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Simulation, validation and demonstration of energetic optimization possibilities of industrial drying processes by use of an unique heat pump dryer test bench

Jamy Logie, Minh Cuong Tran, Bruno Vanslambrouck*

Department of Flow, Heat and Combustion Mechanics, Ghent University – UGent, Campus Kortrijk
Research Group Thermal Energy in Industry (TEI)
Graaf Karel de Goedelaan 5, 8500 Kortrijk, Belgium

Abstract

It is well established that heat pump dryers provide significantly higher energy efficiencies and lower greenhouse gas emissions than conventional dryers. Heat pumps have been around for several decades, yet research and development of heat pump drying technology has only recently advanced to industrial applications. This paper describes the development and simulation of an electrically driven heat pump dryer test bench, in which a drying process is replicated by means of water evaporation. The test bench is developed as a part of the CORNET-HP4Drying project (www.hp4drying.eu)**. This project aims to integrate heat pumps into different industrial drying processes in order to exploit the corresponding advantages by energetic and environmental optimization. By adapting heat pumps to operate under varying temperature levels and developing innovative control systems, HPs could better follow the drying process, including batch processes and thus increase efficiency.

Keywords: heat pump drying; test bench; development; simulation; HP4Drying

1. Introduction

Drying plays an integral role in almost every industrial branch and consumes vast amounts of energy. In the industrialized regions, drying processes currently account for 10-25% of the total industrial energy consumption [1]. For the paper processing sector, drying is responsible for 30% of the total energy consumption, for textile this is 50% and up to 70% in the case of wood [2]. Furthermore, most drying

* Corresponding author. Tel.: +32-56 24 12 27
E-mail address: bruno.vanslambrouck@ugent.be

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installations rely on fossil fuels for the production of heat, hereby contributing significant amounts of greenhouse gas emissions. The exhaust air of dryers is hot and contains large amounts of water vapor, but it is not always straightforward to recover this sensible and latent heat. Heat pump (assisted) drying provides a way to recover large amounts of this heat, reducing energy consumption by 60-80% at a given temperature [3].

2. Test bench description

2.1. General

Within the HP4Drying project a dryer test bench is being developed. The test bench aims to demonstrate various optimization possibilities to the air and heat pump cycle and expands the field of application towards a higher temperature range. The air cycle optimizations include an adjustable recirculation ratio, air-to-air heat exchanger between in- and output. In case the electrically driven heat pump is enabled, cycle optimizations include the use of a suction line heat exchanger, subcooler, evaporator bypass, high temperature fluid, an innovative control system and real-time visualization on the respective diagrams. The heat pump itself is designed to be transportable, which allows demonstrations or research activities on other locations. The heat pump has a thermal capacity of 10kW$_{th}$ at a heat source temperature of 35°C, with higher output up to 30kW$_{th}$ possible at a heat source temperature of 80°C. Fig. 1 presents the layout and simplified schematics of the test bench as installed in the lab.

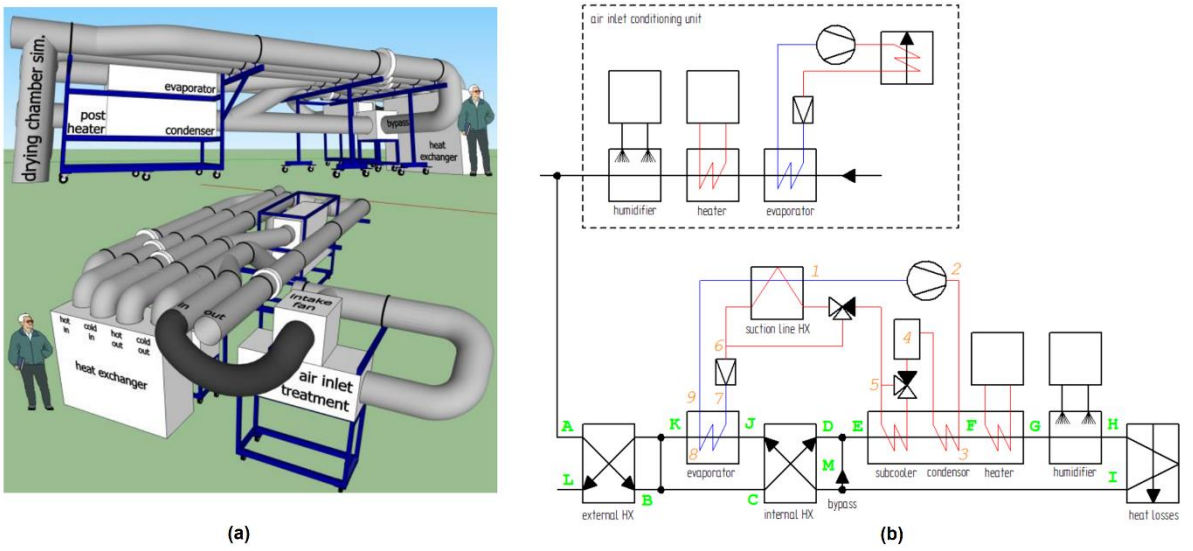


Fig. 1. Test bench layout (a) and simplified schematics (b)

To get the cycle architecture as flexible as possible, all heat exchangers can be (internally or externally) bypassed and all flows controlled (Fig. 1b). The heat pump features an open compressor, which enables precise mechanical power measurement and factors out compressor motor efficiency. In turn, this allows practical simulation of a gas engine powered heat pump dryer. Heat that would be recovered from the gas engine and exhaust at a given mechanical output, can be added as electrical heat injected after the condenser, by using data from CHP units. To aid repeatability and optimization of a climate dependent control algorithm, the test bench is fed with conditioned air, capable of following a preset summer/winter day/night cycle.

2.2. Drying process reproduction

The aim is to reproduce in- and outgoing conditions of a drying process from monitored drying plants. A drying process can be quantified as a difference in humidity and enthalpy between in- and outgoing air of the drying chamber as shown in Fig. 2. The evaporation of moisture from the product is responsible for a rise in humidity (G-H), whereas an enthalpy drop (H-I) is caused by thermal losses through the drying chamber walls and gaps and heating up of the product.

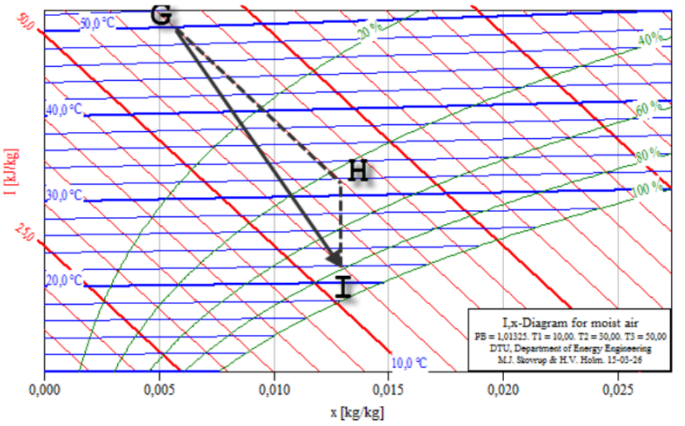


Fig. 2. Drying process reproduction

Reproduction is achieved by means of spraying water into the airstream, of which temperature and mass flow are controlled. This way the heat pump dryer can be tested for a wide range of different drying schemes and products with corresponding temperature ranges, allowing simulation of batch and continuous processes from ongoing case studies. To our knowledge a similar test bench has not yet been demonstrated.

3. SYSTEM CONFIGURATIONS AND PROCESS SIMULATIONS

To aid dimensioning of the test bench, a basic process simulation has been made with Matlab, using the Coolprop thermodynamic and psychrometric property libraries for fluids and humid air [4]. With this model, state points and performance parameters of the drying and heat pump cycle are calculated using the inputs, among others:

- System configuration and ambient conditions;
- Moisture extraction rate, fresh air ratio and evaporator bypass air ratio;
- Air temperature after the evaporator or after the condenser;
- Desired dryer inlet temperature;
- Relative humidity at dryer outlet.

To understand all optimization possibilities that can be demonstrated, the test setup is completed step by step with calculation of the corresponding energy use and saving.

3.1. Basic open-cycle dryer

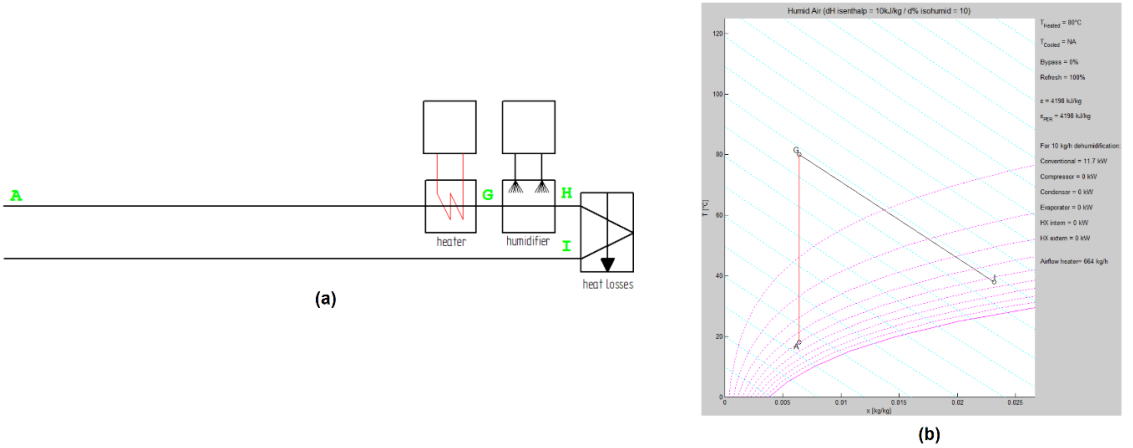


Fig. 3. Open-cycle dryer: simplified schematics (a) and an example of the simulation output in psychrometric chart for the drying air (b)

So far pressure losses, fan power input and drying mechanics have not been taken into account. Later on, the process simulation will be optimized and validated with the test bench, following the sequence as provided next. As a benchmark to compare the energy efficiency of more advanced dryers with, the first system configuration to be simulated is a basic open-cycle conventional dryer, as shown in Fig. 3a.

The efficiency of the heater is fixed at 90% and the drying process is considered to be isenthalpic, with a dryer outlet relative humidity of 55%. Ambient conditions are 18°C, 50% relative humidity. As shown in Fig. 3b, this results in a specific energy consumption ε of 4198 kJ/kg.

3.2. Partially closed-cycle dryer

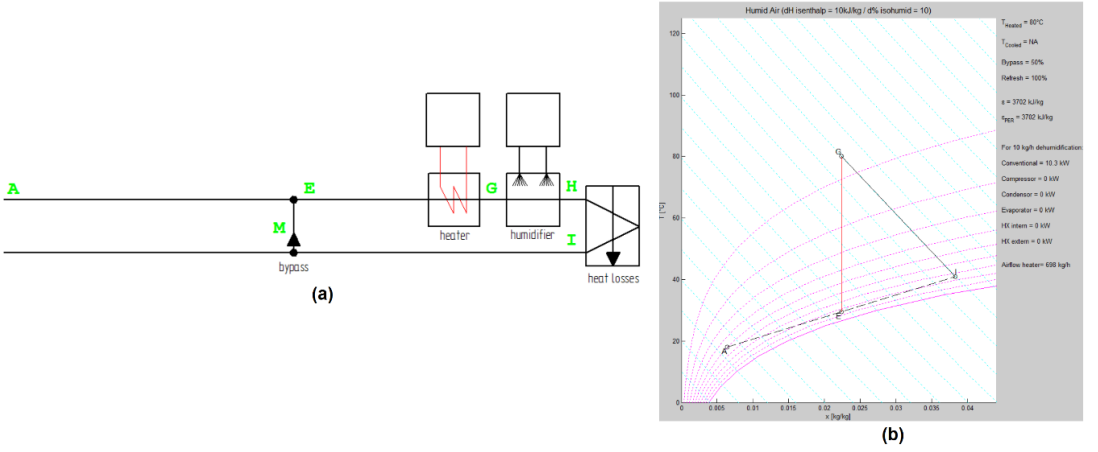


Fig. 4. Partially closed-cycle dryer: simplified schematics (a) and an example of the simulation output in psychrometric chart for the drying air (b)

One of the most widely used optimizations for energy saving in a drying process is partially recirculating of outgoing air, because additional moisture can be absorbed after reheating. The addition of a bypass fan is shown in Fig. 4a. Having a higher temperature than ambient air, less energy is needed to heat it up to the desired temperature, but a higher moisture content means the driving potential of moisture transfer from product to drying air diminishes. If the other parameters remain constant, recirculating drying air leads to a higher moisture content of the dryer outlet and thus efficiency at the cost of dryer capacity.

When calculating the energetic benefits by recirculation, it is suggested to keep the absolute humidity difference of the drying chamber constant [3]. By close approximation at these temperatures, this results in a rise from 55% to 75% of the dryer outlet relative humidity. Recirculation is set at 50%. In this example, as shown in Fig. 4b, specific energy consumption ε is decreased from 4198 kJ/kg to 3702 kJ/kg. Correspondingly, the two cycles require respectively 11.7kW and 10.4kW for a moisture extraction rate of 10kg/h. The test rig allows experimental dynamic optimization of the recirculation ratio for different drying conditions, even changing in- and outgoing drying air conditions.

3.3. Partially closed-cycle dryer with heat exchanger

Presented in Fig. 5a, a heat exchanger between in- and outgoing air offers clear advantages to the energy efficiency of a dryer, especially for high temperature drying applications. The ingoing air is preheated by cooling down outgoing air, possibly condensing water vapor. To keep the model simple, the heat exchanger effectiveness, defined as the ratio of the temperature rise of the ingoing air compared to the potential temperature rise, is set to 70%. As shown in Fig. 5b, this results in a specific energy consumption ε of 3113 kJ/kg, down from 3702 kJ/kg as shown in the previous example.

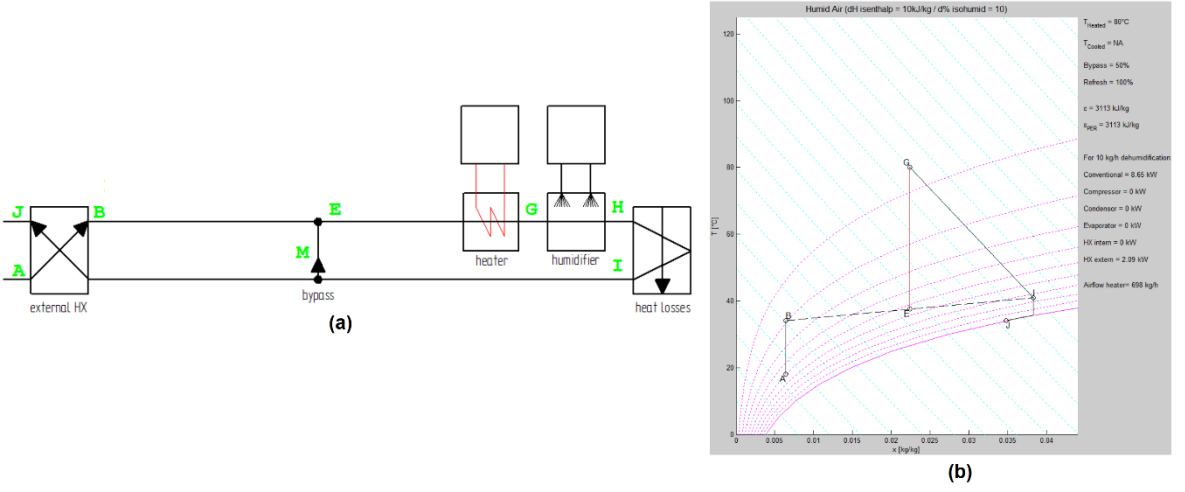


Fig. 5. Partially closed-cycle dryer with heat exchanger: simplified schematics (a) and an example of the simulation output in psychrometric chart for the drying air (b)

3.4. Heat pump dryer with evaporator bypass and external heat exchanger

A heat pump can be implemented in the previously shown system as shown in Fig. 6a. Whereas an air-to-air heat exchanger cannot transfer heat across the pinch point, a heat pump enables heat transfer from lower to higher temperatures. The selected heat pump fluid is R245fa, an HFC fluid with very similar characteristics as R134a, but with higher critical temperature and saturation temperatures related to the fluid pressure. Moreover,

compared to R134a, the theoretical COP is consistently slightly higher, with lower compressor discharge temperatures as an additional advantage. The fluid is fully compatible with standard commercially available HFC lubricants and components, respecting the normal limits for temperatures and pressures.

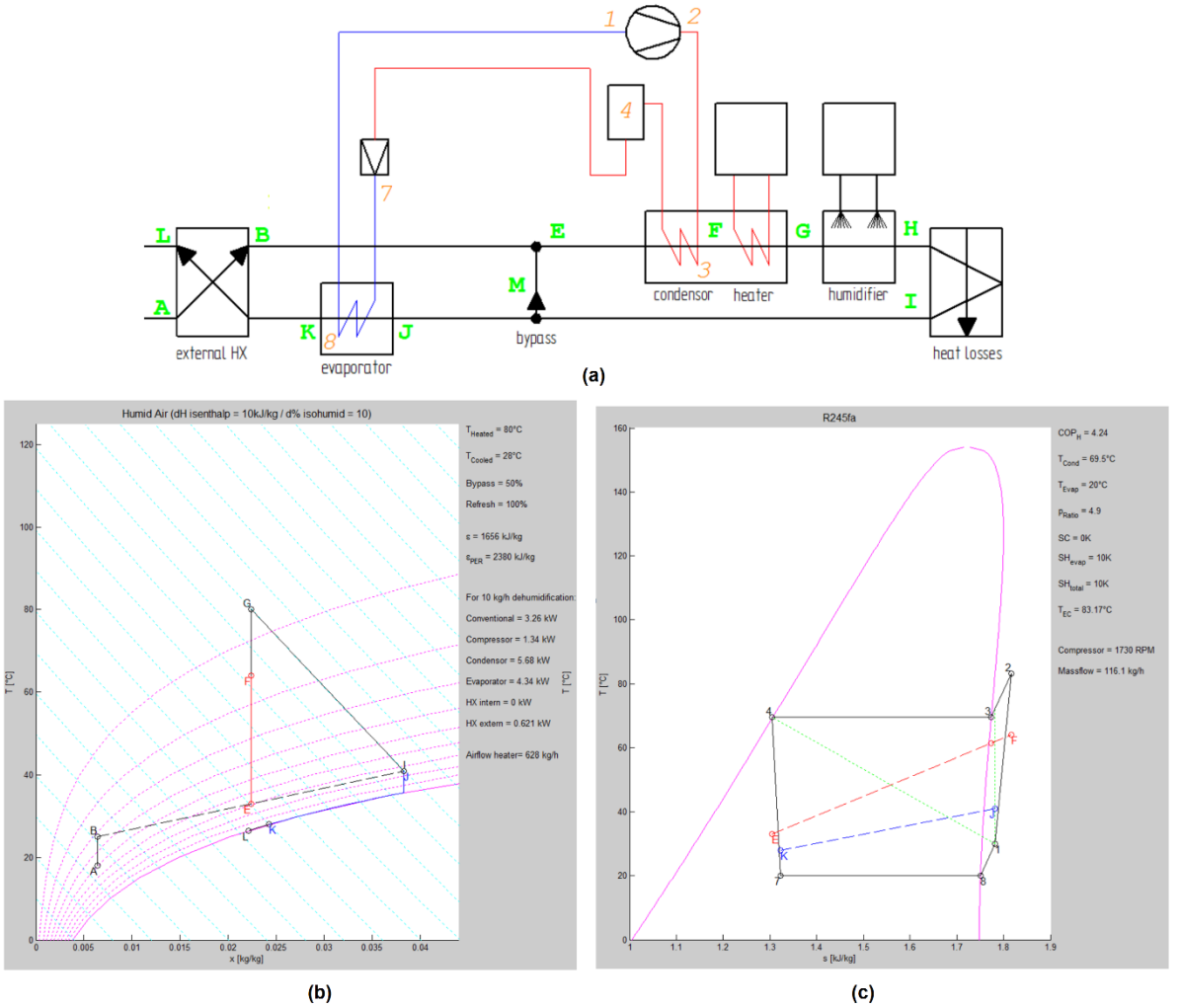


Fig. 6. Heat pump dryer with evaporator bypass and external heat exchanger: simplified schematics (a), an example of the simulation output in psychrometric chart for the drying air (b) and T-s diagram for heat pump cycle (c)

In the simulation, compressor isentropic efficiency is set to 70% and 8K is chosen as pinch points of condenser and evaporator. Superheat is set to 10K and subcooling is negligible, because in steady state conditions with a vessel that is not entirely full, liquid enters and leaves the vessel at saturation conditions. In this example, the energetic optimum is reached when the saturation temperatures of the heat pump are 20 and 69.5°C. As shown in Fig. 6b, this results in a specific energy consumption ε of 1656 kJ/kg and 2380 kJ/kg if a primary energy ratio of 2.5 is taken into account, referring to a commonly used 40% average conversion efficiency of electricity generation. Note how the external heat exchanger has a much smaller impact on the

system as the evaporator cools down the outgoing air close to ambient temperature. Fig. 6c represents the accompanying heat pump cycle.

3.5. Heat pump dryer with evaporator bypass, external heat exchanger and recirculation

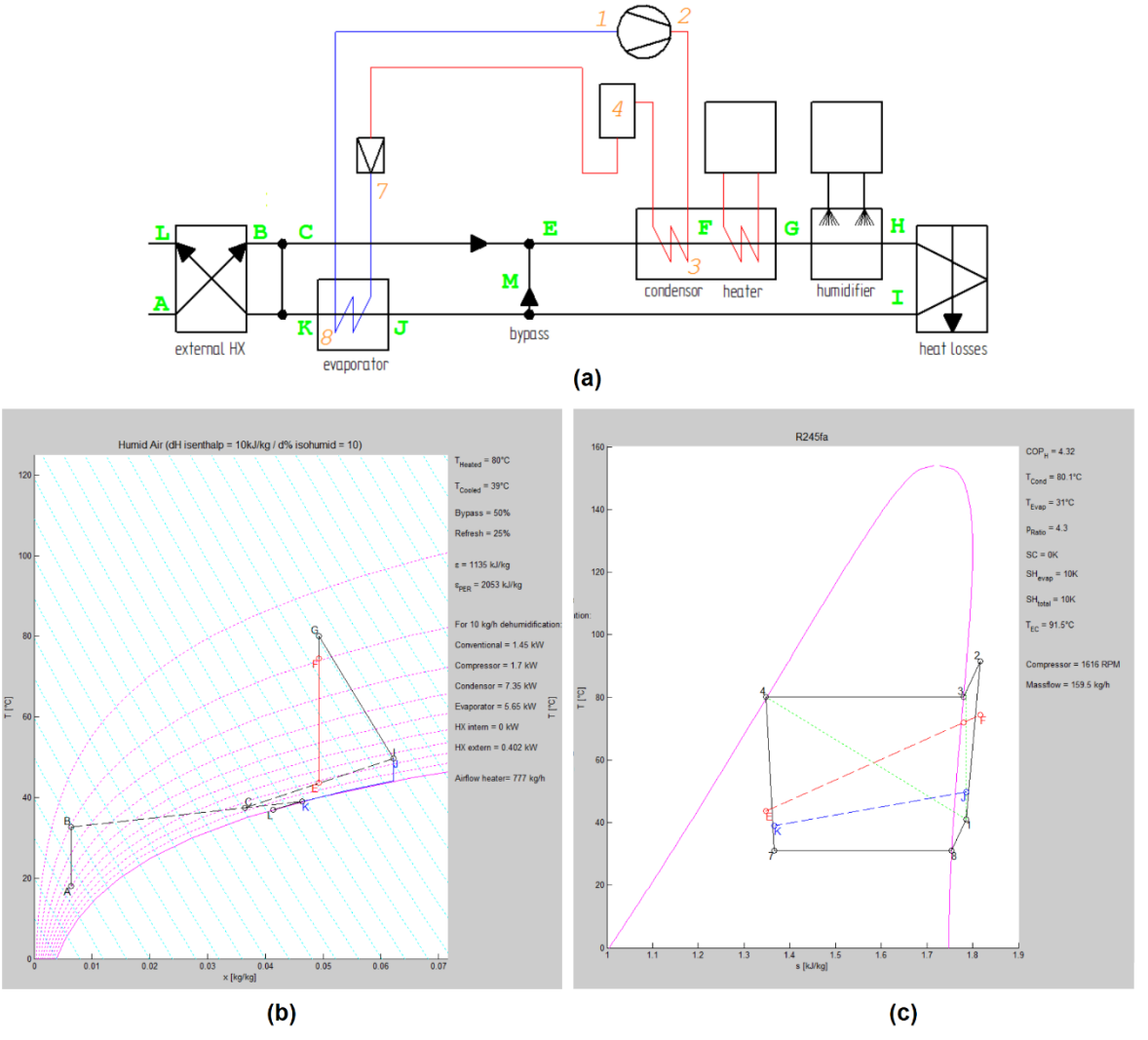


Fig. 7. Heat pump dryer with evaporator bypass, external heat exchanger and recirculation: simplified schematics (a), an example of the simulation output for in psychrometric chart for the drying air (b) and T-s diagram for heat pump cycle (c)

As with conventional drying, the air can be (partially) recirculated leading to a (partially) closed cycle (see Fig. 7a). In most cases the energetic optimum for medium/high temperature drying is a more closed cycle, whereas low/medium temperature drying favors a cycle with more fresh air. Note that a fully closed cycle with a near-isentropic drying process requires an external condenser to dissipate excess heat, as the condenser has a higher thermal capacity than the evaporator because of added energy by the compressor . While the dryer inlet

temperature is fixed, different cycles may yield a different dryer inlet humidity. To retain dryer capacity, this would require additional fans in the drying chamber for increased air speed and/or possibly additional (constructive) changes depending on the product. As there is no sub-model for drying mechanics at this point, outlet humidity will remain constant regardless of air flow.

Note how the ideal evaporator pressure is increased as only 25% fresh air is being introduced in the cycle. As the air cycle shifts to a higher absolute humidity, the air going to the evaporator becomes a better heat source. As shown in Fig. 7b, this results in a specific energy consumption ε of 1135 kJ/kg and 2053 kJ/kg if the primary energy ratio is taken into account. Fig. 7c represents the accompanying heat pump cycle.

Maintaining the higher-pressure ratio from Fig. 6c would result in too much heat extraction and could result in a too high temperature after the condenser. With this lower pressure ratio, a higher COP is obtained.

3.6. Heat pump dryer with evaporator bypass, internal/external heat exchangers and recirculation

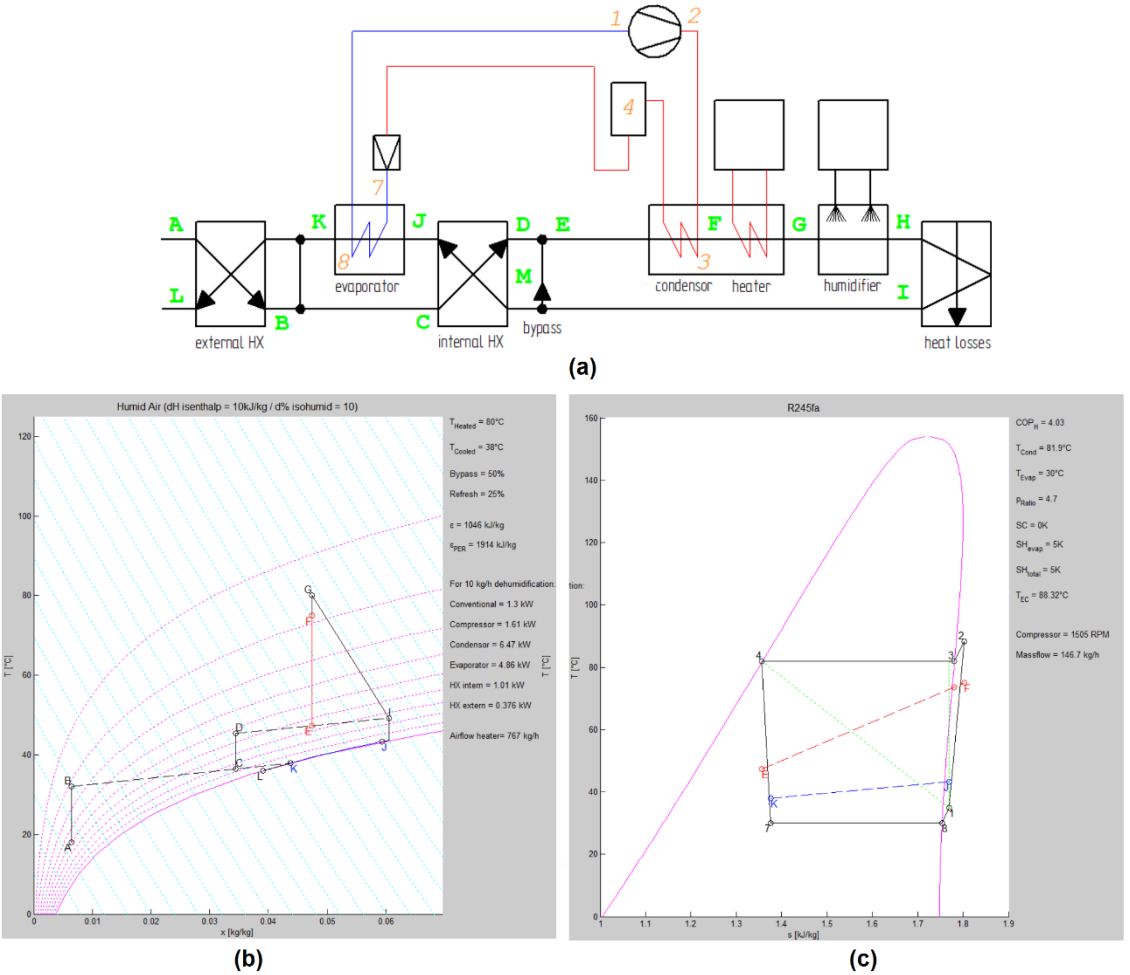


Fig. 8. Heat pump dryer with evaporator bypass, internal/external heat exchangers and recirculation: simplified schematics (a), an example of the simulation output for in psychrometric chart for the drying air (b) and T-s diagram for heat pump cycle (c)

The principles of pinch technology state that as much as possible heat should be recovered by heat exchangers before using a heat pump. By employing an internal heat exchanger, as shown in Fig. 8a, the air is precooled before it goes to the evaporator and air is preheated before it goes to the condenser, decreasing the heat load and therefore the size of the heat pump. But this way, the temperature difference between heat sink and – source is increased and also leads to a decrease in COP. By experiments, it should be investigated if the pinch rules are still to be followed or a deviating energetic optimum can be found. A possible reason for this is a decrease in isentropic efficiency as pressure ratio rises and suction gas density decreases.

The internal HX also has its effectiveness set to 70%. As shown in Fig. 8b, this results in a specific energy consumption ε of 1046 kJ/kg and 1914 kJ/kg if the primary energy ratio is taken into account. Fig. 8c represents the accompanying heat pump cycle.

In this case, the internal HX takes away all the sensible heat from the outgoing air of the dryer. This leads to a much smaller air temperature gradient over the evaporator and limits the ability to obtain superheated fluid at the end of the evaporator. To avoid shifting of the fluid-side pinch point from the beginning to the end of the evaporator, thus decreasing the evaporator pressure, superheat is limited to 5K instead of 10K. When applying this technique to a practical setup, precaution is advised in the form of a liquid separator before the compressor.

3.7. Heat pump equipped with a suction line heat exchanger

The addition of a suction line heat exchanger (SLHX) is presented in Fig. 9a. In this configuration, heat is transferred from the high temperature liquid leaving the condenser to the low temperature refrigerant vapor leaving the evaporator. Thus liquid refrigerant is subcooled before entering the expansion valve while vapor refrigerant is superheated before entering the compressor. Besides energetic benefits, a higher degree of subcooling leads to a smaller amount of flash gas after expansion and improves the evaporator filling grade and heat transfer. Additionally, using a SLHX helps to prevent flash gas formation at the expansion valve inlet and the risk of liquid refrigerant at the compressor inlet [5].

The SLHX effectiveness is set to 70%. Optimum pressure ratio is slightly increased as heat pump performance improves. As shown in Fig. 9b, this results in a specific energy consumption ε of 728 kJ/kg and 1688 kJ/kg if the primary energy ratio is taken into account. Fig. 9c represents the accompanying heat pump cycle.

Depending on the superheat at the outlet of the evaporator, effectiveness of the SLHX and temperature difference between condensing and evaporating side, suction gas superheat might become too elevated, as is the case in Fig. 9c. High superheat causes high compressor discharge temperatures and lower isentropic efficiency. This lower isentropic efficiency is caused due to a decrease of suction gas density, requiring a higher compressor speed despite a lower mass flow. For this reason the SLHX of the test bench is equipped with a 3-way proportional bypass valve allowing to decrease the effectiveness of the SLHX if needed. Desuperheating with fluid injection may be implemented as well.

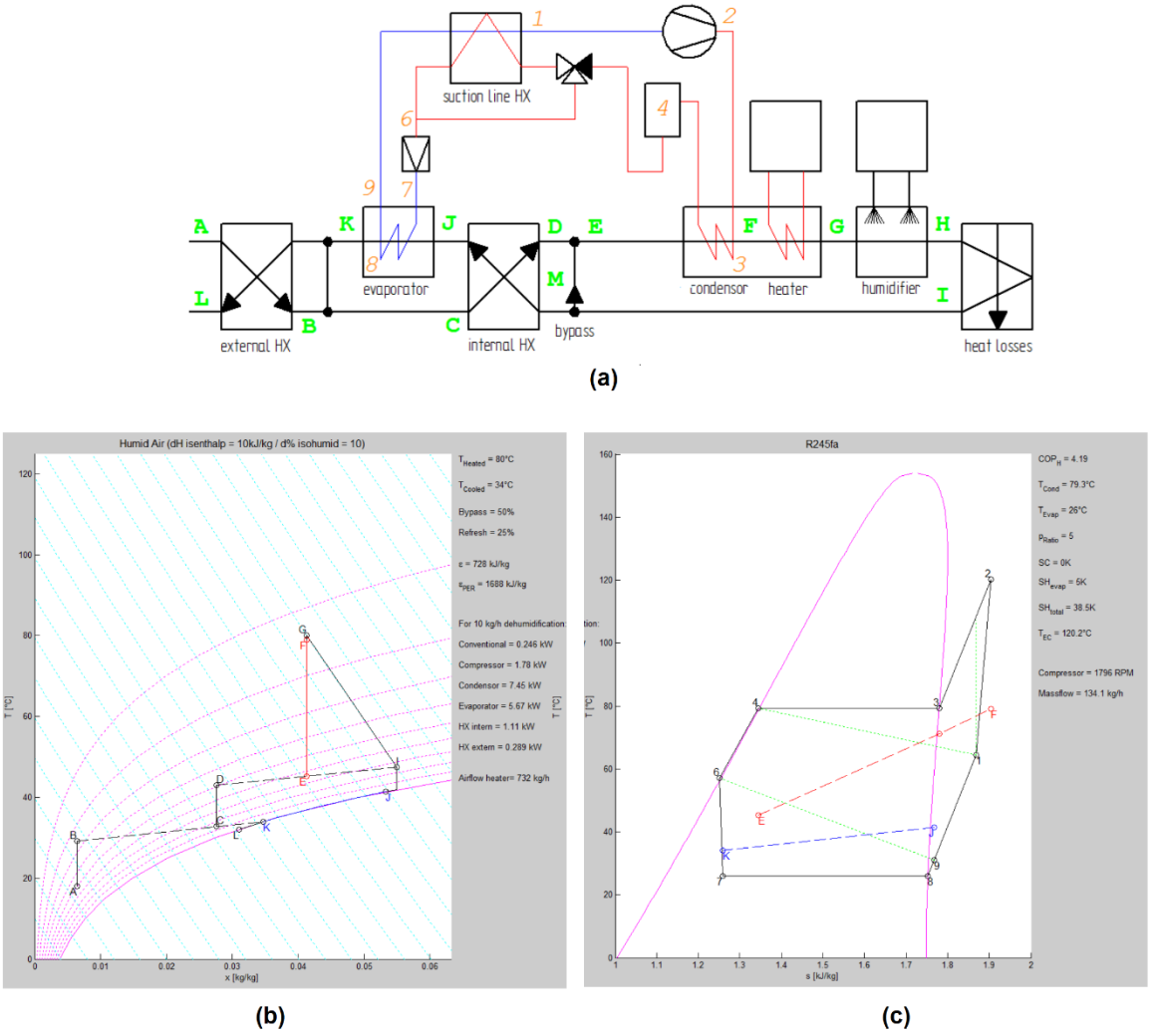


Fig. 9. Heat pump dryer with evaporator bypass, internal/external heat exchangers, recirculation and SLHX: simplified schematics (a), an example of the simulation output in psychrometric chart for the drying air (b) and T-s diagram for heat pump cycle (c)

3.8. Heat pump equipped with a subcooler

Fig. 10a presents the use of a fluid-to-air subcooling heat exchanger. A subcooling heat exchanger offers a substantial improvement to the COP of a single stage heat pump when there is a large heat sink temperature gradient. Having a similar application range as CO_2 heat pumps, simulations suggest subcooling R245fa systems may prove to be an alternative, while still allowing low heat sink temperature gradients and without other usual limitations of a CO_2 system. Most notably the heat source temperature of a CO_2 system shouldn't be too high to prevent supercritical conditions in the evaporator.

The simulated subcooler has a surface area of 35% compared to the condenser, where the assumed heat transfer potential is based solely on the LMTD (log mean temperature difference). Performance with a

crossflow, non-ideal heat exchanger is expected to be worse than the simulation suggests, so a heat exchanger with a surface ratio of 50% was selected for the practical setup. Optimum pressure ratio is slightly increased as heat pump performance improves. As shown in Fig. 10b, this results in a specific energy consumption ε of 637 kJ/kg and 1528 kJ/kg if the primary energy ratio is taken into account. Fig. 10c represents the accompanying heat pump cycle.

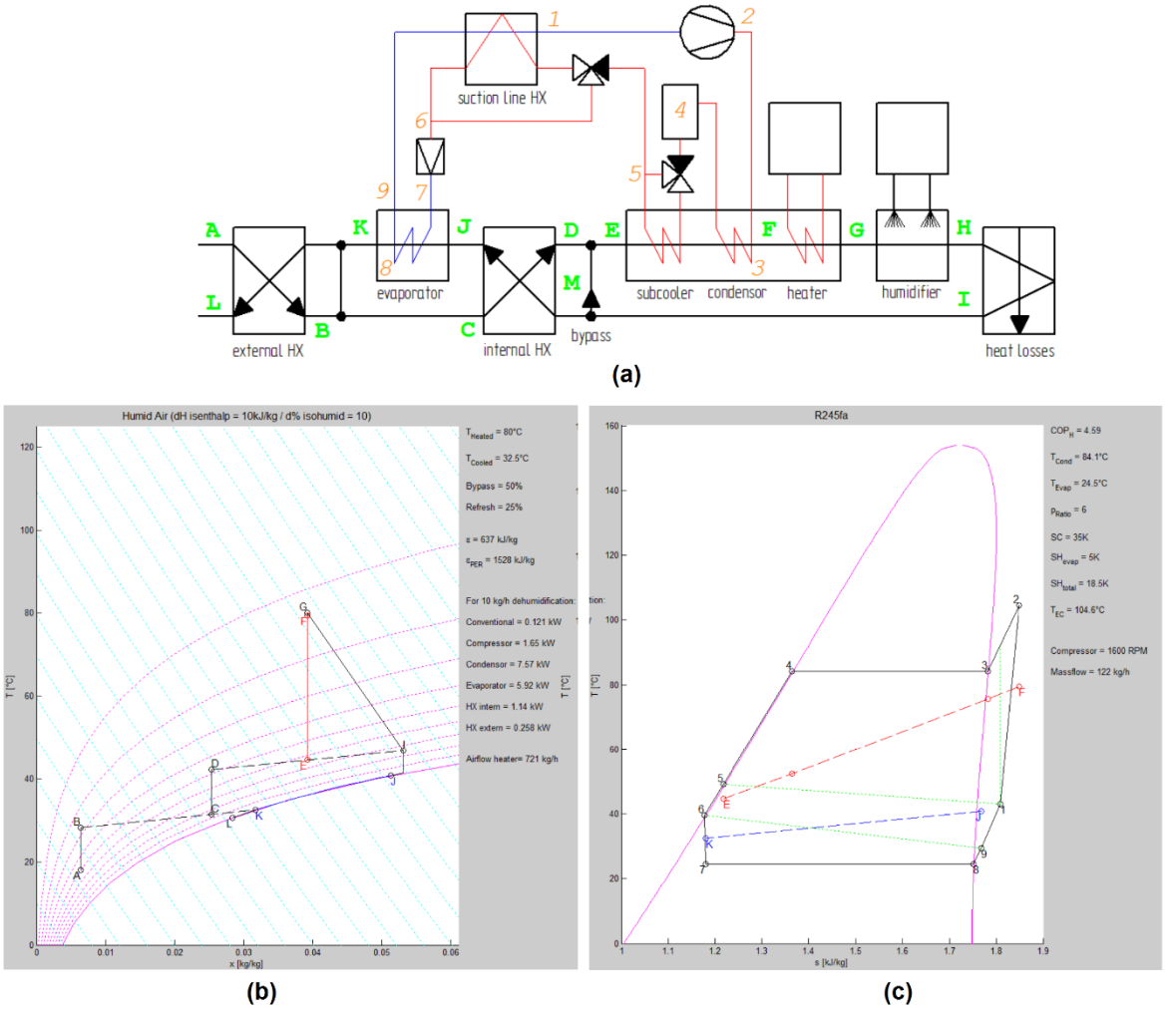


Fig. 10. Heat pump dryer with evaporator bypass, internal/external heat exchangers, recirculation, SLHX and subcooler: simplified schematics (a), an example of simulation output in psychrometric chart for the drying air (b) and T-s diagram for heat pump cycle (c)

4. CONCLUSIONS AND FUTURE WORK

This paper presents the possibilities of a heat pump dryer test bench currently in the final stage of testing and fine tuning, aided by a simulation model. By comparing fig. 1 to 3 the model predicts potential energy savings up to 25,8% just by optimizing a basic drying process but without integration of a heat pump. By adding a well adapted heat pump (fig. 10) the energy input can drop with 85% related to the original one or with 63,6% if the electrical part of it is recalculated to the primary energy input supposing a conversion efficiency of 40%.

Once the model is optimized and validated by tests, it will allow to design energy efficient drying systems, starting from an existing installation or from scratch. Accompanying demonstrations should convince the project owners, starting with the ongoing case locations. These possible concepts also include the use of a gas engine driven heat pump which is favored in regions with a higher electricity to gas price ratio, but is yet to be included in the model.

Simulations reveal R245fa to be an all-round well performing fluid for heat pump dryers, especially for the higher temperature range. Depending on heat sink and – source conditions, significant energetic benefits can be realized with a SLHX and subcooler. By utilizing an internal heat exchanger in the air cycle, the required heat pump capacity decreases. Furthermore, the amount of heat extracted can be controlled not only by the heat pump pressure ratio, but can be complemented by an evaporator bypass. Besides demonstration of optimization possibilities on the drying cycle, the test rig will offer an excellent environment for research on advanced heat pump systems including novel control systems.

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