefrom is supplied to the building. Additionally, substantial heat is also available from condenser **44** of the steam-water cycle portion of the overall system. For this reason a pair of connecting duct means **74**, **76** are provided between ducts **48** and **66**. Flow of air through these ducts is controlled by means in the form of four dampers **78**, **80**, **82** and **84** positioned at the junctions of the ducts.

During cooling operation the dampers would be in the position shown by solid lines; however, when the system is shifted to heating operation the dampers are moved to the positions shown by dotted lines. With the dampers in these positions, the air coming through return air duct **68** is passed first through heat exchanger **42**, then via duct **74** to duct **48** and across condenser **44**. The air is then conducted through duct **76** back to duct **66**, and then to the building air supply duct **70**. In this manner, heat in condenser **44**, which would otherwise be lost to the outside atmosphere, is utilized to heat the building air and the overall efficiency of the system substantially improved.

The means used to move the dampers between their heating mode and cooling mode positions could be of a variety of forms, either manual or automatic. For example, the dampers could be provided with a mechanical interconnection with mode valve **54** so that when the valve is shifted from heating to cooling operation the dampers would be simultaneously shifted.

Because during extremely cold weather, the above described arrangement may not be adequate to supply sufficient heat at the necessary temperature ratio, means are provided to increase the amount of heat available at condenser **44**. These means comprise a line **86** which extends between steam line **18** and the inlet to condenser **44** and by-passes turbine **22**.

A valve **88** is provided to control the flow of steam through line **86**. The valve could be of a variety of types either manual or automatic, but is shown as of the pressure-opening type to respond to increased system operating pressure. The increase in system pressure is achieved through use of the two-step thermostat **36**, which modifies the set point of pressure control valve **32** when room temperature is a predetermined amount below the thermostat set point.

As can be seen, the above-described damper and steam by-pass arrangement provides the system with a high degree of flexibility and greatly increases its overall efficiency.

Turbo-compressor unit C

As shown in FIGURE 2, the turbo-compressor unit C comprises a compressor **56** which is drivingly connected through a specially designed magnetic coupling **100** with

vapor turbine **22**. A feed pump **16** is carried on the lower end of the turbine shaft.

All of the various components are mounted in a housing indicated in general by the numeral **90**. Housing **90** could take many forms and be constructed in a variety of ways; however, as shown it comprises cylindrical drum **91** having its upper end sealed by plate **91*a*** which is clamped to the drum by a circumferentially extending clamp ring **91*b***. A partition forming member **92** divides the interior of the housing into first and second sealed chambers. A pair of transversely extending supports **94** and **96** are rigidly supported in spaced relation in the lower of the chambers by the rods **95** and sleeves **95*a***. A plurality of large O-rings **97** are provided to mechanically isolate and seal the various support and partition members from the drum and to transfer the weight of the components to the drum at ring **91*c***.

Compressor 56

The exact construction of compressor **56** is not important to the present invention. However, according to the preferred embodiment it comprises a relatively conventional two stage supersonic type compressor. As shown it comprises a main housing assembly **101** which carries a vertically extending shaft **120** rotatably mounted on a pair of bearing assemblies **130** and **130'** and carrying the impellers **104** and **104'**. The compressor assembly is rigidly tied to the partition plate **92** by a plurality of bolts **93**.

Because both stages of the compressor are substantially identical only the first stage will be described in detail and the corresponding parts of the second stage will be identified by the same reference numerals with the addition of a prime (') suffix. A description of one part is to be considered equally applicable to the corresponding part unless otherwise noted.

The first stage of the compressor comprises a housing section **101*a*** provided with scroll shaped diffuser-collector section **102** and impeller **104**. The fluid to be compressed is supplied to the impeller **104** from line **106** through inlet **108**. Inlet **108** is sealed where it passes through plate **91*a*** by a flexible seal **108*a***.

The compressed fluid exiting from the diffuser-collector section **102** is conducted from outlet **109** to the inlet section **110** of the second stage by lines **110*a***. The connection between outlet **109** and inlet section **110** is not shown but comprises a simple T or Y fitting and the necessary connecting lines. The outlet **111** from the second stage passes in sealed relationship through the container **91** into connection with line **112** (see FIGURE 1).

As previously mentioned, bearing assemblies **130** and **130'** support shaft **120** for free rotation in housing assembly **101**. In general these assemblies are substantially identical with the exception that assembly **130** serves as the thrust bearing and supplies the vertical support for the shaft **120**.

As shown, bearing assembly **130** includes a plurality of annular shaped housing members **132**, **134** and **136**. These members are joined together with sections **101*a*** and **101*a'*** and the corresponding lower housing sections **132'**, **134'** and **136'**, by vertically extending bolts not shown. Positioned within the housing thus formed, and carried by circumferentially extending O-rings **144** is a carbon bearing forming member **146**. Bearing forming member **146** is of annular configuration, and is provided with a plurality of radially extending passages **150** for supplying lubricant from lines **152** to the bearing surface between shaft **120** and the bearing forming members. Member **146** also is provided with a vertically extending passage **154** adapted to supply lubricant to the bearing surface between it and thrust washer **124**. A drain line **156** serves to carry away the lubricant.

The lower surface of member **136'** of bearing assembly **130'** is connected to the radially extending flange **160** of the barrier member **186** of magnetic coupling **100** which is in turn connected to, and supported by, internal support member 92. The joints between these connections are sealed by O-rings 164 and 166.

Magnetic coupling 100

As previously discussed, past attempts to design a vapor turbine driven refrigerant compressor unit of small size have been largely unsuccessful because of the incompatibility of the most desirable fluids for use in the refrigerant cycle and the vapor turbine cycle, and the inability to adequately seal between the two fluids.

The present device overcomes this problem by providing a magnetic coupling between the turbine and compressor in a manner which eliminates the need for mechanical seals. Because of the necessary high rotative speeds of the turbine (in the range of 50,000 r.p.m's and higher), and the consequent high centrifugal forces imposed on the magnetic coupling, a coupling of special design is utilized.

As shown in FIGURE 2, this coupling comprises a cylindrically shaped inner magnet assembly 180 carried on the lower end of shaft 120, and in outer magnet assembly 182, of cup shaped cylindrical configuration, carried on the upper end of the turbine shaft 184. A cup-shaped barrier member 186, formed from an impervious non-magnetic dielectric material, such as "Pyrex" glass, is positioned between the inner and outer magnets and connected in sealing relation at its flanged end 160 to support 92.

Inner magnet assembly 180 comprises a plurality of individual magnet discs 188 formed from Alnico and snugly fitted to the lower end of shaft 120 and positively connected thereto by a nut 190.

The outer magnet assembly 182 includes a hub member cast integrally with, or shrink-fitted to the turbine shaft. Positioned immediately above hub member 192 is outer magnet member 195. As best shown in FIGURE 3, the outer magnet member includes a plurality of vertically extending magnet poles formed in a body 198 of Alnico.

Outer sleeve 204 provides the main structural strength for the assembly and is formed from fiber glass and resin. The main portion of the fibers is circumferentially wound because of the large circumferential load generated during rotation. The sleeve 204 is shrink fitted to the outer magnet member, preferably by hydraulically expanding it by positioning the sleeve vertically between a pair of end plates which close the end of the sleeve and are held in position by a clamp or tie rods. This permits the sleeve to sealingly retain hydraulic fluid. The top end plate is provided with a recess which receives the magnet member and permits it to rest on the top edge of the sleeve while the sleeve is in the unexpanded condition. Consequent pressurization of the hydraulic fluid by a pump communicated with the fluid in the sleeve through a line in one of the end plates, causes the sleeve to be expanded and the magnet member to drop into position in the sleeve.

When constructed in the above-described manner, the magnetic coupling is capable of transmitting the necessary torque from turbine 22 to compressor 56 at rotational speeds substantially above 50,000 r.p.m.

Vapor turbine 22

Vapor turbine 22 is similar to the conventional Terry type turbine, and includes a rotor 220 shrink fitted or otherwise positively connected to shaft 184. The outer periphery of the rotor is provided with a plurality of fins or blades (not shown) which are supplied with steam from line 20 by nozzle 222. A shroud 226 is positioned around the rotor.

The rotor is mounted for rotation by an upper bearing assembly 228 and a lower bearing assembly 230. These bearing assemblies are constructed in basically the same manner as previously described bearing assembly 130.

In particular, upper bearing assembly 228, includes an annular shaped bearing housing member 232 and a pair of inwardly extending bearing support members 234. A pair of O-rings 236 serve to support the carbon bearing 238 on support members 234. Radially extending lubrication passages are provided in bearing 238 and function to provide lubricant from line 240 to the bearing surface between shaft 184 and bearing 238.

Positioned immediately below the bearing assembly 228 and connected thereto is a circumferentially extending bearing lubricant receiving member 242 provided with drain line 244. Normally steam or feedwater will serve as the bearing lubricant for the turbine bearings. In the case of feedwater, the drain line 244 is connected with line 47, and lubricating fluid returned to the system.

Lower bearing assembly 230 serves to provide the main support for the turbine and for this reason includes a thrust washer 246 carried on a carbon bearing member 248 which is mounted between a pair of O-rings 249 carried by a pair of radially extending support rings 250 positioned on opposite sides of bearing housing forming member 252. The bearing housing member 252 is connected to the underside of support member 96 in any manner which will properly support the weight of the turbine and seal the turbine chamber.

Bearing lubricant in the form of feedwater or steam is supplied to the bearing surfaces of bearing 248 from line 254 and passages 256. No separate bearing lubricant drain line is provided for bearing assembly 230, since the bearing is in direct communication with hot-well 260 which is connected in sealed relation with the bearing support.

Feed pump 16

Of particular importance to the present invention is the construction of feed pump 16. As previously mentioned because of the high vacuum conditions under which the turbine operates, and the small static fluid head available in a small system like the present, conventional small centrifugal and piston pumps do not function properly. In general, when attempting to operate conventional pumps of the smaller sizes under the conditions present in the subject system, priming difficulties, as well as vapor binding, are experienced. For these reasons, the present invention includes a specially designed feed pump.

In general, the pump could be referred to as an inverted cone, surface-action pump. In particular the pump includes a generally conically shaped member 270 formed on the lower end of turbine shaft 184 which extends into a fluid container means in the form of a hot-well 260. Although, shown as a cone, the exact shape of the member could vary slightly and could for example, be of semi-elliptical cross-sectional configuration. Positioned immediately above the end portion 270 and firmly connected to shaft 184, such as by welding or shrink fitting, is a disc shaped fluid receiving member 272 having a circular recess 274 in its lower side. A member 276 is connected to the lower side of member 272 and in combination with recess 274 provides a circumferentially extending inwardly opening fluid receiving channel means. A fluid receiving means shown as a stationary pick-up tube 278 extends into the channel and has an open end 280 facing opposite the direction of rotation of shaft 184. The other end of tube 278 is connected with feedwater supply line 14 through a connection carried in bearing housing forming member 252.

Obviously the most efficient angle for the conically shaped member 270 will vary depending upon factors such as the speed of rotation, the characteristics of the fluid, etc. In the present application, with water near its boiling point, and with the pump rotating in the range of 40,000 to 70,000 r.p.m. an angle of from 20 to 40° is satisfactory.

The operation of the pump can best be explained by reference to FIGURE 4. All that is required to initiate pumping action is rotation of the cone when its end is submerged in the liquid. As shown, when the end of the cone is submerged in the liquid, the liquid rises a short distance up the cone in a meniscus-like fashion at 280.

As the cone is rotated, the forces acting on the liquid mass include a centrifugal force indicated by vector D, and a gravitational force indicated by vector E. These forces are counteracted by a force which results from the surface tension of the liquid and is indicated by vector C. Vector C, is, of course, comprised of an axial component indicated by vector A, and a radial component indicated by vector B.

The centrifugal force D increases with film thickness, whereas the resultant surface tension force C does not. Therefore, for a given tangential velocity on the cone's surface, there is an equilibrium film thickness, at which the radially inward component B of the surface force balances the centrifugal force D. The remainder of the surface force vector revolves axially, toward the large end of the cone, which for steady state flow will be balanced by the force of gravity.

As can be seen, the equilibrium film thickness reduces as the cone diameter increases. This is also a requirement for flow continuity for steady velocity up the cone. Any voids on the cone surface will then occur at the upper end of the cone, which will increase the lifting force on the fluid, drawing it into the void. Any vapor which forms as a result of heating caused by fluid slip on the cone surface will be ejected from the film since surface forces on a vapor are minimal compared to those on the liquid.

As the liquid moves up the cone it is accelerated tangentially because of drag between it and the cone's surface. Since there is slip between the cone surface and the liquid, the liquid will have a tangential velocity with respect to the cone surface. This tends to keep a uniform film thickness around the cone at any given axial level. As the liquid reaches the receiving channel **276**, the tangential velocity (or dynamic head) possessed by the liquid is recovered as pressure via a rapid diffusion process in the impact tube or collector tube **278**.

As is apparent, the advantages of the above-described pump are numerous and include its extreme simplicity and freedom from seals, as well as its immunity from cavitation problems even when functioning in a vacuum environment within one millimeter of mercury from the liquid's vapor pressure. Additionally, the pump is self-priming under the same vacuum conditions and is ideal for the low flow-high head operation required in the subject system.

As can be seen from the foregoing specification, a fuel-fired reverse cycle heat pump system has been provided which is highly simple in construction and operation, and overcomes the problems previously encountered in such systems.

The invention has been described in great detail sufficient to enable one of ordinary skill in the heating and cooling art to construct and use the same. Obviously modifications and alterations of the preferred embodiment will occur to others upon a reading and understanding of the

specification, and it is our intention to include all such modifications and alterations as part of our invention insofar as they come within the scope of the appended claims.

Having thus described our invention, we claim:

1. A vapor turbine driven heat pump system comprising: a closed vapor cycle including a vapor cycle generator means supplying vapor from said vapor generator to said turbine, a first heat exchanger for receiving and condensing said vapor after it has passed through said turbine, pump means for returning said condensed vapor to said vapor generator; a closed-circuit reversible refrigerant cycle including a compressor and means drivingly connecting said compressor to said turbine, a second heat exchange and means to selectively cause said second heat exchanger to function as the condenser or evaporator for said refrigerant cycle; air duct means interconnecting said first and second heat exchangers, and means for causing air to flow through said duct means first over said second heat exchanger and thence over said first heat exchanger when said second heat exchanger is functioning as a condenser.

2. The system as defined in claim **1** wherein a bypass line means is provided for bypassing said turbine to permit vapor to be conducted directly from said vapor generator to said first heat exchanger.

3. The system as defined in claim **2** including a valve in said bypass line means.

4. A system as defined in claim **1** wherein said pump means includes: a container means for receiving said condensed vapor, an elongated member having a longitudinal axis and extending downwardly into said container means, said member having a lower end adapted to extend into the condensed vapor in said container and having a circular cross-sectional configuration as defined by planes perpendicular to said longitudinal axis which gradually increases upwardly from said lower end, means defining a circumferentially extending fluid receiving recess upwardly spaced from said lower end, fluid pick-up means positioned within said recess for receiving liquid received therein.

5. A system as defined in claim **4** wherein said turbine has an output shaft rotatable about a longitudinal axis and said elongated member is carried by said output shaft in axially aligned relationship.

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