

HIGH-TEMPERATURE HEAT PUMP FOR RESINOUS TIMBER DRYING

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ABSTRACT

In the 1970s and 1980s, the Canadian kiln drying industry promoted dehumidifier concepts, but the performance of such systems was often disappointing due to air flow and control problems, inadequate dehumidifying capacity and inappropriate kiln structures. The reliability of the systems was often poor and equipment suppliers did not provide enough information on the actual performance of their systems. However, the best solution that would allow heat pumps to be effective as a means for drying wood would be to use them in combination with a traditional energy source at high temperatures to obtain similar drying speeds and energy savings. Resinous timber from Eastern Canada lends itself well to high-temperature drying with traditional dryers, but this technology does not yet exist on the market, all the more so since early 90s environmental issues imposed the replacement of traditional CFCs that were used as high-temperature refrigerants. In order to reduce the duration of the drying cycles, to obtain a better quality of wood and to save energy, a small-scale high-temperature heat pump prototype was developed and laboratory tested. This paper summarizes the original features of this system as well as its main thermodynamic parameters and energy performance. The current research program aims at ensuring the maximum reliability of the product, developing strong Canadian design and manufacturing skills, and preparing the deployment of the product on the national wood drying market.

Key Words: *heat pumps, wood drying*

1 INTRODUCTION

Canadian forests account for about 10% of the planet's forests (45 % of the country's territory) and the wood industry contributes close to 3% of the country's GDP. In 2002, Canadian sawmills supplied 72 million m^3 of resinous timber (including white and black spruce, jack pine and balsam fir), of which approximately 24% was produced in the province of Québec (eastern Canada) (Canada Statistics 2002). About 85% of this production is presently dried with bark boilers, 10% with fossil fuels (natural gas, propane or oil), and only 3% with low-temperature heat pumps. The potential reduction in energy use by dry kilns in Canada would be 5.5 PJ per year, or 335 kilotons per year in carbon dioxide (CO_2) emissions. Furthermore, it was estimated that CO_2 emissions could be reduced by an additional 90 kilotons per year through a decrease in the amount of lumber that is downgraded (NRCAN 2003). So far and despite numerous attempts, the heat pump technology has not succeeded in securing a significant portion of the market share, the maximum operating temperature (50 to 55°C) being one of the main reasons. However, in order to ensure a short drying time, lumber producers opt for drying temperatures varying between 70°C and 120°C. One solution seemed to be the development of electrical high-temperature heat pumps (HT-HP) in combination with fossil combustibles as back-up energy sources (hybrid systems). This would allow the drying of resinous lumber with performances and velocities similar to conventional technologies, while generating energy savings and enhancing product final quality. The capacity of heat pump dryers to recover energy represents lower operating costs, which should compensate for the higher initial capital costs, decreases the wood drying industry's vulnerability to frequent fluctuations in the fossil fuels prices, in spite of their limitations in terms of maintenance costs (motors, compressors, filters, heat exchangers) and refrigerant leak risks. There are also retrofit possibilities, without any major additional investment for new boilers, generally very expensive. High temperature heat pumps are not yet available on the Canadian market, all the more so since, at the beginning of the 90s, the production of traditional CFC high-

temperature refrigerants was prohibited. Because of the complexity and the strongly non-linear behaviour of the thermodynamic processes involved in wood drying, laboratory development and experimentation were undertaken. The main objective of the tests presented here was to stabilize the heat pump's thermodynamic parameters and thus validate the capability of the refrigerant to work at high temperature conditions.

2 SYSTEM DESIGN AND CONTROL PRINCIPLES

The rail-loaded, air-forced laboratory dryer (19 m³), built with insulated aluminium panels, is normally filled in a single operation, and the wood stacks remain stable during the drying process (Figure 1). It contains two independent heating systems (steam and electricity) and a 3-speed, 4-transversal bladed central fan that periodically changes its direction of rotation (Figure 2). A combination of two control approaches has been used: setting up time-based drying programs and continuous control of the air dry/wet temperatures. The nominal inlet pressure (960 kPa,a) and mass flow rate of the saturated steam, generated by a natural gas-fired boiler (annual average efficiency of 80%), are continuously adjusted to maintain, as a back-up energy source, the kiln dry bulb temperature, according with pre-established drying schedules. The actual wet bulb temperature controls the compressor operating status in direct relation with the wet bulb setting. The components of the control loop include temperature measurement instruments, a PID regulator and a comparator for determining the difference between the actual temperature in the kiln and the target set by the drying schedule. The air vents automatically open when the dry bulb temperature exceeds the setting point by a pre-determined value.



Fig. 1. View of the Experimental Dryer

The main original feature of the first laboratory HT-HP prototype was the use of a new working fluid-blend (HFC) which replaces the best traditional high temperature refrigerants (CFCs) banned by the Montreal Protocol (1987), and of several improved devices and control strategies. The critical temperature of the refrigerant, 125 °C (257 °F), is sufficiently high compared with the highest temperatures that can occur in the system. The prototype includes a cabinet housing the heat pump's heat exchangers and air blower. An open, reciprocating compressor and other mechanical components and controls are placed on an adjacent platform (Figure 3). The refrigerant flow is controlled by an electronic step-motor expansion valve with microprocessor controlled by the actual refrigerant pressure and temperature. The hot and humid air, drawn up from the dryer by a bottom-sited grille, is conditioned (cooled, dehumidified and reheated by using both the sensible and latent heat recovered and the electric energy of the compressor/blower), before returning inside the dryer. A motorised damper continually adjusts the airflow rate.

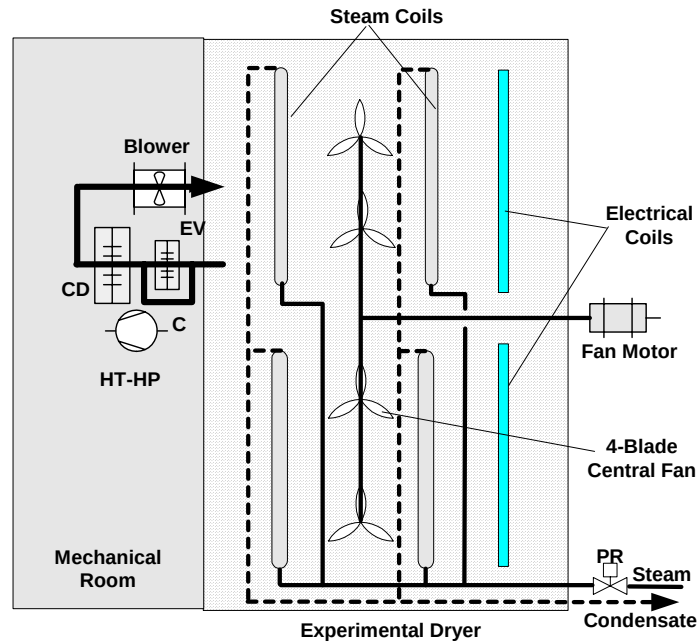


Fig. 2. Relative Location of the High Temperature Heat Pump's (HT-HP) and Dryer's Main Components. C: Compressor; EV: Evaporator; CD: Condenser; PR: Pressure Regulator

Each drying cycle has begun by a preheating cycle. As soon the dry bulb temperature setting has been reached, the compressor started up (step 1) and then the steam valves modulated the flow to compensate the dryer heat losses and maintain the dry bulb temperature around the setting point. The compressor normally has run continuously, but the electrical frequencies can be varied from 60 to 30 Hz upon the actual drier's wet bulb temperature versus the setting point. The heat pump's blower can also run at variable speeds corresponding at 10 to 100 % of its full air flow rate. When the actual wet bulb temperature drops under the setting point, the compressor shuts down. A 5-minute delay is allowed before restarting, but only if a minimum pre-established suction pressure is detected. The typical resinous timber dehumidifying cycle includes up to six successive drying steps. The final shutdown occurred automatically in accordance with the pre-established duration of the last drying step, or manually, when the wood's average humidity content (dry basis), measured twice a day, reached the final desired value (17 to 20 %).



Fig. 3. View of the High Temperature Heat Pump Prototype

3 PROTOTYPE IMPROVEMENTS AND DRYING SCHEDULES

The first laboratory prototype, built in November 2002, wasn't able to be immediately operational with its original programmable electronic chips. The main reasons included inappropriate time-based starting sequences, lack of a solenoid valve preventing liquid flooding during the off periods with indoor hot dryer conditions, use of a non-tight expansion valve with deficiencies at the software level and a poorly located temperature probe, and completely inefficient control of the air dampers. Several improvements to the prototype, as insertion of additional devices and control sequences, as compressor and blower variable speed-drives, manual step-motor control for the electronic expansion valve, appropriate adjustments for low and high compressor pressure limit switches, solenoid valve on the liquid line and adjustable time relay on the low pressure limit, have finally allowed the unit to become fully operational.

Four hybrid drying tests, with HT-HP as main heating system, and steam as back-up energy source, and one conventional test (steam heating only) were performed with resinous wood (white spruce, 4.9 x 9.8 x 250 cm) (Table 1). As noted, the first specific objectives were to stabilize the thermodynamic parameters of the heat pump and then determine their optimum values, establish and progressively improve the drying schedules, the back-up heating control, and quantify the overall energy performance of the drying system. The dryer's dry and wet bulb temperatures were chosen depending on the wood's humidity content, while the setting changes were determined on the basis of a pre-established drying time schedule for each step. Because sapwood discoloration may occur during the first drying cycles if the dryer temperature is kept too low, each batch had to be preheated over a period of 5 to 8.2 hours at dry bulb temperatures ranging between 87.7°C (190°F) and 93.3°C (200°F) and wet bulb temperatures ranging between 68.3°C (155°F) and 79.4°C (175°F), in order to destroy any micro-organisms that were causing the sap-stains. The preset duration of the following steps (1 to 5) varied between 10 and 20 hours. The heat pump was in operation for between 30.8 hours (test #3) and 43.8 hours (test #5). As noted, the drying schedules varied depending upon the wood's humidity content, defined as the total water mass inside a piece of wood divided by its mass in dry state, measured at the beginning of each drying cycle (Cech & Pfaff 2000). For example, for an initial humidity content of 40 to 30%, the dry bulb temperature varied from 76.6°C (170°F) to 85°C (185°F), and the wet bulb temperature, from 62.7°C (145°F) to 68.3°C (155°F). Finally, at humidity contents lower than 30%, the dry bulb temperature increased to 87.7°C (190°F) and the wet bulb temperature, to 71.1 °C (160°F). The actual initial moisture content of the dried wood (dry basis) varied from 32.4% (test #2) to 47 % (test #3), whereas it reached 14.7% (test #5) to a maximum of 16.9% (test #6) at the end of the respective tests. In the case of the conventional cycle, the final wood moisture content was of 16% (Table 1).

Table 1. Performance Test Durations and Wood's Initial and Final Humidity Contents

Drying Test	Wood Average Moisture Content (dry basis) (%)		Duration (hours)	
	Initial	Final	Preheating	Hybrid with HT-HP
#2_PERF	34.4	16.8	5.05	32.5
#3_PERF	47.0	16.6	5.56	30.8
#5_PERF	41.1	14.7	8.20	43.8
#6_PERF	43.5	16.9	8.03	41.0
CONV_PERF	60.2	16.0	53 - total	

4 PERFORMANCE TEST RESULTS

The experimental performance tests presented here were performed with very stable thermodynamic parameters, such as pressures, temperatures and humidity, while operating the dryer at constant and relatively high temperatures according with the drying schedules. The average compressor suction pressure was stabilized around 480 kPa,a (55 psig), whereas the average discharge pressure varied between 1,410 and 1,825 kPa,a (190 to 250 psig) with a compression ratio of 2.9 to 3.8 (Figure 4). The pressure variations on this graph are due to the periodical changes in the central fan's rotation (every 2 or 3 hours), and to the fact that the heat pump takes in and returns air on the same side of the dryer. As a result, when the air

passed across the wood stack before entering the heat pump, the entering air temperature was approximately 5.5°C (10°F) lower than when the air directly entered the heat pump without passing across the wood stack.

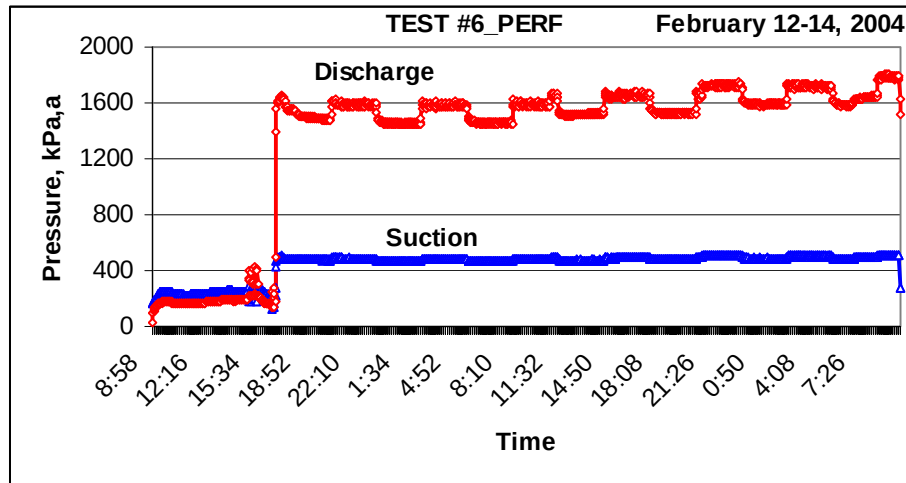


Fig. 4. Compressor's Suction and Discharge Pressures (TEST #6_PERF)

Under these conditions, the suction temperature varied from 64°C to 72°C (147.2°F to 161.6°F), while the average discharge temperature reached 104°C (220 °F), with highest values of up to 112°C (233.6°F) (Figure 5). Despite these relatively high temperatures, no superheat at the discharge or instability due to the proximity of the refrigerant's critical zone have been observed, and the isentropic compression efficiency always got near 95%.

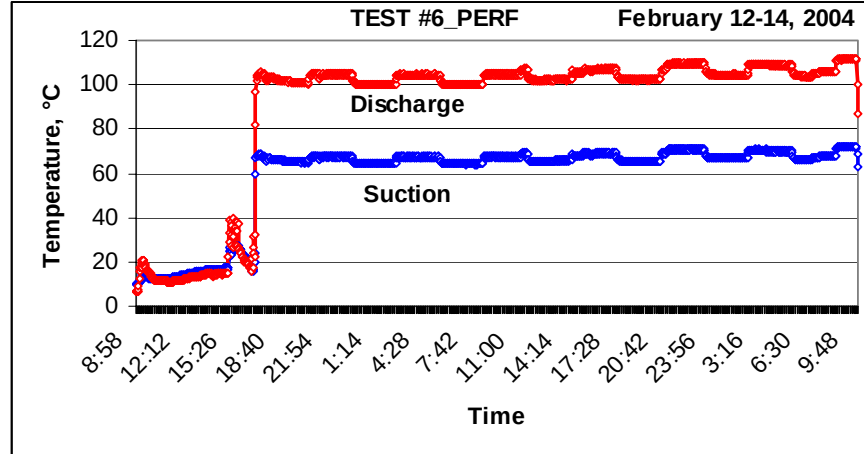


Fig. 5. Suction and Discharge Temperatures (TEST #6_PERF)

The evaporation temperatures were stabilized at around 44.8°C (112.7°F), which indicates that the suction superheating varied from 15°C to more than 22°C (27 to 39.8 °F). These values, which can seem to be relatively high, were necessary because of the particular configuration of the isentropic curves on the refrigerant's superheated zone. Stable and sufficiently high condensing temperatures (91°C to 97.5°C or 195.8°F to 207.5°F), which are essential parameters in this particular application, allowed to heat the supplied air up to 93 °C (199.4°F), which was about 5.5°C to 11.1°C (10°F to 20°F) higher than the dryer's actual dry bulb temperature (Figure 6). However, at higher dryer operating temperatures, the condensing temperatures may easily increase to up to 115°C (239°F). The condenser sub-cooling amount always varied over a normal range for this application (4 to 5°C or 7.2 – 9°F).

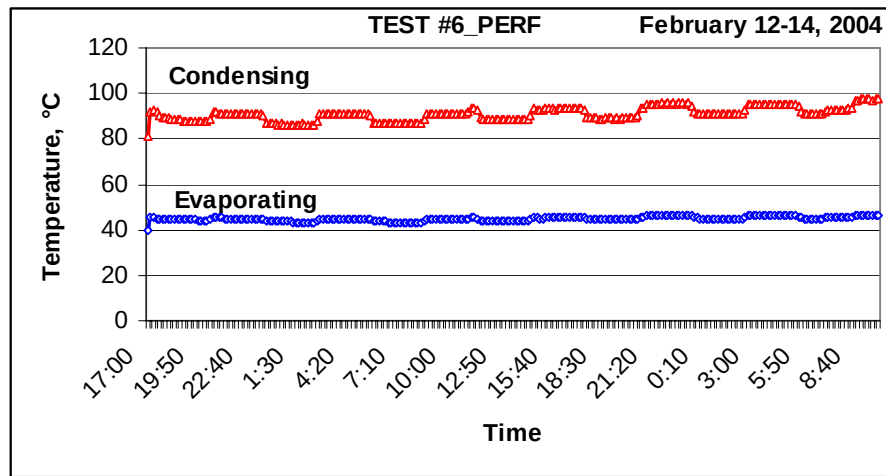


Fig. 6. Heat Pump Saturated Temperatures (TEST #6_PERF)

All these parameters were obtained with an average compressor input electrical power of 7 kW and a heat pump blower shaft power varying from 0.94 kW to 1.49 kW, upon the imposed rotation speed, and a refrigerant mass flow rate of about 0.4 kg/s (at 60 Hz). The maximum temperature of the air going through the condenser (air supply) increased from 80.7°C (177.2°F) to 93°C (199.5°F) (Figure 7), but the maximum values constantly exceeded 87.7°C (190°F), and the relative humidity varied around 44%. As noted, the temperature variations observed on graphs are attributed to periodical changes in the fan's rotation.

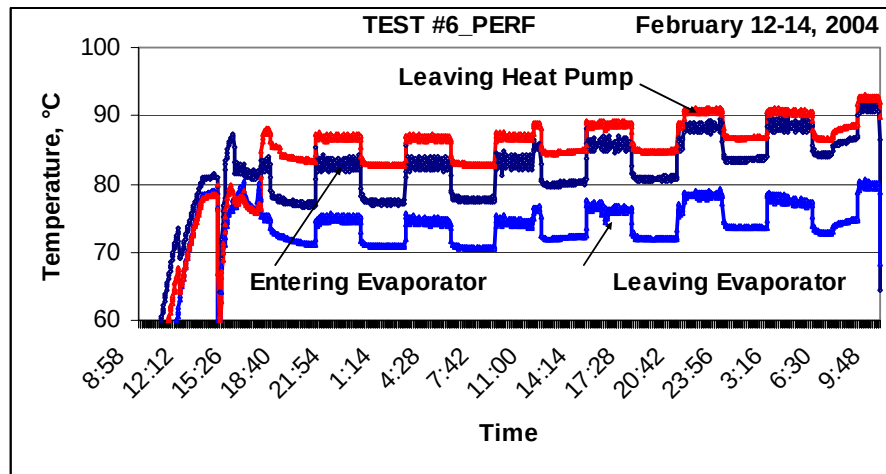


Figure 7. Air Dry Bulb Temperatures Entering and Leaving the Heat Pump (TEST #6_PERF)

The low average pressure losses across the evaporator (59 kPa) and the condenser (35 kPa) (Figure 8) demonstrate that these heat exchangers have been correctly designed from a hydraulics point of view and in consideration of the transport properties of the new refrigerant.

The thermal power recovered from the dryer during a typical hybrid dehumidifying process was 28.2 kW, while the corresponding thermal power supplied to the dryer amounted to 42.2 kW. In consideration of the thermal output of the air bypassing the evaporator, the heat pump's coefficient of performance, defined as the ratio between the useful thermal power supplied and the electrical power input, was 6.3. However, taking into account the dehumidifying thermal power (sensible and latent) attributed to the evaporator, the actual COP was only 5.02.

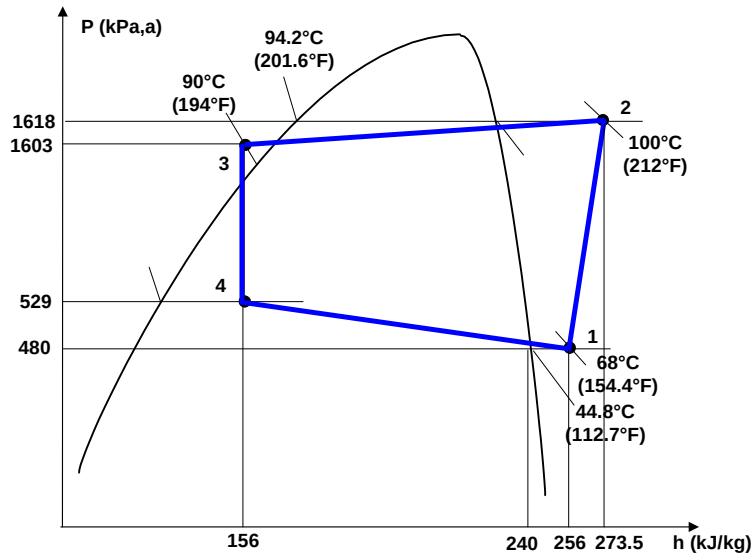


Fig. 8. Typical Average High Temperature Heat Pump's Thermodynamic Cycle

Figure 9 represents an example (test #5) of the relationship established between the actual and setting temperatures inside the dryer. On the dry bulb temperature curve (actual DT), only the maximum values are relevant because the readings were only registered on the side of the heat pump inlet, whereas the dryer was controlled based on the dry bulb temperature alternately measured on the side where the fan was circulating the air. The slope of the wet bulb temperature curve (actual WT) was obtained so as to avoid crossing the setting wet bulb temperature curve established by the drying schedule. In automatic operating conditions, the compressor must shut down when the actual wet bulb temperature is 1.6°C (3°F) (adjustable value) lower than the setting point, and restarts when this temperature reaches a certain value compared to its setting (for example, 1.1°C or 2°F higher).

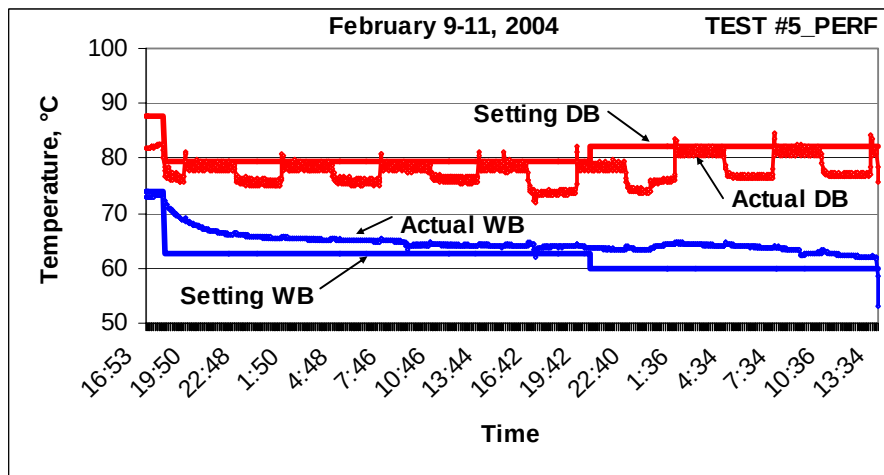


Fig. 9. Actual and Setting Dryer's Dry (BD) and Wet (WB) Bulb Temperatures (TEST #5_PERF)

By assuming a number of hypotheses based on the local costs (2003) of electricity (0.0356 US \$/kWh) and natural gas (0.31 US \$/m³), and on the annual average thermal efficiency of boilers (80%), it was estimated the system's energy performance and relative costs. During the preheating step, the electrical energy consumption (central fan) represented less than 1.4% of the total equivalent energy consumption during this step, whereas, during the following dehumidifying period, it has averaged 25% of the total electrical energy consumption. The heat pump's electrical energy consumption (compressor and fan) was established at 11% of the total consumed energy during the dehumidifying cycle. The total water quantities extracted by the prototype were respectively 534 kg/cycle (test # 3), 451 kg/cycle (test #3) and

472 kg/cycle (test #6). Taking only the heat pump's energy consumption (compressor and fan) into account, the heat pump's Specific Moisture Extraction Rate (SMER), a parameter defined as the water mass extracted by the heat pump (kg_{water}) divided by the electrical energy consumed (compressor and heat pump fan) (kWh), varied from 1.35 to $1.93 \text{ kg}_{\text{water}} / \text{kWh}$, performances that are considered satisfactory at this stage of the study. Consequently, the total specific energy costs (including natural gas consumption), varied between $5.3 \text{ US}\$/\text{m}^3$ and $8.9 \text{ US}\$/\text{m}^3$ (2003 local energy rates), while the specific cost of the conventional test was of $12.2 \text{ US}\$/\text{m}^3$.

5 CONCLUSIONS

This paper proves that the drying technology with high-temperature heat pump can be successfully applied to the drying of resinous wood. After several improvements to the original prototype, including the use of variable speed devices (for heat pump compressor and blower) and a new generation expansion valve and controllers, as well as the refining of the back-up heating system controls, the heat pump operated at high entering air temperatures in very stable thermodynamic conditions. Any negative phenomena, caused by the refrigerant critical temperature, its chemical properties or compatibility with the lubricant, were identified. The heat pump supplied the air to the dryer at dry bulb temperatures up to 11.1°C (20°F) higher than the dryer setting values, and the wood's moisture content (dry basis) varied between a maximum of 60.2% at the beginning of the drying cycles and a minimum of 14.7% at the end. The maximum Specific Moisture Extraction Rate (SMER) was 16% lower than the initial objective, which – however - had previously been obtained during several field tests with two similar industrial-scale high temperature heat pump prototypes (Minea 2004). The actual energy consumption was 27.3% to 56.7% lower than the energy consumed during the conventional (steam) drying test, whereas the average reduction in specific energy costs, compared to the costs of the local wood drying industry, was approximately 35 %. The current step of the research aims to integrate an advanced process controller, preferably based on an efficient moisture content measurement, build and test an industrial-scale drying prototype.

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