Purdue University
Purdue e-Pubs

International Refrigeration and Air Conditioning Conference

School of Mechanical Engineering

2010

Improvements of High-Temperature Drying Heat Pumps

Vasile Minea
Hydro-Québec Research Institute

Follow this and additional works at: http://docs.lib.purdue.edu/iracc

Minea, Vasile, "Improvements of High-Temperature Drying Heat Pumps" (2010). *International Refrigeration and Air Conditioning Conference.* Paper 1103.
http://docs.lib.purdue.edu/iracc/1103

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.
Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at https://engineering.purdue.edu/Herrick/Events/orderlit.html

Improvements of High-Temperature Drying Heat Pumps

Vasile MINEA*

Hydro-Québec Research Institute, Laboratoire des technologies de l'énergie (LTE), 600, avenue de la Montagne, Shawinigan, Canada

Telephone: 819-539-1400 (1507); Fax: 819-539-1409; E-mail: minea.vasile@lte.ireq.ca

* Corresponding Author

ABSTRACT

Inappropriate integration of drying heat pumps may lead to wrong operating parameters, problems such as high discharge pressures and temperatures, and even compressor mechanical damage. This paper presents a number of design and control improvements of high-temperature drying heat pumps aimed at avoiding such undesirable incidents and increasing overall drying energy efficiency. The scope is to provide safe operating conditions and, thus, accelerate the implementation of industrial high-temperature drying heat pumps.

1. INTRODUCTION

The basic components of high-temperature heat pumps include a compressor, at least two heat exchangers (evaporator and condenser) and an expansion device. Appropriate heat pump/dryer integration, design and control must provide normal and safe thermodynamic parameters, as well as good quality of dried materials. This paper is intended for drying scientists, engineers and research students working with industrial and/or laboratory-scale drying heat pumps.

2. SYSTEM INTEGRATION

System integration is a major issue with any kind of drying heat pumps. Operating and energy performance is highly dependent. Consequently, a number of key components and configurations are needed to provide normal thermodynamic conditions. The most significant deficiency of several published configurations relates to the air ducting across the heat pump evaporator and condenser. The same flow rate through the evaporator and condenser is often used. However this does not provide optimal conditions because a number of problems may occur. For example, if the air flow through the evaporator is too low, or there is poor air distribution through the coil, the unit may trip off when reaching its *low* suction pressure. Also, if the air flow rate through the condenser is too low, the unit may trip off on *high* pressure control. On the other hand, if the air temperature at the condenser inlet is too high, the compressor may shut down on reaching its *high* limit pressure.

2.1 System integration

An optimized drying heat pump system is shown in Fig. 1 (Minea, 2008). On both the refrigerant and drying air circuits, it points out a number of components that sometimes are missing in some published studies. On the heat pump refrigerant-side, the liquid valve LV is used to control refrigerant migration during the on/off cycles and standby periods. The internal sub-cooler/super-heater SC improves the overall efficiency of the thermodynamic cycle. The heat rejection heat exchanger HR is a liquid sub-cooler with a cycling fan, which controls the refrigerant content and/or temperature leaving the heat pump condenser. On the air-side, bypass and motorized dampers around the evaporator EV are required to vary the air flow through the coil. This arrangement helps control the dehumidification rate, and supplies a sufficient air flow rate to the condenser. This design also avoids excessive condensing pressures and temperatures, and helps attain the desired air supply temperature without over-designing

the condenser heat transfer surface. The correlation between the evaporator air flow rate ($\dot{m}_{EV}$) and the condenser air flow rate ($\dot{m}_{CD}$) is:

$$\dot{m}_{EV} = \varepsilon \dot{m}_{CD} \tag{1}$$

where the air mixing factor ε varies between 0 and 1.

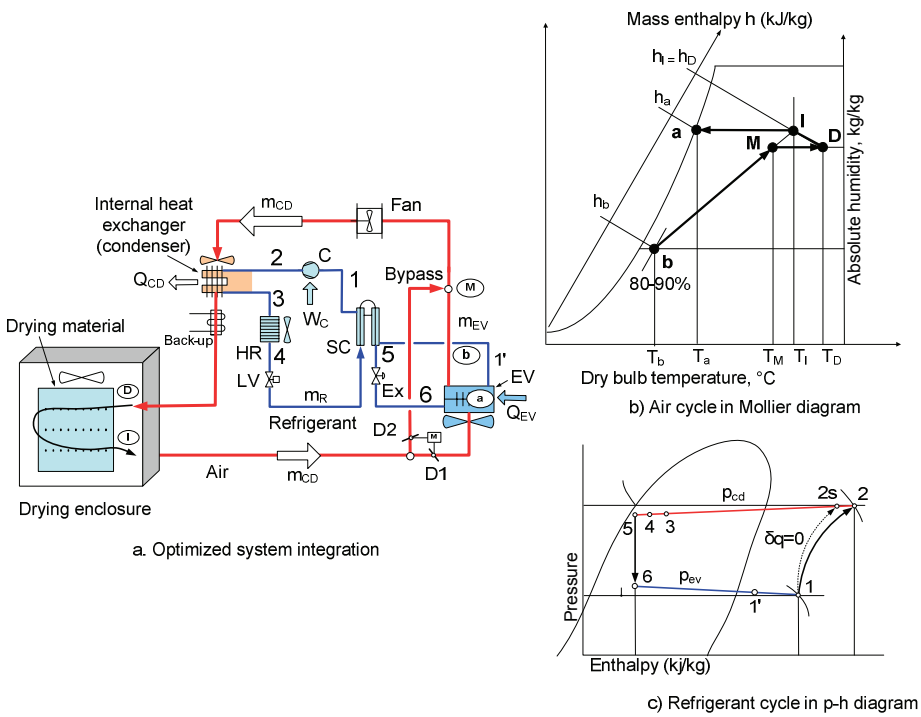

Figure 1: An optimized heat pump drier integrated system; m: air flow rate; q: heat density; p: pressure; s: mass entropy; T: temperature; Q: thermal flux; W: electrical power

2.2 Compact and split drying heat pumps

The optimized integrated system shown in Fig.1 may be adapted for both compact and split drying heat pumps. Compact systems contain both the evaporator and condenser inside the same cabinet located outside the drying enclosure (Figure 2a). The air flow rates through the evaporator EV (state I) and the condenser CD (mixing air – state M) are provided by constant or variable speed heat pump blowers. In the split units, there are two chambers thermally isolated from one another: the drying enclosure with remote condensers and the mechanical (dewatering) room with compressor C, evaporator EV, blower, air ducts and controls (Figure 2b) (Lewis, 2003). In this case, all temperature-sensitive components are thermally separated from the hot drying area.

2.3 System operation

In the compact units, the warm and humid air from the drying enclosure at state I is divided into two flows (Fig. 2a). One of them is first cooled sensibly (process I-a) and then dehumidified (a-b) across the refrigeration coil EV. The rest of the air at state I bypasses the evaporator. The mixed air (state M) is heated at a constant absolute humidity rate by the condenser CD (process MD) up to temperature T_D, which is higher than the surrounding air dry-bulb temperature T_I (Fig. 1b). Inside the dryer enclosure, the air heated in the condenser passes through the material stacks and picks up moisture from them. The back-up heating coil supplements the heat that the drying atmosphere receives from the heat pump so as to further raise the air temperature before it is returned to the material stacks being dried. The isenthalpic dehumidification process DI removes humidity from the material. In the split units (Fig. 2b), the indoor air, which is cooled and dehumidified by the evaporator EV (process I-a-b), is mixed inside the drying enclosure, that acts as a large air plenum, with indoor air at state I. The mixing air at state M is heated by the

condenser CD and, when needed, the back-up heating coil. In both cases, the refrigerant absorbs heat from the air and undergoes a two-phase change from vapour-liquid mixture (6) to superheated vapour (1') inside the evaporator EV (Fig. 1c).

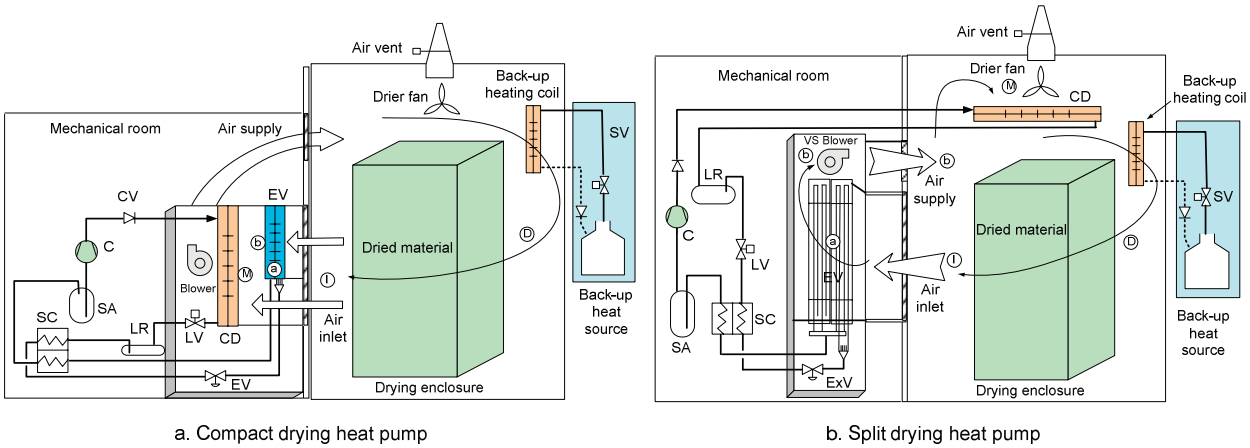

Figure 2: Schematics of compact (a) and split (b) drying heat pump systems; SA: suction accumulator; LR: liquid receiver; VS: variable speed; CV: check valve

Sensible and latent heat is absorbed for boiling and superheating the refrigerant. The refrigerant superheated vapour 1' enters the sub-cooler SC where it absorbs heat from the warm liquid refrigerant coming from the condenser or heat rejection coil HR at state 3 or state 4, respectively. The superheating process 1'-1 improves the overall performance of the cycle. The vapour at state 1 enters the compressor C where the electrical energy input is converted to mechanical work to adiabatically raise the vapour pressure. When increasing the vapour pressure, the condensing temperature is raised to a level higher than that of the dryer temperature. The work of the compressor and the total energy recovered inside the evaporator are transferred to condenser CD. Inside the condenser, the refrigerant vapour first undergoes a change from superheated (2) to saturated vapour state, and then partially condenses up to state 3. During this process, heat is rejected at a temperature higher than the surrounding air temperature. The heat rejection coil HR then acts as a liquid sub-cooler (process 3-4) in order to control the dehumidification capacity and to reject any excess heat outside. The internal heat exchanger SC further condenses and/or sub-cools the refrigerant up to state 5. The expansion valve EX expands the liquid (5-6) in order to reduce its pressure and temperature before entering the evaporator EV at two-phase state 6. At this point, the thermodynamic cycle ends and restarts again.

2.4 Dryer capacity – softwood case

The heat pumps may not accurately operate as dryers if the total quantity of (sensible and latent) heat extracted is not high enough to provide appropriate operating parameters. Moreover, the heat pump efficiency ratio and drying performance may not be accurately determined if the evaporator heat extraction capacity is too low, and/or the drying heat is mostly supplied by the compressor. Appropriate correlations between the initial mass and moisture content, the moisture migration/extraction rate of each dried product and the heat pump dehumidification capacity have to be provided for the drying processes. That means that a sufficient quantity of dried material has to be used inside the drying enclosure to provide the required thermal input to the evaporator. An insufficient quantity of dried material leads to a small quantity of moisture and heat extraction, evaporating pressures and temperatures that are too low, low compressor suction pressures and temperatures, and short drying cycles. In addition, these poor operating conditions may lead to a number of heat pump problems. If the heat pump capacity and performance are known in the case of *existing drying heat pumps*, the required initial quantity of dried material may be calculated. In this case, the dehumidification capacity $\dot{Q}_{EV}$ (kW) and/or the nominal compressor input power W_C (kW), as well as the heat pump coefficient of performance (COP), are known. In the case of *softwood* drying, the objective is to determine the required green wood volume to be dried V_G (m^3) in relation to the compressor input power (W_C), the

humid wood initial ($MC_{in,hum}$) and final ($MC_{fin,hum}$) moisture contents, and the wood basic density (D_b) (Minea, 2008):

$$V_G = \frac{105 * (COP - 1) * W_C * t}{(MC_{in,hum} - MC_{fin,hum}) * D_b} \tag{2}$$

where t is the dehumidification time (hours).

2.5 Heat pump capacity – softwood

If the dryer volume is known, the required heat pump capacity has to be calculated. In the case of existing softwood dryers, the dryer enclosure capacity and the green wood (saturated) volume (V_G , m^3) are known. The capacity of the drying heat pump, expressed in terms of compressor input power (W_C) and/or dehumidification capacity ($\dot{Q}_{EV}$), will be, respectively (Minea, 2008):

$$W_C = \frac{(MC_{in,hum} - MC_{fin,hum}) * D_b * V_g * h_{fg} * (1+\eta)}{100 * t * (COP - 1)} \tag{3}$$

$$\dot{Q}_{EV} = \frac{(MC_{i,h} - MC_{f,h}) * D_b * V_g * h_{fg} (1+\eta)}{100 * t} \tag{4}$$

where η represents the percentage (%) of sensible heat compared to the total latent heat removed by the heat pump, and h_{fg} – the water condensing mass enthalpy (kJ/kg).

3. CONTROL IMPROVEMENTS

Inappropriate control strategies allow drying heat pumps to operate with a low dehumidification efficiency and may reduce the compressor's technical life and/or lead to mechanical damage. The most significant dangers come from the liquid refrigerant entering the compressor suction line, from the vapour condensing during the adiabatic compression process and from frequent and short compressor on/off cycles. Inadequate correlations between the heat pump dehumidification capacity and the material dewatering capability may also compromise the overall heat pump performance. This section presents a number of improved control strategies for both compact and split drying heat pumps.

3.1 Refrigerant migration

A significant temperature differential exists between the drying enclosure and the mechanical room. During the heat pump stand-by periods or the preheating steps, the refrigerant migrates inside the refrigeration piping. The compressor C and the suction accumulator SA are large metallic masses with a high thermal inertia (Fig. 2). Their average temperatures are lower and the refrigerant may condensate inside. When the compressor starts, it will first pump the liquid refrigerant. Because the liquid is incompressible, the pressure differential is low and the compressor automatically shuts down. After a short time period, it starts up again. Several on/off cycles may occur until the whole liquid is pumped out. Too many such on/off cycles may reduce the technical life of reciprocating compressors and damage their moving parts (pistons, connecting rods) and cylinders. To avoid the liquid entering the compressor suction line, a standard liquid valve LV must be installed between the condenser and the liquid receiver (Fig. 2). The following sequence will provide safe operation of the compressor. When the compressor shuts down, the liquid valve immediately closes but the compressor continues to run for a few additional seconds. The refrigerant from the evaporator EV is pumped down and stored inside the liquid receiver LR. At the end of this operation, the evaporator is practically empty of any refrigerant and the liquid receiver stores the majority of the refrigerant charge. When the compressor is ready to restart, the liquid valve first opens for a few seconds, while the compressor is still off. During this period of time, a certain amount of liquid flows from the liquid receiver to the inlet port of the expansion valve, and then the compressor restarts.

3.2 Refrigerant superheating

Typically, the low pressure vapour is superheated inside the evaporator prior to entering the compressor suction line. This process is controlled by the expansion valve based on the saturated evaporating pressure and temperature, and

the actual suction temperature. For most common refrigerants, suction superheating of about 5°C is high enough to prevent saturated vapour from entering the compressor and to avoid generating liquid droplets during the adiabatic compression (process BC in Fig. 3a). However, with some new high-temperature refrigerants, if the superheating (process DE in Fig. 3b) is not sufficient, the compression adiabatic curves can cross the refrigerant saturation line, and the discharge fluid may contain a saturated liquid-vapour mixture.

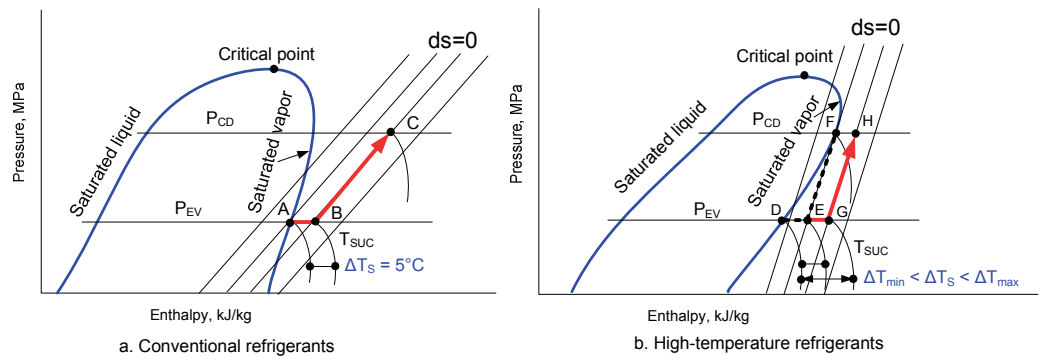

Figure 3: Comparison of adiabatic compression processes in p-h diagrams; s: specific entropy

To avoid that situation, much higher suction superheating has to be achieved inside an optimum range. For several high-temperature refrigerants, the optimum range is between 16.6°C and 22.2°C at any saturated suction pressure (Minea, 2007). In order to keep superheating inside the recommended range, special suction pressure and temperature sensors send proportional signals to the system process controller. It will automatically calculate the actual evaporating temperature according to the refrigerant saturated pressure-temperature correlation. For each pair of suction and evaporating temperatures, the actual suction superheating (ΔT_s) of the refrigerant is automatically calculated. Based on the actual superheating figure, the process controller will automatically modulate the opening of the electronic expansion valve to keep it inside the desired range.

3.3 Dehumidification rate

Dewatering is a complex thermodynamic process, which depends on the type of material and its initial temperature and moisture content. The efficiency of the heat pump drying process highly depends on the correlation between the material dewatering properties and the heat pump dehumidification rate. Also, the quality of the material at the end of the drying process depends on the heat pump moisture extraction rate. The heat pump dehumidification rate can be adapted by varying the air flow rate through the evaporator. Because the humidity content of the material is very high at the beginning of the dehumidification process and much lower at the end, it is necessary to continuously increase the air mass flow rate across the evaporator. This operation makes it possible to cool the warm and humid air sufficiently to reach the dew point and provide a quasi-constant dehumidification rate according to the material dewatering capacity. It also avoids too fast drying processes and stresses to the dried material. The following time-based equation allows the heat pump blower flow rate to be changed (Minea, 2007):

$$v = C * t + 25 \tag{5}$$

where v is the blower electrical current frequency (Hz); t - the drying time (hours) measured from the beginning of the dehumidification process with the heat pump; C – empiric constant. For softwood, such as white spruce and pine, the dehumidification cycles with high-temperature heat pumps take up to 60 hours. In this case, the constant C could vary from 0.2 (high dehumidification rate) to 0.5 (low dehumidification rate) (Fig. 4a). Fig. 4b shows an example of a heat pump blower optimized operating profile based on the described control strategy.

3.4 Wet bulb temperature setting

As noted, a major issue consists in increasing the technical life of the drying heat pump compressors. Too many short on/off cycles, i.e. frequent transient regimes, contribute to the wear and tear of these sensitive mechanical devices and to lowering the system overall efficiency. To further fix this problem, it is recommended in this section to reduce the number of compressor short on/off cycles by automatically setting the dryer wet bulb temperature based on the actual measured values (Minea, 2007). Generally, before starting a new drying cycle, the dryer air dry-bulb (DB_{SET}) and wet-bulb (WB_{SET}) temperature settings are preset for each step of the dehumidification process. Then, the actual air dry-bulb (DB_{ACT}) and wet-bulb (WB_{ACT}) temperatures are measured inside the dryer and

automatically transferred to the process control program. The heat pump compressor starts at the end of the preheating step. At this point in time, the actual air wet-bulb temperature (WB^1_{ACT}) is much higher than the first preset wet-bulb temperature (WB^1_{SET}). Because the compressor is running and the dryer is a tight enclosure with all air vents closed, the difference between the actual air wet-bulb temperature and the first preset wet-bulb temperature gradually decreases over time. When this difference reaches, for example, 5.5°C, the process control program will automatically lower the preset wet-bulb temperature, for example, by 2.7°C.

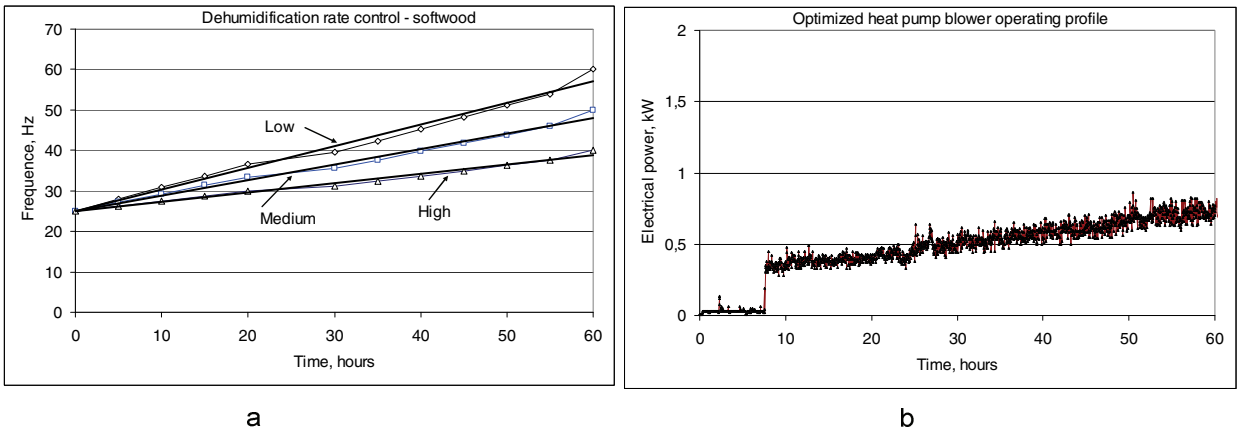

Figure 4: Control method of the softwood dehumidification rate (a) and example of a heat pump blower optimized operating profile (b)

After modifying the first air wet-bulb temperature setting the heat pump compressor continues to run in dehumidification mode. The difference between the actual air wet-bulb temperature and the second preset wet-bulb temperature (WB^2_{SET}) continues to gradually decrease over time. When the difference between the actual air wet-bulb temperature and the second preset wet-bulb temperature also reaches 5.5°C, the control program will lower once again the second preset wet-bulb temperature by 2.7°C. This control strategy has to be applied several times depending on the physical characteristics of the dried material. The wet-bulb temperature is consequently set according to the following equation (Minea, 2007):

$$WB^{i}{}_{SET} = WB^{i-1}{}_{SET} - 2.7°C \tag{6}$$

where i is the number of successive wet-bulb temperature settings.

4. CONCLUSIONS

This paper presents a number of improvements concerning the integration of high-temperature heat pumps and dryers on both the refrigerant and air-sides. Strategies are also developed to control refrigerant migration and superheating in order to avoid compressor short on/off cycles and thus improve safe system operation, and extend the compressor technical life. Finally, the methods presented for controlling the dehumidification rate and the dryer wet-bulb temperature settings are aimed at improving system reliability and preserving the final quality of the dried material.

NOMENCLATURE

$\dot{m}$	flow rate (kg/s)
COP	coefficient of performance
D_b	wood basic density (kg/m^3)
h	mass enthalpy (kJ/kg)
V_G	green wood volume (m^3)
$\dot{m}$	mass flow rate (kg/s)
$\dot{W}$	electrical power (kW)
MC	moisture content (dry basis, %)
ν	frequency (Hz)
$\dot{Q}$	heat flux (kW)
WB	wet bulb temperature (°C)
t	time (hrs)

Subscripts

EV	evaporator
CD	condenser
C	compressor
in	initial
fin	final
hum	humid

REFERENCES

Lewis, D., 2003, High temperature dehumidification drying system, U.S. Patent 0208923.

Minea, V., 2007, High-temperature heat pump for wood drying – refrigeration circuits and control improvements, Hydro-Québec patent disclosure.

Minea, V., 2008, Design and control optimization of drying heat pumps, *Proceedings of the 16th International Drying Symposium - IDS 2008*, Volume B, p. 739-745, Hyderabad – India.