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12th June 2003

- 1 -

A MODULAR HEAT PUMP SYSTEM FOR DRYING AND AIR-CONDITIONING

FIELD OF INVENTION



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This invention relates to the design of a modular heat pump assembly for drying and
5 air-conditioning. The present invention, at least in one embodiment, relates to a heat pump
assembly design in which the refrigeration components can be easily coupled and de-coupled
to any air handling chamber for the purpose of drying or air-conditioning.

BACKGROUND OF THE INVENTION

Commercially available heat pump dryers are designed with both the refrigerating
10 components and drying chamber integrated as one complete drying unit. For such dryers, the
required refrigeration capacity of the heat pumps to cover the heat duties, both latent and
sensible, are sized according to the size of the drying chamber and the maximum product
loading. These dryers are inflexible for further scale-up of refrigeration equipment to deal
with increased product drying capacity, resulting in the partial or complete replacement of
15 the refrigerating components. As a result, money has to be invested by manufacturers to
scale-up their drying process to meet consumer demands. Furthermore, in the event of a
component breakdown, the drying operation for that heat pump dryer has to be terminated,
resulting in down time of the dryer and loss in production capacity. In the case of air-
conditioning servicing and maintenance, chillers have to be shut down for repair work,
20 resulting in changes of the air conditions in the rooms or climatic control chambers which
the conditioner is servicing. Room occupants may then experience drastic changes in air-
conditions and discomfort due to this. In the case of climatic control chambers for storage,
spoilage of the stored products may result.

The present invention advantageously provides a versatile heat pump assembly
25 whereby the refrigerating equipment is assembled exclusively from the air-handling chamber
before they are interfaced through the use of couplers, such as industrial air couplers. These
couplers allow air to be received for conditioning and the conditioned air discharged to the
chamber. The conditioning of the air takes into account air temperature, velocity and
absolute humidity.

- 2 -

Commonly available heat pump assemblies, such as for use as dryers, are single-staged. A single-staged heat pump has one evaporator. This evaporator has to undertake both the duties of sensible and latent heat absorption.

According to the invention, a two-stage heat pump has now been developed incorporating a two-stage evaporation system where one evaporator operates at a higher pressure with respect to the other. Advantageously, in a drying system, a two-stage system enables the recovery of more latent energy from the drying air and flexibility to implement several control strategies for better humidity control. For many drying applications, in order to reduce drying energy for product moisture removal and to improve the product quality, both energy recovery and air conditioning play an important role.

In improving heat recovery, the high-pressure evaporator advantageously assumes part of the cooling duty. It has been found that this allows the low-pressure evaporator to dedicate more of its thermal capacity for dehumidification. For better humidity control, the arrangement of evaporators in the air duct is advantageously such that it allows the humidity to be controlled by two methods. One is to use an air bypass process through the evaporators to control the dehumidification. The other method involves the regulation of the mean coil surface temperature of the high-pressure evaporator for sensible and latent heat extraction.

Another development of the present system is the incorporation of an external evaporator. The external evaporator plays a dual role. Its first role is to assist in the quick start-up of the heat pump. This is accomplished by 'absorbing' sensible heat from the surroundings during the on-cycle process until the condenser reaches the right pressure level. Its second role is to serve as a 'dummy' load to inject auxiliary load to the refrigeration system when low latent load in the chamber is encountered. The external evaporator advantageously ensures a consistent external load for heat recovery when the latent load in the drying chamber is low. This intervention is beneficial to drying processes where the initial latent load is high but diminishes progressively as drying proceeds into the falling rate region.

The present two-stage heat pump assembly advantageously also forms the modular building block for designing advanced multi-stage heat pump dryers to meet demands for higher product quality and energy efficiency as described above.

As such, the present invention advantageously provides better energy recovery by

- 3 -

improving the latent to total heat recovery ratio, and better control of the drying air through better sensible and latent heat exchange through the evaporators and condensers. Further, the invention advantageously provides a heat pump that is equipped with a quick start-up mechanism as well as a system that ensures consistent load for heat recovery for example
 5 throughout the drying process, and an energy efficient two-stage field heat pump that is easily integrated to any chamber.

That is, a versatile heat pump assembly is advantageously provided such that the heat pump components, assembled as a modular unit with portable or transportable features, can be coupled and de-coupled to any portable or transportable chamber for drying and general
 10 air-conditioning applications. The heat pump design is flexible to enable easy scale-up of dryers with varying chamber size and configuration to meet higher production demands. The invention also enables easy maintenance and repair work to be carried out on the heat pump components without production down time in drying processes, without product spoilage in storage chambers and without causing thermal discomfort to occupants in the air-
 15 conditioned chambers.

SUMMARY OF INVENTION

According to the present invention, there is provided a modular heat pump assembly adapted to be attached to a conditioning chamber, the heat pump assembly comprising:

- 20 a two-stage evaporator system comprising high and low pressure evaporators for conditioning of air;
- a hot gas condenser for introducing sensible heat to the air;
- a compressor;
- an external evaporator for facilitating constant latent load maintenance;
- at least one inlet port for receiving air from the conditioning chamber for
 25 conditioning by the heat pump assembly; and
- an outlet port for discharging conditioned air from the heat pump assembly to the conditioning chamber.

There is also provided a heat pump assembly comprising:

- 30 a heat pump unit comprising (i) a two-stage evaporator system comprising high and low pressure evaporators for conditioning of air; (ii) a hot-gas condenser for introducing sensible heat to the air; (iii) a compressor; and (iv) an external evaporator for

- 4 -

receiving the air from the hot-gas condenser and for facilitating constant latent load maintenance; and

a drying chamber adapted to receive conditioned air from the heat pump unit and recirculate air to the heat pump unit.

5 The invention is advantageously appropriate for drying and air conditioning applications. The drying air entering the heat pump unit is first conditioned by the evaporators followed by the hot-gas condenser before returning to the conditioning chamber through another port. The air may be cooled or heated depending on the desired chamber air conditions.

10 The configuration of the heat pump is preferably such that the high pressure evaporator operates as a partial dry/wet coil, and the low pressure evaporator operates as a dedicated wet coil. Furthermore, in a preferred embodiment a back pressure regulating valve is installed at a discharge line of the high pressure evaporator to ensure constant evaporating pressure.

15 In a preferred embodiment, the hot-gas condenser includes a sub-cooler arrangement and preferably comprises a hot-gas condenser and two sub-coolers. Such an arrangement advantageously provides the required sensible heat for the drying air to attain the desired temperature. Preferably, the assembly further comprises control means for ensuring sufficient sensible heat is provided from the hot gas condenser and sub-cooler arrangement
20 to the drying air to achieve a predetermined air temperature. Thus, the air temperature of the drying air may be regulated as desired.

In order to regulate the temperature of the air, the assembly preferably comprises a condenser, such as an air-cooled condenser, which facilitates the discharge of excess heat from the drying air in the system. Furthermore, in a preferred embodiment, the assembly
25 comprises a bypass damper system for regulating the flow of bypass air through the assembly. Also, an additional control means may be provided for modulating the bypass damper system to maintain a desired humidity of the drying air.

In order to cater to higher temperature requirements of the drying air, the assembly may further comprise an auxiliary heater for facilitating the introduction of additional
30 sensible heat to the drying air. Preferably, a further control means is provided for activating the auxiliary heater as required.

The heat pump assembly advantageously achieves maximum heat recovery for

- 5 -

efficient and economic drying of, for example industrial and agricultural products. This is achieved using the low and high pressure internal evaporators and the external evaporator for cooling and dehumidification of the drying medium. The exit air from the drying chamber is first passed through the high pressure evaporator to undergo pre-cooling process
5 before the low pressure evaporator dehumidifies the air for latent heat recovery. The arrangement of the evaporators allows different percentages of air to bypass and be mixed downstream of the coils. This enables the humidity of the drying air to be controlled through a simultaneous process of dehumidification and mixing as discussed above.

On the condenser side, two subcoolers are preferably used for heat recovery. The
10 subcoolers ensure that for high temperature operations more heat from the evaporators is recovered by making available more heat transfer area between the air and the refrigerant.

The system further incorporates a passive evaporator-economiser for pre-cooling of the air before the evaporators and pre-heating before the condensers. The facility can also operate as a chiller unit thus enabling both cold and hot air drying to be performed.

15 BRIEF DESCRIPTION OF THE DRAWINGS

A more detailed description of the invention will now be provided with reference to the drawings which illustrate embodiments of the invention and serve to explain the principles of the invention. It is to be understood, however, that the schematic drawings are designed for the purpose of illustration and explanation of the present design, and not as a
20 definition of the limits of the invention for which reference should be made to the claims appearing at the end of the description.

FIG. 1 illustrates a 2-stage heat pump dryer unit. As shown, the two evaporators, high and low, are arranged in series. An economiser is arranged around these two evaporators for pre-cooling and pre-heating of the drying air;

25 FIG. 2 is a schematic diagram showing the cycle of the heat pump system;

FIG. 3 is a schematic diagram showing a modular heat pump assembly with refrigeration components assembled in their respective localities;

FIG. 4 is a schematic diagram illustrating the application of the heat pump assembly for air conditioning purpose;

30 FIG. 5 is a side view of the assembled heat pump dryer showing a conditioning chamber and the heat pump components positioned in a modular unit;

- 6 -

FIG. 6 is a schematic diagram showing the high and low pressure evaporator system;

FIG. 7 is a schematic showing a control mechanism for the external evaporator to enable quick system start-up and consistent latent load absorption;

FIG. 8 is a circuit diagram showing the connections for the condenser and subcoolers. The diagram shows the refrigerant flow from a 3-way modulating valve to the vapour chamber and air-cooled condenser; and

FIG. 9 is a schematic diagram showing a control mechanism for shutting down the heat pump and maintaining the air temperature with an auxiliary heater bank.

DESCRIPTION OF THE EMBODIMENTS OF THE INVENTION

The present design is conceived from experiments conducted with a laboratory prototype two-stage heat pump dryer. The new design considers better control strategies and more efficient energy recovery. The design has been scaled up to that of a field dryer with the features for easy interface between the chamber and the modular heat pump unit.

In the following sections, the details of the chamber configuration, the refrigeration components of the heat pump cycle and the control strategy, for a parallel evaporator configuration, to regulate the humidity of the air will be presented.

Figure 1 shows a two-stage heat pump dryer unit in connection with a drying chamber. It can be observed that the dryer 10 has been designed such that the refrigerating components of the heat pump cycle 11 are partitioned from the drying chamber 12. This design advantageously allows further scale up of the drying chamber without changing the position of the coils and other refrigerating components. Further, such a design permits the construction of a field dryer where the refrigerating equipment can be readily integrated with any containerised chamber. This mode of construction allows flexibility for chamber scale-up as well as system maintenance. In this illustration, circulating air (shown by arrows) passes through internal dampers 13 into the heat pump 11. There is also provided a by-pass damper 14 which is usually used at the end of the drying cycle to remove moisture remaining in the air.

The air is fed from the chamber 12 to a pre-cooling economiser 15a and then on to the high pressure and low pressure evaporators, 16 and 17 respectively. The air is then passed through a second economiser 15b for pre-heating. Following this, the air travels through a heating arrangement including a hot gas condenser 18 and two sub-coolers 19a and

- 7 -

19b and optionally a heating booster 20. The air is then reintroduced to the chamber 12 for drying purposes.

External dampers 14a may also be provided to facilitate inlet of fresh air and outlet of used air from the assembly.

5 Although not shown in the Figure, there are advantageously provided condensate collection tray(s) and drain pipes for removing condensate from the chamber.

Figure 2 shows the refrigerating cycle of the heat pump. The heat pump cycle comprises two evaporators, one evaporator operating at a higher pressure with respect to the other. The advantages of such a two-stage system include recovery of more latent energy
10 from the drying air and the flexibility to implement several control strategies for better humidity control. Although not shown in the Figure, there are advantageously provided condensate collection tray(s) and drain pipes for removing condensate from the chamber.

Figure 2 shows two internal evaporators, the high-pressure evaporator 26 and low-pressure evaporator 25, and an external evaporator 28. Drying air from the chamber first
15 passes through the high-pressure evaporator 26 to be pre-cooled sensibly followed by the low-pressure evaporator 25 to be dehumidified. The dried air absorbs sensible heat from a hot gas condenser 35 to regulate to the desired temperature before it is discharged to an air-handling unit. The temperature control process is accomplished by regulating the required amount of flow rate, through a 3 way modulating valve 21, to the hot gas condenser 35 and
20 bypassing the rest to a mixing chamber 33. The refrigerant, exiting from the hot gas condenser 35, is mixed with the refrigerant from a bypass line 34 at the mixing chamber 33 before discharging to an air-cooled condenser 32. The refrigerant, is then collected at a liquid receiver 31 before it is allowed to expand to the three evaporators 25, 26 and 28 by three manual thermostatic expansion valves (TEV) or electronic expansion valves (EEV) 27,
25 29 and 30. Vapour refrigerant from the low-pressure evaporator 25 and external evaporator 28 are collected at a vapour chamber 23. Refrigerant from the high-pressure evaporator 26 is regulated to the pressure level of the low-pressure evaporator 25 by a back pressure regulating valve 24 before entering the vapour chamber 23. The vapour from the chamber 23 enters the suction line of the compressor and is compressed by the compressor 22 to its
30 discharge pressure level. When EEV are used, the surface temperature of both the high and low pressure coils can be set by entering the desired temperature values on an EEV control panel. This enables flexibility in controlling the pressure of both evaporators, since the

- 8 -

saturation refrigerant temperature corresponds directly to the refrigerant pressure during the two-phase evaporation process. When TEV are used, only the back pressure regulating valve is used to regulate the high pressure evaporator temperature. In both cases, by regulating the evaporator coil(s) temperature, either high or low pressure (when EEV are used) or high pressure (when TEV are used), the absolute humidity of the air can be regulated.

Figure 3 shows the modular heat pump assembly with one discharge port 58 and two air receiving ports 55 and 57. The high-pressure evaporator 45 is located above the low-pressure evaporator 52 in parallel configuration. Beside the coil surface temperature method mentioned in the previous section to regulate the air humidity, the air humidity can also be regulated by a set of interconnecting adjustable motorised-dampers 54 and 56. The modulating process is carried out by sending a control signal from the PID controller 47, installed on the control panel 46, to the dampers 54 and 56. Controlling the dampers regulates the amount of air flow through the high-pressure and low-pressure evaporators. Thus, the dehumidification process of the drying air can be carried out till the desired air humidity is reached. The heat pump design is also made versatile by allowing the compressor 40a to operate based on electrical power supplied by commercially available diesel or petrol generator 40. The liquid condensate from the high-pressure evaporator 45 is guided to a drain-off pan 53 via a drip pan 48 to prevent the condensate from entering low-pressure evaporator 52. The collected condensate from both evaporators 45 and 52 is discharged to the environment through a drain-off pipe 50. The external evaporator 49 is located outside the chassis as shown. Air from the surrounding is sucked through an air vane 51 to be cooled through the external evaporator 49. The absorbing of sensible heat from the surrounding improves the heat recovery for the heat pump system. It has to be pointed out that the design of the modular heat pump unit is not based on architectural preference. The present heat pump design, while adopting the two-stage cycle for air-conditioning, is unique in its arrangement and configuration for technical and spacial reasons. Firstly, it enables the humidity of the air to be regulated using two or more dampers with evaporators arranged in parallel configuration. By parallel configuration, we refer to a path of air flow through each evaporator before the air is mixed downstream from the evaporators. This parallel arrangement has two advantages. Firstly, it ensures even air distribution to food trays, thus preventing any air stagnation regions in the chamber.

- 9 -

Secondly, it enables the heat pump components to be compactly arranged, a feature that is essential for the modular application of the heat pump design.

A circulating fan 44, as shown in Figure 3, which may operate in axial or centrifugal mode, provides sufficient circulating power to push the drying air from suction ports 55 and 57 to the discharge port 58. An air inlet louver 42 is used to introduce some fresh air to the drying air cycle. An air-cooled condenser 41 is installed to reject excess heat to the environment. The control panel 46, housing the PID controllers, start and stop switches, is located at the side of the chassis for controlling and monitoring purposes.

The entire chassis may be coupled to a conditioning chamber via receiving ports 55 and 57 and the discharge port 58. Flexible industrial air couplers are used to join the ports of the assembly to the conditioning chamber.

Other working configurations for the evaporators may also be considered. For example, the modular heat pump unit can also be designed with the high and low pressure evaporators arranged in series for the same purpose of latent and sensible cooling. Another possible design is that of a single evaporator constructed such that both high and low pressure evaporators are integrated but operate at variable conditions.

Figure 4 illustrates an application of the modular heat pump unit for air-conditioning in a cold storage room for thermal-perishable products 64 that are susceptible to spoilage when thermal condition changes for a period of time. Figure 4 shows two ducts 61 and 66 interfacing with the heat pump system for cold air-conditioning. The cold air is discharged to the storage room via air nozzles 65. The remaining air is returned to the heat pump system for re-conditioning. Industrial air couplers 62 and 63, one for discharge of air from the heat pump and one for suction of air to the heat pump, are used to interface the air-handling unit and the heat pump system. This design advantageously provides temporary cold storage for products while the heat pump equipment undergoes servicing.

Figure 5 shows a commercially available transport chamber modified to be attached to the assembly of the invention enabling a drying assembly to be constructed. The use of a readily available transport container for loading and unloading of product and for providing a chamber for directing the drying air from the heat pump unit to the product significantly reduces capital cost. The drying chamber 71 is partitioned into two segments by three partitions 72, 72a and 72b. These partitions, 72, 72a, and 72b, create a U-channel duct for the drying air to flow from the ports 74 and 74a to the loading trolleys 78a with product

- 10 -

trays 78b before the drying air is returned to heat pump via ports 75, 75a, 76 and 76a. A re-circulation fan 73 is installed to provide additional air flow to the products. The products are properly arranged on netted trays 78b before being placed on wheeled-trolleys 78a. The trolleys are then wheeled into the chamber through loading door 79 with the first loaded
 5 trays nearest to the heat pump system. The product, after being dried to the desired moisture content, leaves the drying chamber via the unloading section 77 after opening door 77a. The unloading section 77 with door 77a is configured for quick unloading of the products once they are dried to their desired moisture content. Therefore, the present dryer adopts a first-in first-out process.

10 The coupling ports to the heat pump chassis are located at 74a, 75a and 76a. An air discharge louver 70 is used to expel some of the circulating air from the chamber after fresh air has been introduced to the air cycle via the air damper 42 as shown in Figure 3. From Figure 5, it may be readily realised that chambers of different sizes can be coupled so long as the interface ports are connected. Thus, the design advantageously results in easy system
 15 scale-up. The design of the chamber is a novel concept as it enables fast loading and unloading of the drying product by adopting a first-in first-out process. Also, the partitioning enables the drying air to be uniformly distributed to the product trays, resulting in uniform drying of the products. Further, the chamber may be fitted with cart rails to receive the product trolleys. These rails may be static or may be part of a conveying
 20 system which will permit continuous flow or batch flow of products.

The two-stage heat pump drying system has been designed to improve the heat recovery process and thus the thermal efficiency of the dryer. The basic principle in the heat recovery process is to improve the latent to total energy transfer at the evaporators. The following sections illustrate how the 2-stage system is able to improve heat recovery
 25 process.

Internal evaporators (high and low pressure)

Figure 6 shows the connections for the two internal evaporators. The present system uses two direct expansion internal evaporators to achieve the desired humidity of the air by expansion of the liquid refrigerant from the condenser to two different pressure levels. Two
 30 thermostatic expansion valves 85 and 86 are used to expand the refrigerant leaving the condenser to two different pressure levels. The use of two evaporators 83 and 84 at

- 11 -

different pressure levels advantageously improves the latent to total cooling energy transfer at the evaporators and enables better control of the drying air humidity. The drying air leaving the drying chamber is cooled to its dew point or below after the high-pressure evaporator 83. In this way, the high-pressure evaporator 83 operates as a partial dry/wet
 5 coil while the low-pressure evaporator 84 operates as a dedicated wet coil with more dehumidification capability. The use of direct expansion evaporator coils facilitates modulation of the refrigeration capacity when the latent load in the chamber varies. Figure 6 further shows how the back pressure regulating valve 81 is installed at the discharge line of the high pressure evaporator. This regulator ensures a constant evaporating pressure and
 10 thus a constant surface temperature on the evaporator coil. The regulation process is achieved by throttling in the compressor suction line 81a. The amount of refrigerant through the coil is matched with the evaporator load.

External evaporator

Figure 7 shows a control mechanism for the external evaporator. The external
 15 evaporator 114 plays a dual role. Its first role is to assist in the quick start-up of the heat pump dryer. This is accomplished by pressure switch 116 sending an on-off signal 117 to the normally closed solenoid valve 112 when compressor discharge pressure 120 reaches a pre-set value. This allows the flow of refrigerant, shown by the flow meter 111, to the thermostatic expansion valve 113 and external evaporator 114, resulting in heat being
 20 absorbed from the environment by the external evaporator 114. Another on-off signal is sent to the solenoid valve 112 when the compressor discharge pressure 120 reaches another pre-set value, terminating the refrigerant flow to the thermostatic expansion valve 113 and thus cutting off the external evaporator 114 operation.

The second role of the external evaporator 114 is to serve as a 'dummy' load to inject
 25 auxiliary sensible load to the refrigeration system when low latent load in the chamber is encountered. The external evaporator 114 ensures a consistent external latent load for heat recovery when the latent load in the drying chamber is low. This intervention is beneficial to drying processes where the initial latent load is high but diminishes progressively as drying proceeds into the second falling rate region. At the second falling rate period, a
 30 feedback signal 122 obtained from the weight of the product, a measure of the latent load in the chamber, is sent to the PID controller 119. If the weight of the product reduces to below

- 12 -

a set value, the controller 119 sends an on-off signal 118 to the solenoid valve 112 and activates it. This action allows the refrigerant to flow to the thermostatic expansion valve 113 which in turn distributes the refrigerant to the external evaporator 114 based on the bulb temperature 115. This allows heat to be absorbed from the environment by the external evaporator. Under this condition, the amount of absorption of latent energy required to offset the latent losses in the chamber due to the product's falling rate can be determined and consistent latent absorption for the system can be maintained.

Condenser and subcoolers

Figure 8 shows the connections for the hot-gas condenser 124 and the two subcoolers 125 and 126. One hot gas condenser 124 plus two subcoolers 125 and 126 provide the required sensible heat for the air to attain a desired temperature. The two subcoolers 125 and 126 recover additional heat by making available more heat transfer area for the refrigerant to condense from vapour to liquid form. The use of two subcoolers 125 and 126 maximises the recovery process and improves overall energy efficiency.

Economiser

A refrigerant economiser is installed before and after the high and low pressure evaporator coils. The economiser undertakes part of the cooling duty for the evaporator and heating duty for the condenser. It pre-cools the air before it enters the evaporators and pre-heats the air before it enters the hot gas condenser. The use of the economiser to transfer heat from air on one side of the evaporator to the air on the other side can increase the water extraction capacity of the heat pump dryer. Pre-cooling at the air-on side of the evaporators enables the major part of the cooling coil capacity to be used to extract water from the air.

The desired drying air conditions are generated and controlled with the use of three PID controllers. One controller is used to provide sufficient sensible heat from the hot-gas condenser (HGC), and the two sub-coolers (SC1 and SC2) to the drying air to achieve the air temperature. The second controller is used to control sensible heat input from an auxiliary heater to cater to higher air temperature requirements. The third controller is used to modulate the face and bypass dampers of the evaporators so as to control the dehumidification process and maintain the desired humidity of the drying air.

Air temperature

Figure 8 shows a control mechanism to control the drying air temperature. The air temperature is controlled by sending a continuous PID signal to a 3-way modulating valve 123. This valve regulates the flow of refrigerant vapour to the HGC 124, SC1 125 and SC2 126, and bypassing the rest before the two lines 128a and 128b meet at point 128c. The mixed refrigerant then enters the air-cooled condenser 129. The air-cooled condenser 129 is used to reject the excess heat before the entering liquid receiver 130.

Figure 9 illustrates another control strategy to improve energy saving by using the auxiliary heaters 131 to provide the sensible heat for the drying air during the last stage of drying. Towards the last fraction of drying, approximately 10 to 20% of the drying process, the humidity of the air remains relatively the same. Only the drying air temperature drops because of the continuous provision of sensible heat to remove product moisture. Therefore, in an economical sense, switching the heat pump system off and providing continuous heating with the auxiliary heater bank 131 would reduce operating costs. To accomplish this, a feedback signal from the weighing machine 138 is fed to the PID controller 134, which in turn sends two simultaneous signals 135 and 135a to the compressor and the auxiliary heater bank, respectively. Signal 135a activates the on-off relay 132 and closes the contact for the auxiliary bank while signal 135 de-activates the normally close on-off relay 136a. Once relay 136a is de-activated, the potential free contact 137 is opened. This causes the motor starter relay 136 to cut out the signal from the 3-phase power supply to the scroll compressor 133. This chain of events activates the heaters in the auxiliary heater bank 131 and de-activates the scroll compressor 133, resulting in the shut down of the heat pump and controls the air temperature with the auxiliary heater bank 131.

Air humidity

The third PID controller is used to control the motorised damper. The damper regulates the flow of air through the face and bypass dampers of the evaporators. The dehumidified air through the evaporators is mixed with the bypassed air to maintain the desired humidity of the air.

The second method of humidity control relies on a back pressure regulating valve 81 as shown in Figure 6; to regulate the operating pressure of the HP evaporator. The operating pressure of the HP evaporator serves as an accurate indication of the evaporator-

coil temperature. On many air conditioning applications, the maintenance of a fixed coil temperature is all that is required for accurate control of both sensible and latent heat removal. Through precise regulation of the back pressure regulator 81, the mean coil surface temperature is set to extract the required sensible and latent heat to obtain required air humidity.

Air flow rate

The flow rate of the drying air to the chamber is controlled by a 3-phase frequency inverter. By regulating the frequency of the inverter and reading from the air vane anemometer, the flow rate of the air entering the chamber can be set to the desired volumetric flow.

Refrigerant Circuit Control

To ensure that the scroll compressor works within the recommended operating envelope, the condensing temperature is set to operate in the range of 45°C to 55°C (16.5 bar to 21 bar, respectively, measured at the compressor discharge after de-superheating). The lower limit is proposed for an ambient temperature of 35°C and allowing a temperature difference of 10°C for sufficient heat rejection to the surroundings by the air-cooled condenser. The upper limit is obtained with the purpose of safeguarding the compressor life by preventing overheating of the motor. At the evaporator side, in order to achieve a cooling capacity of 14.2 kW, an SST of 5°C is normally set. A lower evaporating temperature can be set but there is always a possibility of the coil freezing up. At the same time, the corresponding cooling capacity would be reduced. The cut-in and cut-out pressure for the external evaporator is set at 14 bar and 16 bar, respectively, to achieve the desired condensing temperature and maintain the evaporator temperature within the operating range.

Example

Initial commissioning runs have been conducted to evaluate the system stability and time to approach steady-state conditions. Table 1 shows results obtained from air-side measurements without any latent load in the chamber. This example is for illustrative purposes only and should not be construed as limiting on the invention in any way.

Table 1 - Results obtained from heat pump system and air-side measurements

Compressor and heat pump system operating parameters:	
Compressor running ampere	8.5 Amp
Blower running ampere	3.2 Amp at 50 Hz
Suction Pressure	4.0 to 5.0 bar
Discharge Pressure	19 to 21 bar
Chamber temperature	50°C
Relative humidity	25 %
Air flow speed	2.8m/s
Air-side temperature	
3-way temperature PID setting	48.5°C
After drying chamber	47.5°C
After pre-cool coil	44.0°C
After HP evaporator	22.5°C
After LP evaporator	13.5°C
After pre-heat	18.6°C
After SC2	32.0°C
After SC1	37.0°C
After HGC	48.3°C

Based on values in Table 1 and compressor energy consumption, the COP of the heat pump operating at 48.5°C is calculated to be 2.96. We expect a higher heat recovery and improved COP when latent load is available in the chamber.

NOTATION

COP	Coefficient of performance	dimensionless
EC	Economiser	-
HGC	Hot gas condenser	-
HP	High pressure evaporator	-
LP	Low pressure evaporator	-
PID	Proportional-Integral-Differential	-
SC1	Subcooler 1	-
SC2	Subcooler 2	-
SDT	Saturation discharge temperature	°C
SMER	Specific moisture extraction rate	kg/kWh
SST	Saturation suction temperature	°C