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PERFORMANCE POTENTIAL OF HEAT PUMP DRYING CYCLES

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Abstract: There are two thermodynamic models of the closed and vented heat pump drying process described in the paper. The models have been used for optimization of process design parameters with respect to drying rate and energy efficiency, respectively. The optimization was performed in points of parametric space defined by refrigerant and air flow rates. Results of the optimization shows the same maximal drying rate performance for closed cycle and both objective functions. There is higher potential of drying time reduction for vented cycle in comparison to closed cycle. Results of the optimization showed strong correlation of the MER parameter with refrigerant flow rate for both closed and vented heat pump drying cycle. There is same maximal MER value obtained for both MER and SMER optimized cycle parameters.

1 Introduction

There are numerous reports which identified the drying operation as very energy intensive and demanding process, Minea [1]. In contrast to drying energy consumption a minimal requirement on drying rate are posed by industry users and end consumers on the process, though.

There are numerous ways how to remove water from a material, to be dried out, which were adopted in various industrial applications. A lot of processes utilize humidification and dehumidification of air for drying – so called hot-air drying.

Number of scientist and researchers have tried to improve drying rate and energy effectiveness of the different drying process. The studies analysed different drying cycles, their component arrangement or effect of different cycle parameters on their performance. Conde [2] pointed out that the heat pump is suitable device for heat recovery in drying process. He further proposed two system with heat recovery heat exchanger as the heat pump was concluded to represent high capital costs. Lambert [3] developed a model for partially vented drying process and concluded that optimal air recirculation ratio should be on level of 75%.

Several measures for energy savings were experimentally analysed by Hekmat [4]. The study reported e.g. 10-18% of energy saving by 67% of air recirculated. The highest, 33% of energy saving was achieved by test with heat pump. Though, the drying time should have been very long. Bansal [5] compared performance of different closed and vented drying cycles employing condensing and heat recovery heat exchangers, based on thermodynamic model. The highest energy savings of 14% was predicted for closed system with condensing heat exchanger and heat recovery compared to basic vented cycle.

Performance of several, pure heat pump dryer configurations, with focus on tropical climate conditions, were studied by Saensabai [6,7]. He reported different best operating configuration with respect to high and low drying rate period and ambient temperature level.

Any drying process, which operates in closed air cycle, is connected to heating and cooling of the process air. HP drying represents very energy efficient way of air dehumidification as it incorporates heating and cooling of air in regenerative way. Among the couple of HPs technologies, the vapor compression process became accessible in wide range of power output scales.

Air dehumidification takes place in evaporator, at low HP temperature level when the air is cooled below its dew point [8,9]. The drying potential of air is recovered by its successive heating in condenser. The heat extracted from air in evaporator can be regarded as heat regenerated from low temperature air for its re-use at high temperature in condenser.

The performance of HP drying process can be influenced by the HP cycle configuration and controlled by selected low and high temperature levels – evaporator and condenser pressures.

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The objective of the study is to examine two HP configuration and their drying potentials in terms of energy consumption and drying time.

2 Thermodynamic model

The two configurations described in the paper are represented by closed (Figure 1) and vented (Figure 2) HP dryer. The drying performance of the HP dryer is investigated numerically making use of thermodynamic model.
The HP dryer is represented by steady state model, which is usually employed to simulate constant drying rate period. The period is important in terms of drying performance as the drying rate attains the highest level during the period. The drying process designer thus usually intents to promote the period within the whole drying cycle.

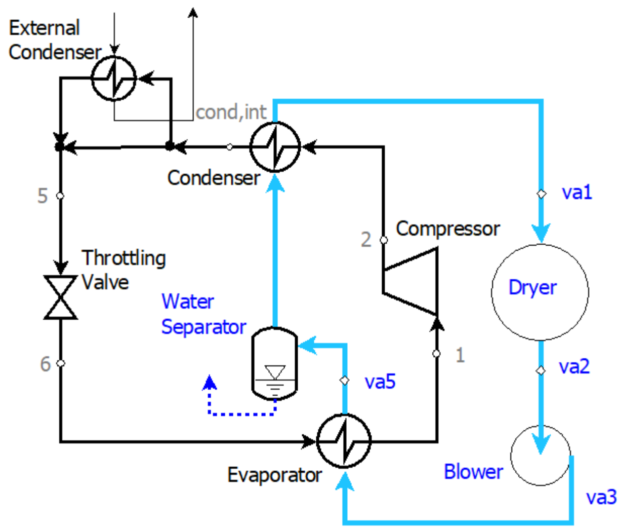

Figure 1 Closed HP dryer components configuration

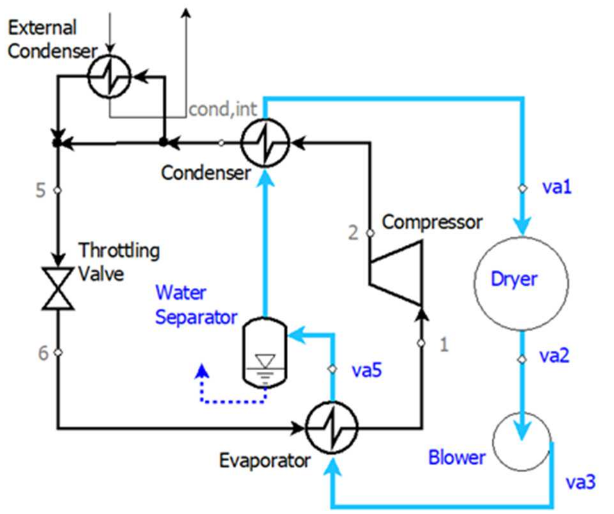

Figure 2 Vented HP dryer components configuration

Following assumptions are summarized as they are applied within the drying model:
• Constant drying rate.
• No pressure drops in refrigerant circuit piping → constant evaporation, condensation pressure.
• Isenthalpic expansion process in throttling device.
• Constant refrigerant superheating in evaporator and subcooling in condenser.
• Negligible heat loss in refrigerant piping and air ducts.
• Constant static pressure in dryer.
• Pinch point temperature difference between refrigerant and air stream in condenser and evaporator.
• Constant power input for auxiliary drives and constant fan efficiency.

The refrigeration cycle is fully defined by evaporation and condensing pressures; p_E, p_C, superheating and subcooling temperature differences; d_{tsh}, d_{tsc} and compressor performance, Figure 3.

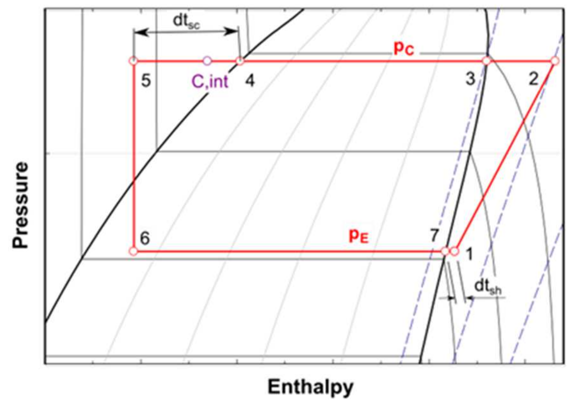

Figure 3 Refrigerant air cycles

2.1 Compressor model

The compressor isentropic and volumetric efficiency model is correlated to measured compressor data according to model described by Carrington [7]. The model is based on six constant polynomials, which relates suction pressure and logarithm of pressure ratio to isentropic and volumetric efficiency parameters.

The model is represented by set of conservation equations for mass of dry air, water/vapor and energy. The equations are constructed for each component in the drying cycle and solved simultaneously, Figure 4.

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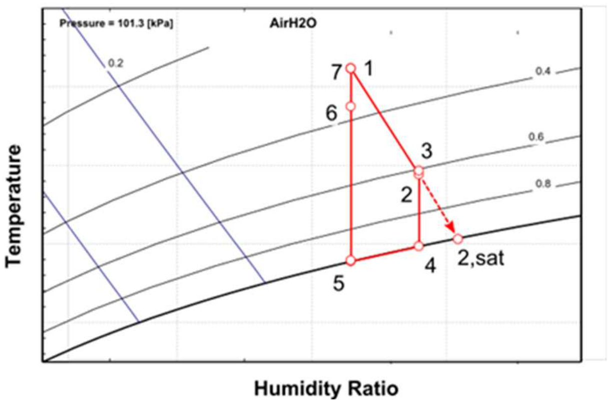

Figure 4 Process air cycles

There is the same mass flow rate in and out of dry air assumed in each of the component. The dry air mass conservation reduces to following equation (1) for each of the component:

$$\dot{m}_{a,in} = \dot{m}_{a,out} = \dot{m}_a \qquad (1)$$

2.2 Dryer model

Conservation of water flow rate in the dryer accounts for water removed from load. The energy accounts for heat loss from the dryer. The constant drying rate assumption is represented by dryer saturation effectiveness parameter, DE.

2.3 Evaporator model

Conservation of water in evaporator takes into account condensed water flow rate from humid air out of the cycle.

2.4 Condenser model

There is no water removed nor added to the air stream in the condenser. The energy equation takes into account energy transfer from refrigerant to drying air.

In contrast to the vented cycle, there is only part of condenser heat transfer from refrigerant to air stream in closed cycle. The part of the heat is referred to as internal condenser; $Q_{(C,int)}$. It is necessary to remove some heat out of the system in order to keep it in thermal equilibrium. There is auxiliary condenser and corresponding heat; $Q_{(C,aux)}$ used in the model for this purpose (Table 1).

Table 1 Summary of closed cycle conservation equations

Component	Quantity	Equations
Dryer	*Water* *Energy*	$\dot{m}_a\omega_1 + \dot{m}_{wD} = \dot{m}_a\omega_2$ $\dot{m}_a h_{va,1} = \dot{m}_a h_{va,2} + Q_{D,loss}$
	Dryer effectiveness	$DE = \omega_2 - \omega_1/\omega_{2,sat} - \omega_1$
Evaporator	*Water* *Energy*	$\dot{m}_a\omega_3 = \dot{m}_a\omega_5 + \dot{m}_{cw}$ $\dot{m}_a h_{va,3} = \dot{m}_a h_{va,5} + Q_E$
Condenser	*Energy*	$\dot{m}_a h_{va,6} + Q_{C,int} = \dot{m}_a h_{va,8}$
Blower	*Energy*	$\dot{m}_a h_{va,2} + P_B = \dot{m}_a h_{va,3}$

Energy balance in the condenser and evaporator is applied assuming minimal approach (pinch) temperature difference; dt_p, between air and refrigerant streams. Example of temperature profile in idealized counterflow condenser and evaporator could be seen in the Figure 5.

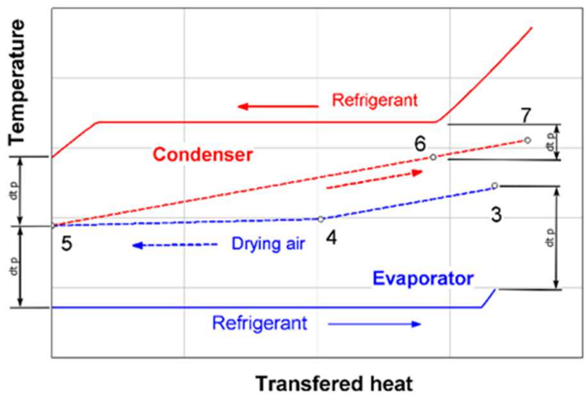

Figure 5 Temperature - duty curve in evaporator and condenser

The two components are treated the same way. There is only energy added to the air stream without affecting water content in dry air. The electric heater is usually not used during the constant drying rate period in order to save energy used for drying.

3 Parametric study with optimized cycle parameters

The performance of the drying cycles is assessed in terms of drying rate and energy consumption for drying of load. Following definitions are adopted for the two parameters: moisture extraction rate (2) and specific moisture extraction rate (3).

$$MER = \dot{m}_{wD} \qquad (2)$$

$$SMER = \frac{\dot{m}_{wD}}{P_{tot}} \qquad (3)$$

Parametric studies have been performed for both of the HP dryer cycle configurations. The refrigerant and air mass flow rate have been selected as independent parameters for closed cycle. As the refrigerant and air mass flow rates are dependent for the vented cycle, the refrigerant mass flow rate has been used as the only independent parameter for the cycle.

A genetic algorithm approach has been used to optimize design parameters within each discrete point of parametric space. The MER and SMER parameter have been used as respective objective functions for the optimization. Optimization parameter settings with additional constraints are summarized in the Table 2 for both cycle configurations.

Table 2 Optimization parameters selection

Parameters				
Cycle	Independent	Design	Constant	Constraints
Closed	m_r, m_a	p_E, dt_{sc}	p_C, dt_{sh}, DE	$dt_p \geq dt_{p,lim}$
Vented	m_r	p_E, p_C, dt_{sc}	dt_{sh}, DE	$V_a \leq V_{a,lim}$

The optimization of the design parameters for both cycles is subjected to constrained value of a minimal pinch point temperature difference and a maximal volumetric air flow rate.

4 Results and discussion

4.1 *Closed cycle*

Performance of MER and SMER optimized closed cycle in parametric space is shown on contour plot in the Figure 6 and in the Figure 7. The MER and SMER performance is normalized to a reference value and shown in normalize coordinate space of parametric variables with contour plot of optimal evaporator pressure.

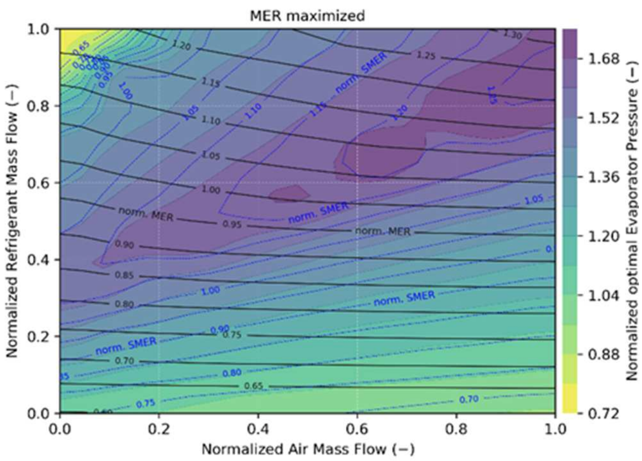

Figure 6 Optimal performance of close cycle for MER objective function

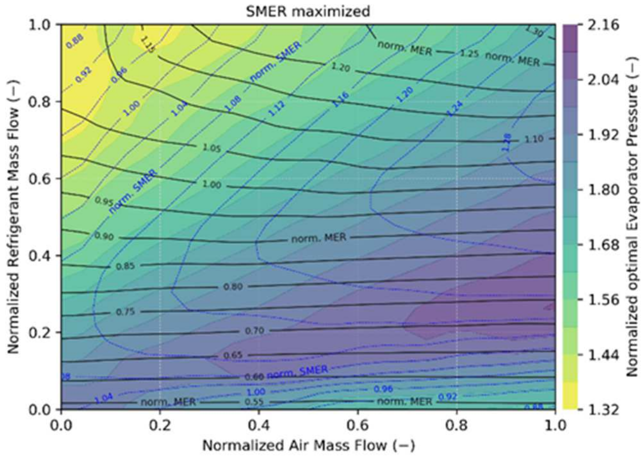

Figure 7 Optimal performance of close cycle for SMER objective function

The results show that there is higher optimal evaporator pressure computed for SMER maximized cycle in comparison to MER optimization. Interestingly, the optimal evaporator pressure is not the lowest possible value from the optimization range. The fact could be explained by increased specific evaporation heat exchanged with increased evaporator pressure.

In both cases there is strong variation of MER parameter with refrigerant mass flow and relatively low dependence on air mass flow rate. Contrary to this, there is stronger dependence of SMER parameter on air mass flow rate, which attain local maximum for a particular value air flow rate.

The contour lines of SMER parameter strongly correlates with optimal value of evaporator pressure for MER optimized cycle. There is no such correlation in case of SMER optimized results.

Both, maximal value of MER parameter and it location in parametric space are almost identical for MER and SMER maximized cycles. The region of the maximal MER value is located in high-high point of refrigerant and air mass flow rate parametric space. Contour plots of SMER parameter show, that there is no more than 2% compromised maximal SMER value for MER maximized cycle. The locus of maximal SMER values, for particular air flow rate, is slightly shifted towards lower refrigerant mass flow rate for SMER maximized cycle.

4.2 *Vented cycle*

Results of the vented cycle optimization are shown in the Figure 8 for MER and in the Figure 9 for SMER objective functions. There is almost linear variation of MER parameter on refrigerant flow rate in most of its range for both MER and SMER maximization. The maximal value of MER parameter achieved is about 6% higher for MER maximization compare to SMER maximization.

The natural selection of operating point for MER maximized cycle is in higher refrigerant flow rate, as both MER and SMER parameter attain its maximal value in the range. Contrary to this, either MER or SMER performance parameter is compromised in case of selection of operating point for vented cycle according to the SMER maximized results. Indeed, the two parameters attain its maximal value on different parts or refrigerant flow range. The maximal value of SMER for SMER maximized configuration is about 22% higher and obtained for refrigerant flow rate in its lower range, in comparison to MER maximized configuration.

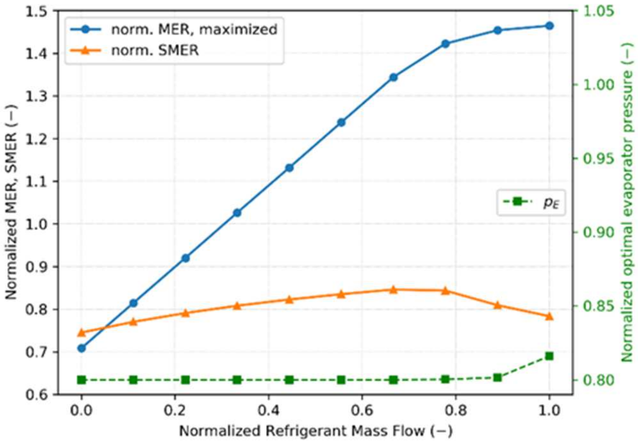

Figure 8 Optimal performance of vented cycle for MER objective function

The flat variation of optimal evaporator pressure for MER maximization is due to restriction by its lower value range and due to maximal air flow rate constraint in upper refrigerant flow rate. The maximal air flow rate constraint applies for all points starting from maximal SMER value through the upper range of refrigerant flow rate for SMER maximized cycle.

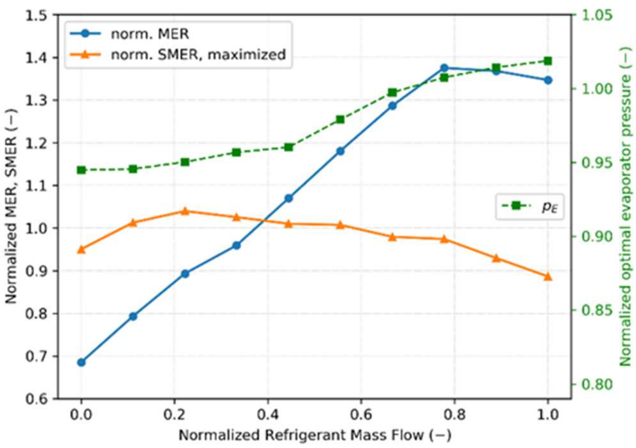

Figure 9 Optimal performance of vented cycle for SMER objective function

5 Conclusions

A thermodynamic model of closed and vented heat pump drying cycle have been described. Optimization results for drying performance parameters; drying rate - MER and energy efficiency - SMER parameters for both cycles have been analysed in the paper.

Results of the optimization showed strong correlation of the MER parameter with refrigerant flow rate for both closed and vented heat pump drying cycle. There is same maximal MER value obtained for both MER and SMER optimized cycle parameters. The difference in optimal SMER value accounts only for 2% difference between SMER and MER optimized cycles.

It has been shown that refrigerant and air flow rate are dependent parameters for vented heat pump drying cycle. Optimized parameters of vented cycle showed better drying performance compared to closed cycle. There is computed improvement of more than 10% and 4% for MER and SMER parameters, respectively.

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