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Modelica-based modelling of heat pump-assisted apple drying for varied drying temperatures and bypass ratios

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ABSTRACT

Drying is an energy and time intensive process in which thermal energy demand is mostly provided by fossil resources. It is important to increase the energy efficiency of drying processes especially in the food processing industry in terms of organic products and sustainability. The potential for using a heat pump with CO_2 as a working media to provide heating and cooling in the drying cabinet was investigated for typical food drying temperatures of 50–70 °C and for various ratios of moist air being bypassed.

A dynamic heat pump-assisted dryer model was developed and experimentally validated. The model was created with respect to heat transfer, pressure loss and flow requirements. The simulation results show that a closed loop heat pump-assisted drying process reduces the energy demand by up to 84% compared to open loop drying processes with fossil resources as the energy source. The specific moisture extraction rate for a heat pump dryer is up to four times higher than that of an open loop dryer. However, in the heat pump dryer case the drying time is increased by up to 69%.

1. Introduction

Drying is a thermal dehydration method which is one of the most frequently used preservation methods for food. In consideration of increasing fossil fuel depletion and environmental pollution, the process of drying needs to be reevaluated: The main energy sources are still fossil fuels, and the depletion of fossil fuels harbors a future risk for energy shortages. Furthermore, fossil fuel consumption is a large cause for environmental pollution. It is therefore necessary to develop new drying systems that are independent of fossil fuels and at the same time offer increased efficiency of both the drying process and the overall system.

The most commonly used drying technology is convective drying [1]. The water content of the product is reduced by a flow of hot air as drying agent, resulting in an extended shelf life compared to fresh products.

The process of drying is widely used in all kind of industries and is a highly energy-intensive process. It is estimated that around 10–20% of the total energy used in all industries in developed countries is used by drying processes, whereas the major amounts are demanded by the food and paper industries [2,3]. Due to its high energy demand, the process of drying is a very important field of research in terms of energy efficiency but also for product quality. Especially in the food processing sector, which generally has high quality requirements for dried products, the thermal efficiency is often relatively low in the range of 25–50% [4].

Typically, batch dryers are used for small and medium production runs and for relatively thin products like fruits or vegetables [5]. Food is loaded onto trays in a cabinet and left until the drying process is complete. Due to the simple design, cabinet dryers usually have limited turnover rates and the drying process is not uniform throughout the drying space.

The basic advantages of heat pump driven convective drying results from the ability to recover energy from the already used drying air and reuse it in the process. This results in high values for the specific moisture extraction ratio (SMER), often between 1 and 4 kg of evaporated water per 1 kWh, since heat is being recovered from the moist air [6]. This can result in a drying efficiency of up to 95% compared to 30–40% for other hot air drying methods [7]. Industrial heat pumps with a heat sink of up to 100 °C using natural refrigerants like R717, R718 or R744 are already conventionally established. This makes heat pumps very well suited for food drying, which is usually performed at temperature levels of up to 70 °C. The natural refrigerant R744 is an environmentally friendly alternative to common heat pump refrigerants. It offers a global warming potential of 1 and it is not flammable. No restrictions regarding the utilization are existent or planned.

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Nomenclature

A	Heat transfer coefficient W m^{-2} K^{-1}
Β	Mass transfer coefficient m s^{-1}
Γ	Parameter for activity coefficient estimation -
Δ	Parameter for activity coefficient estimation -
Ε	Parameter for activity coefficient estimation -
Λ	Thermal conductivity W K^{-1} m^{-1}
Ρ	Density kg m^{-3}
Φ	Moisture content -
A	Effective product surface area m^2
act	Activity coefficient -
c_p	Specific heat capacity J kg^{-1} K^{-1}
D	Diameter of apple slice m
Le	Lewis-factor -
m	Mass kg
$\dot{m}$	Mass flow rate kg s^{-1}
P	Pressure bar
Pr	Prandtl-number -
Re	Reynolds-number -
SMER	Specific moisture evaporation rate kg_{water} kWh^{-1}

Subscripts

d.b.	Dry basis
Evap	Evaporated water
Lam	Laminar flow conditions
Surf	Related to product surface
Turb	Turbulent flow conditions

R744 as refrigerant in heat pump dryer systems for the food industry is evaluated to give better performance than classical refrigerants like R134a, but with higher irreversibility of the expansion device [8–10]. The system performance of a R744 heat pump drying system yields large potential for utilization in different industries and the technology is generally considered as mature [11].

However, its industrial application has still not been widespread. Heat pump-assisted drying systems are considered more expensive in both investment and running costs. Commercial viability is impacted by uncertain heat pump reliability for potential users and a lack of experimental and demonstration installations for different industrial applications. Dryers that are enhanced with heat pumps become complex systems due to the close interdependency of their components and thermodynamic parameters. There is a need for further R&D and demonstration activities for optimal integration of dryers and heat pumps, aiming at developing heat pump dryers that are both cost-effective and reliable.

The present study assesses design parameters of heat pump-assisted dryers by modelling the heat pump cycles and their respective drying processes. A model for the drying cabinet was developed and investigated for the thermal drying performance of apples.

2. Model description

2.1. System setup

The present study investigates a closed loop drying process that has been enhanced with a heat pump as shown in the scheme in Fig. 1, using R744 as refrigerant. A model for this convective food drier was developed and the model validation was conducted with measurement data from an actual drying system in the lab. The energy efficiency of the heat pump driven dryer was then compared with conventional methods. To evaluate the potential increase of energy efficiency, an open loop drying process with a fossil fuel-based burner serves as a benchmark.

The heat pump-assisted dryer consists basically of two cycles, as portrayed in Fig. 1. The CO_2-cycle is used to first heat up the air flow into the drying chamber and afterwards to dehumidify the moist air leaving the drying chamber while reusing the waste heat.

The second system cycle comprises of the air cycle, which interacts directly with the drying product. Hot air leaving the gas cooler is led to the drying chamber via a fan, where it heats up the product and receives evaporated water. The moist air releases its sensible and latent heat to the CO_2 in the evaporator of the heat pump cycle.

2.2. Drying cabinet model

The drying cabinet model has been implemented in Modelica using already existing model libraries, TIL and TIL Media [12]. TIL is a model library for thermal components and systems, while TIL Media contains a model library including thermophysical properties for the utilized fluids.

It was decided not to base the model on the approach of semi-theoretical drying equations or empirical thin layer drying models [13–15], since the effect of the air humidity on the drying progress needs to be considered when air is being bypassed within the closed-loop air cycle. The focus of the study was not to model as accurate drying curves as possible, but to evaluate the potential of heat pump assisted drying with bypass. The newly derived cabinet model is based on the film theory for mass transfer and consists of a volume cell representing the drying cabinet, where the drying product is exposed to the moist air. The model focusses on the heat and mass transfer between the incoming moist air, the product that is to be heated and dried, and the outgoing moist air.

The drying cabinet model is connected to the rest of the system via two gas ports for moist air supply and exhaust. Heat and mass transfer quantities are calculated with the help of corresponding coefficients: The heat transfer coefficient α is estimated based on convective heat transfer fundamentals for overflowed disks [16] according to the following set of equations:

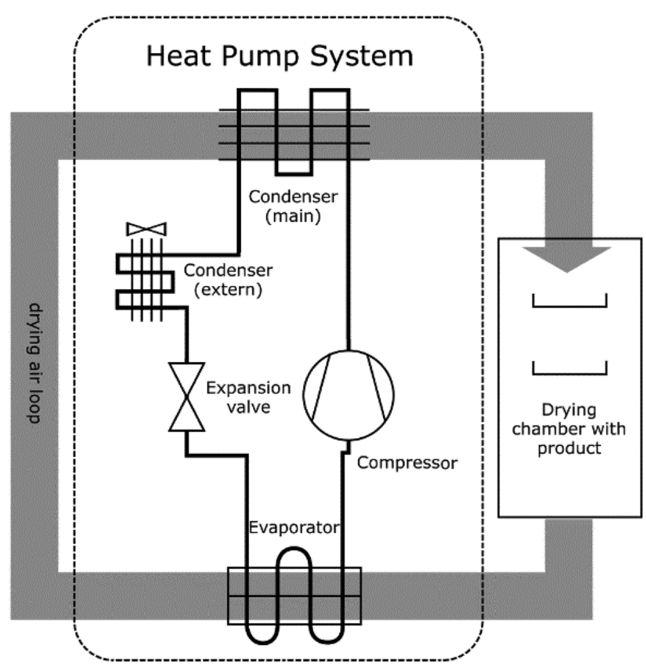

Fig. 1. Scheme of a heat pump driven convective drying process.

$$Nu_{m,turb} = \frac{0,037Re^{0,8}Pr}{1 + 2,443Re^{-0,1}(Pr^{2/3} - 1)} \quad (1)$$

$$Nu_{m,lam} = \frac{\sqrt{\pi}}{2}Re^{1/2} * \frac{Pr^{1/2}}{(1 + 2,55Pr^{1/4} + 48,66Pr)^{1/6}} \quad (2)$$

Equations (1) and (2) utilize the dimensionless flow quantities of the Reynolds number Re and Prandtl number Pr. Since flow conditions in the range between laminar and turbulent flow are at hand, the Nusselt number is averaged by Eq. (3).

$$Nu_m = \sqrt{Nu_{m,lam}^2 + Nu_{m,turb}^2} \quad (3)$$

Finally, the heat transfer coefficient α is calculated by Eq. (4).

$$\alpha = \frac{Nu_m * \lambda}{d} \quad (4)$$

The mass transfer coefficient β is based on essentially the same principles, combined with the film theory for mass transfer, taken from [16].

$$\beta = \frac{\alpha * Le}{\rho * c_p} \quad (5)$$

Equation (5) considers the empiric Lewis-factor [17]. Properties of the drying product need to be provided as input parameters. Property data such as the heat capacity for liquid water are provided by a temperature- and pressure-dependent object class. For moist air, however, two of these object classes are used to consider the special conditions in the drying cabinet: One pressure- and enthalpy-dependent moist air object that represents the incoming dry air; and one pressure- and temperature-dependent air object that estimates the property data of the moist air directly at the surface of the drying product. For this second air object, it is assumed that the air on the product surface features the same temperature as the product and is considered as saturated air with a relative humidity of 99%.

By combining the quantities outlined above, the mass flow rate for the water that is being removed of the product over time can be calculated based on film theory fundamentals [16] for mass transfer:

$$\dot{m}_{water,evap} = \beta A \frac{\rho_{air,surf}}{p_{air,surf}}(act * p_{i,airsurf} - p_{i,air}) \quad (6)$$

In Eq. (6), the parameter A is the effective surface area of the dried product while p_i is the water vapor partial pressure in the air. Values labelled with *air*, *surf* are considering moist air directly at the product surface. The activity coefficient *act* in Eq. (6) is needed to represent the characteristic progress of the drying [18].

$$act = 1 - \frac{\varepsilon * \varphi_{product} + \delta}{1 + \delta^{(\gamma * \varphi_{product})}} \quad (7)$$

The activity coefficient considers the decelerated mass transfer at the end of a drying process when the capillary forces are limiting the water release from the product [19]. The drying process of apples is characterized by a very high initial moisture content of about 85%. Therefore, another slow-down factor is the reduced heat and mass transfer area between the product and the moist air, since an increased shrinkage of apple slices occurs with proceeding drying progress. The activity factor is characteristic for each type of product. It depends on the residual moisture content of the product and the three parameters δ, γ and ε, which need to be estimated from measurements [18].

Finally, the energy and mass balance for the drying cabinet is augmented by Eq. (8), which considers the drying progress.

$$\frac{dm_{product}}{dt} = -\dot{m}_{water,evap} \quad (8)$$

The modelling of the dryer cabinet was conducted in an object-oriented way and can be universally implemented in both open and closed loop drying system models. The drying cabinet model features the potential to theoretically investigate various types of drying products. As for apple drying, the activity coefficient *act* was determined with measurement data from the SINTEF Energy test rig. Further measurement data test series were conducted to validate the model.

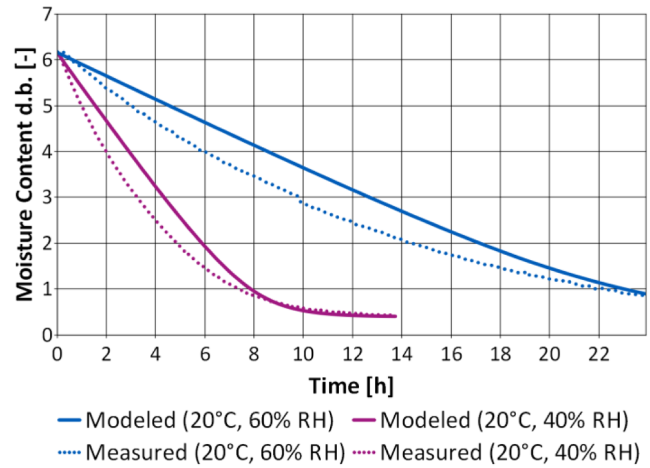

Fig. 2. Drying of initially 0.5 kg of apples: Measurement SINTEF Energy vs simulation results.

2.3. Model validation

The model for the drying cabinet is validated against measurement data from an apple drying test rig at the laboratories of SINTEF Energy (Fig. 2). This test rig consists of a closed loop air cycle with an electrical heater to heat up the air and a CO_2 heat pump to cool it down again. To further increase the general quality of validation, an additional measurement data set was used from Innotech GmbH (Fig. 3), a German manufacturer for drying systems. In this case, an open loop system with a fossil fuel-based burner was applied.

The validation is therefore conducted with various types of measurement data sets, which differ in general scaling and drying conditions for the drying processes. The model was provided with measured input parameters for the inlet relative humidity, the air velocity within the drying cabinet and the inlet temperature of the moist air.

Figs. 2 and 3 compare the modelled and measured drying curves for various conditions for the initial apple mass, the drying temperature and the air humidity. During the time period of constant drying rate, a maximum deviation of up to 28% is existent between the model and the measurement. However, since the overall trend of the drying curve and

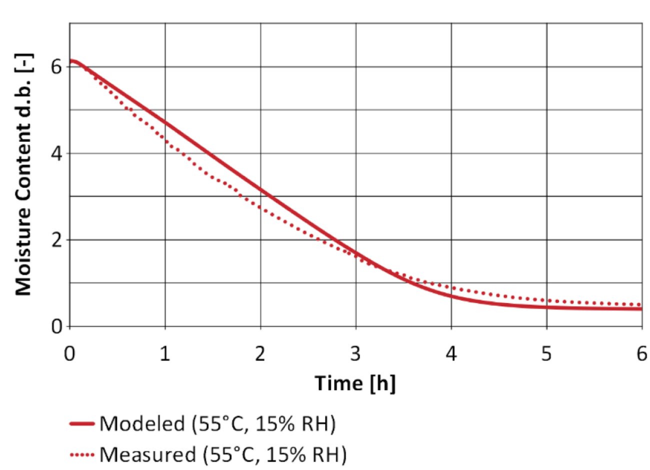

Fig. 3. Drying of initially 3 kg of apples: Measurement Innotech GmbH vs simulation results.

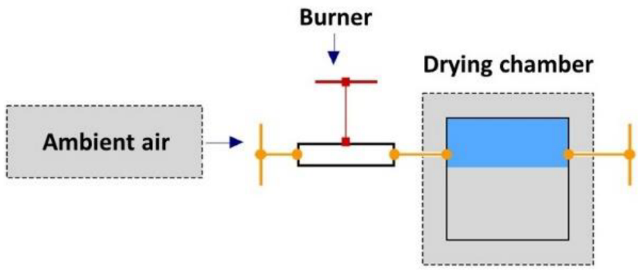

Fig. 4. Model layout for the open loop benchmark setup.

the end time in particular are forecasted in a correct way (especially for varied air humidity values), the validation is nevertheless considered to be successful. It shall be noted that more accurate validation results would have been achieved if semi-theoretical approaches with fitted coefficients had been chosen. However, these tend to neglect the effect of the air humidity on the drying process and were therefore not taken into consideration.

Therefore, the chosen modelling approach reproduces the system behaviour for all measurement data sets in a satisfying way and can be thus used to investigate the drying behaviour for heat pump-assisted dryers with varied bypass ratios.

2.4. System model

The final aim of this study is to utilize the validated drying cabinet model to evaluate heat pump-assisted drying systems. In Section 3, two drying system setups will be investigated to be able to make more precise statements on the energy efficiency of a drying process. An open loop system without heat pumps is the simplest system for a drying process and will therefore serve as a benchmark. Ambient air is heated up via a fossil fuel burner to the desired drying temperature at the drying cabinet inlet and then released back into the environment after passing the cabinet, as depicted in Fig. 4. In this case, the general requirements for the system regarding specific components or control schemes are very low.

The first step to increase the energetic efficiency of any drying process is to apply a closed loop system on the air side to reduce energy losses to the ambience. The required heating effect to provide a specific drying temperature can be drastically decreased by direct re-utilization of the hot and most of the time still relatively dry air from the drying cabinet outlet. However, the humidity within the closed loop system would gradually increase with advancing drying progress. Therefore, the air needs to be cooled down to reduce the water content in the air cycle via condensation.

A heat pump with CO_2 (R744) as refrigerant is used to cool and heat the moist air in the closed loop cycle, as shown in Fig. 5. To drain the absorbed water, the moist air needs to be conditioned to a lower temperature. The air is cooled down in the evaporator to enforce condensation and remove moisture. Afterwards, it is reheated again by the gas cooler.

Since drying is an isenthalpic process and since both the air cycle and the CO_2-cycle are closed looped, a second gas cooler is required as a heat sink in the CO_2-cycle to dispose the extra heat that is being introduced to the system by the CO_2-compressor. Furthermore, the second gas cooler is also beneficial to cool down the refrigerant sufficiently to constantly ensure a crossing of the critical point for CO_2 before the following expansion in the expansion valve.

A bypass is added within the air cycle to recirculate moist air from the dryer outlet directly to the gas cooler inlet. By reducing the volume flow in the evaporator and by increasing the air-side temperature in the gas cooler, less energy consumption is achieved due to a reduced compressor load. In the present study, direct recirculation ratios of 20 and 80% of the overall moist air volume flow are examined. In the existent model, the percentage of recirculation is adjusted by choosing appropriate pressure drop values for the tubes in both the bypass and the evaporator sub-circuit. It should be noted that the investigated bypass systems are not optimally controlled in this study, leading to a constant recirculation of moist air and thus to a non-optimal utilization of the bypass.

As an overall control strategy, the moist air temperature in the gas cooler and evaporator is controlled via the high and low pressure levels of the heat pump.: The speed of the CO_2 compressor is used to control the gas cooler outlet temperature for the moist air. In this way, the inlet temperature for the drying cabinet can be set to 50, 60 or 70 °C for each of the three investigated system configurations. On the other hand, the throttling area of the expansion valve is being controlled in a way to regulate the evaporator outlet temperature for the moist air. To improve the comparability between the closed loop setups for varied drying temperatures, the control of the air temperature at the evaporator outlet is always set to be 40 K below the drying cabinet setpoint temperature. It shall be furthermore noted once again that for the

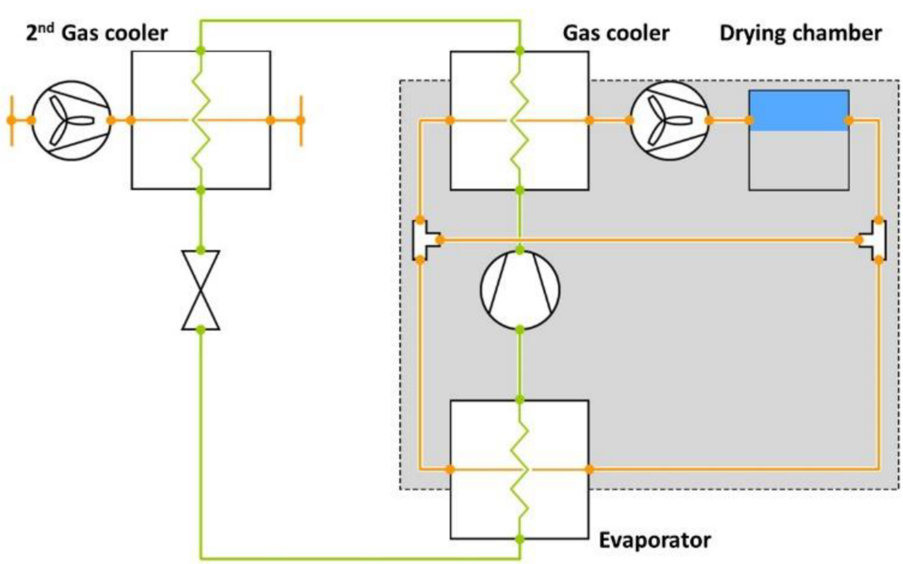

Fig. 5. Model layout for the heat pump-assisted drying setup (R744 sub-cycle with green lines, moist air sub-cycle with orange lines). (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

open loop system the moist air is just being released to the environment without further treatment.

3. Simulation results and discussion

The drying performance of a benchmark open loop system is compared with heat pump-assisted systems that apply closed loop air cycles. The evaluation is conducted for varied drying air temperatures and bypass ratios. For both the open and closed loop system a moist air volume flow rate of 5400 m^3/h is considered to dry initial 100 kg of apples. For the simulation result Figs. 6–9, the following system names will be used to distinguish the various setups:

- *Open loop*: An open loop system without heat pumps serves as a benchmark, as depicted in Fig. 4.
- *20% bypassed*: In a closed loop air cycle, a CO_2 heat pump is used to heat and cool the moist air, as shown in Fig. 5. A bypass is added to provide an air recirculation ratio of 20%.

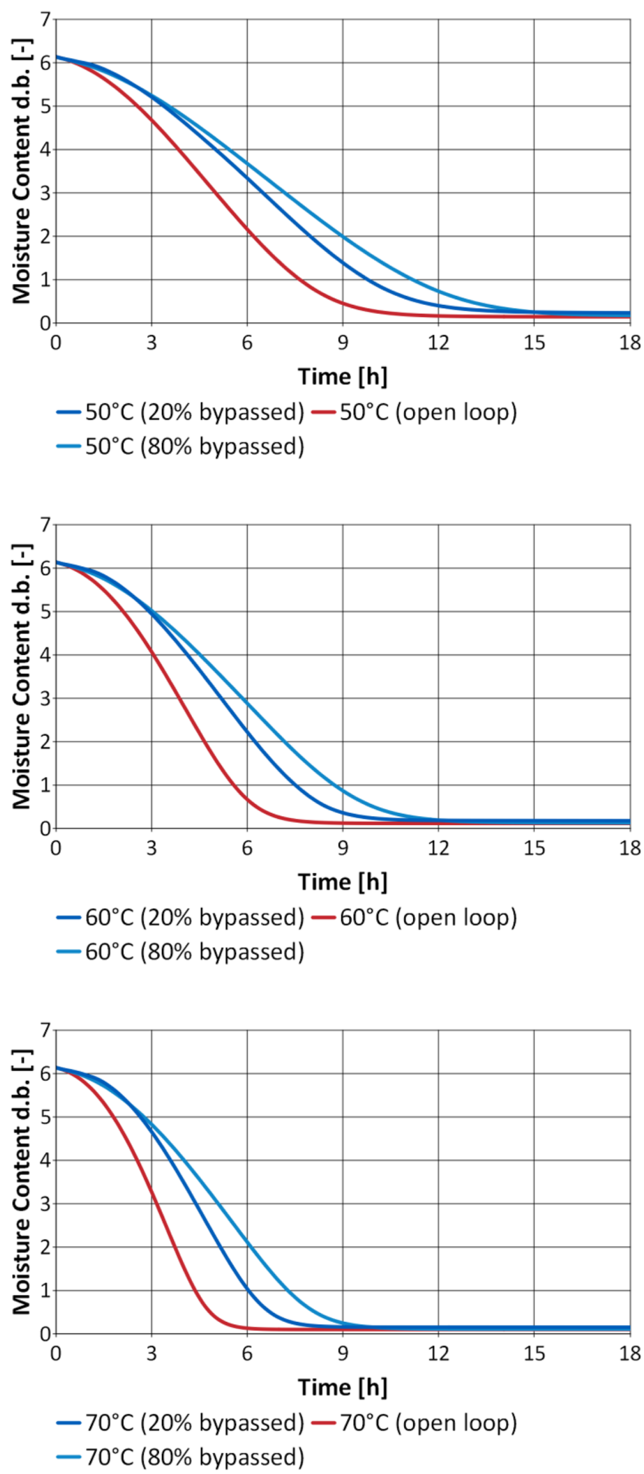

Fig. 6. Simulation results for the drying process of 100 kg of apples for varied drying temperatures and bypass ratios.

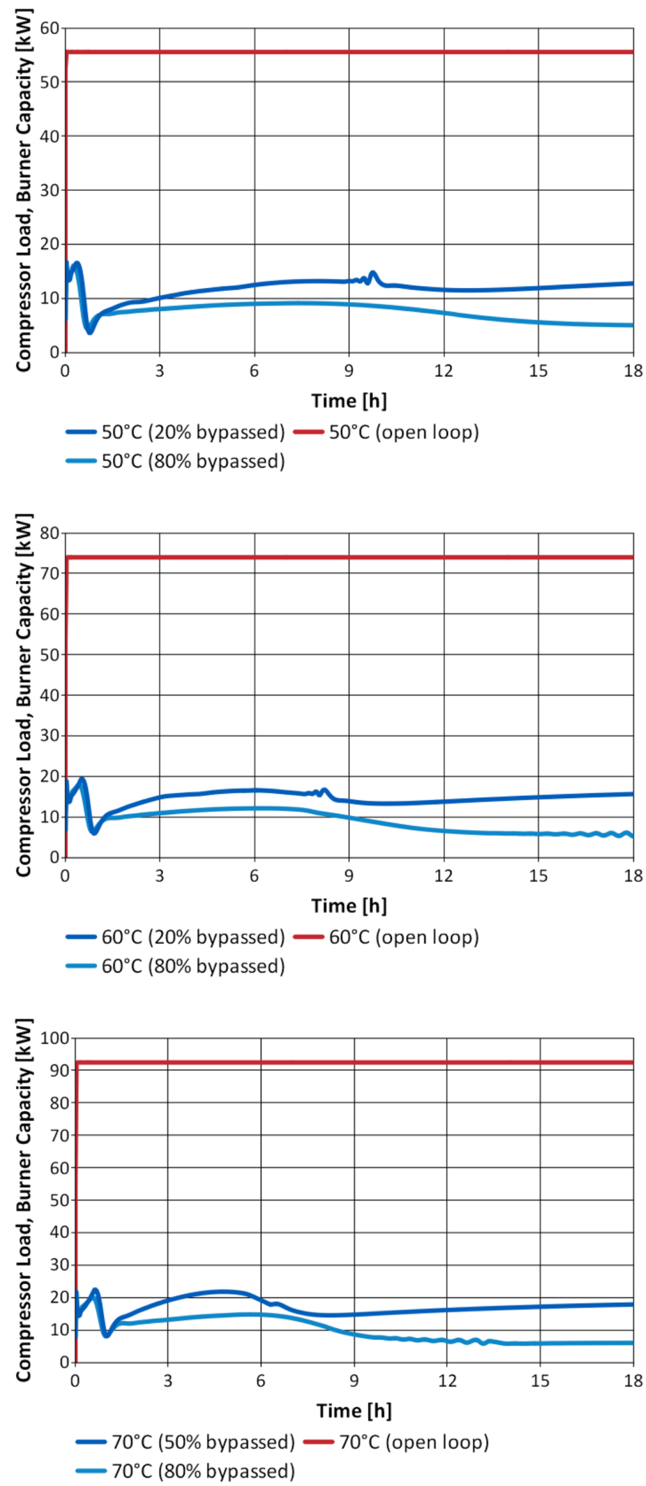

Fig. 7. Simulation results for the compressor load [kW] of the closed loop heat pump-assisted dryer and for the burner capacity [kW] of the open loop dryer.

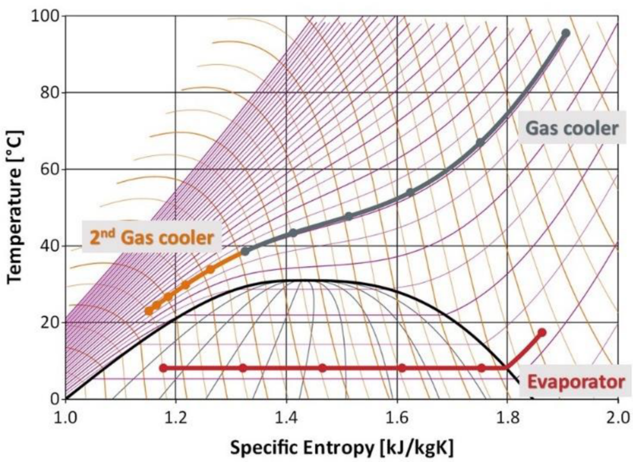

Fig. 8. Temperature-entropy-diagram for a R744 heat pump assisted drier system for a drying temperature of 50 °C.

- *80% bypassed*: In a closed loop air cycle, a CO_2 heat pump is used to heat and cool the moist air, as shown in Fig. 5. A bypass is added to provide an air recirculation ratio of 80%.

Fig. 6 illustrates the simulated drying curves for the investigated setups. For the sake of improved comparison, all graphs are depicted for 18 h. The drying temperature was always controlled to be constant for both the benchmark open loop and the heat pump closed loop dryer. For a drying temperature of 50 °C, the apple drying took 9.1 h for the benchmark dryer and 11.7 h and 13.3 h for the heat pump-assisted dryer with 20% and 80% bypassed air, respectively. Therefore, an increase of 29% and 46% in drying time is observed, when heat pump-assisted drying is utilized instead of an open loop setup. For the 60 °C case, the drying time amounts to 6.4 h for open loop, 8.7 h for 20% bypassed heat pump dryer and 10.2 h for the 80% bypass system, resulting in an increased drying time by 36% and 59%. For 70 °C, 4.9 h are needed in the open loop case, while 6.8 h and 8.3 h are the respective times for the heat pump dryers, leading to a drying time increase by 39% and 69%.

The required drying time is shortest for the open loop benchmark system, as fresh relatively dry air from the environment is heated up and fed to the drying cabinet. Hence, dry air is continuously provided without dealing with the increased moisture content of the air afterwards as it is simply ejected to the ambience. The heat pump dryer reutilizes the moist air and the air circulation within a closed loop setup leads to a slowed down drying process as the air humidity increases with the drying progress. The advantage of the heat pump dryer lies however in its increased energy efficiency, as shown in Figs. 7 and 9.

Fig. 7 compares the required energy input for the systems. For the open loop system, the burner capacity is the only energy input, while for the heat pump system-assisted closed loop systems the compressor load is taken into account for energetic evaluation. To achieve a constant drying temperature of 50 °C, the open loop system requires 55.7 kW, while the heat pump systems use up to 13.3 and 8.9 kW (for 20% and 80% bypass ratio, respectively). For 60 °C, 74.3 kW are needed for the open loop case, while the heat pump systems demand up to 16.6 and 12.3 kW. The 70 °C drying process requires 92.8 kW (open loop) and up to 21.8 and 14.8 kW (heat pump-assisted, 20% and 80% bypassed) of capacity input.

Therefore, the utilization of a heat pump dryer with a 20% bypass setup results in reduced energy demand by approximately 77%, while the consideration of an 80% bypass yields a decrease in energy demand by 84%. The energetic savings were roughly the same for all three investigated drying temperature cases. This is due to a COP decrease with increasing drying temperature. If the COP for the heat pump system is

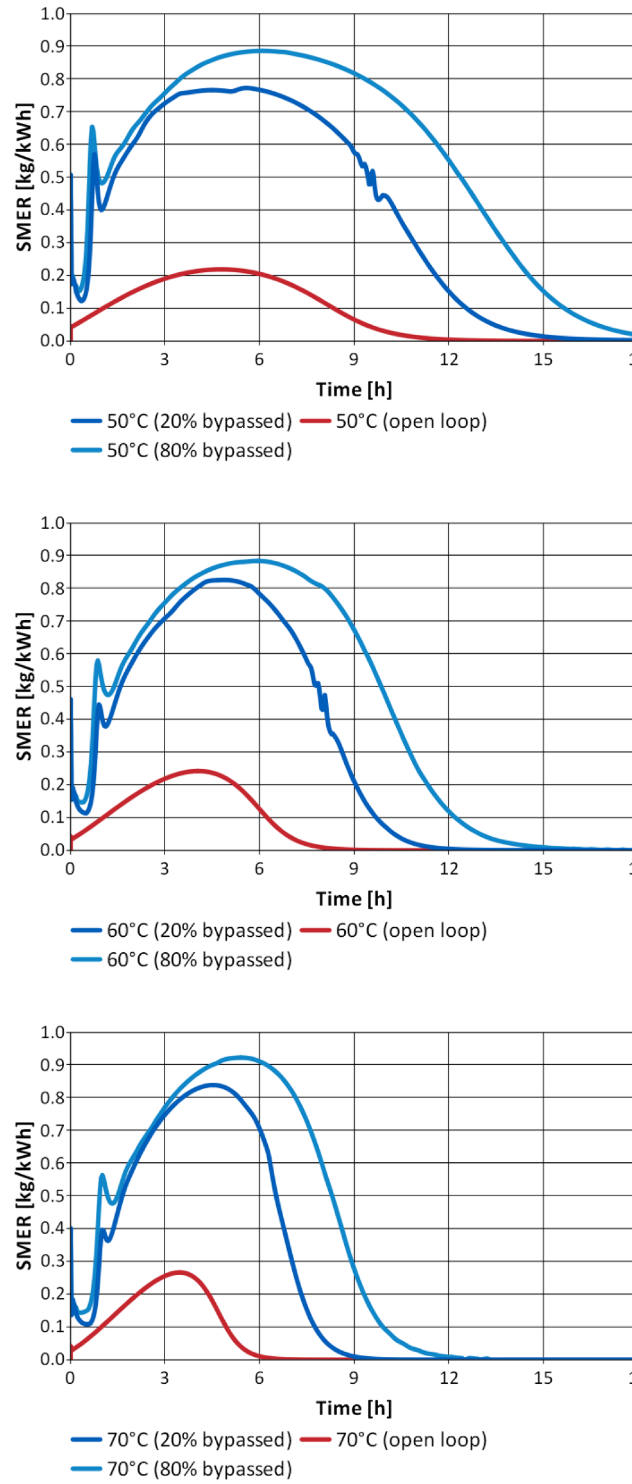

Fig. 9. Simulation results for the specific moisture extraction ratio (SMER) for varied drying temperatures and bypass ratios.

defined as the ratio between the heat being transferred to the moist air in the gas cooler and the compressor load, the following COP values are obtained from the simulation results: At 50 °C, the COP is approximately 2.6 and 2.1 (20 and 80% bypass ratio, respectively). At 60 °C, the COP is 2.0 and 1.6. At 70 °C, the COP is decreased again, with approximately 1.7 and 1.4.

The COP decrease can be explained by the simplified control strategy for the heat pump. The compressor is controlled by the air temperature set point as described in Section 2.4. This leads to higher pressure levels on CO_2 side than actually needed. Furthermore, the lack of an internal heat exchanger on CO_2 side results in further losses. Consequently, the heat pump is being operated in a non-optimal operation mode. The COP decreases with increased drying temperature, as the high pressure rises as well in that case. This results in increased heat losses in the second gas cooler.

Fig. 8 shows a typical CO_2 heat pump cycle in a temperature-entropy diagram. With the applied methodology described above it is apparent that the losses via the second gas cooler can be quite high depending on the system conditions and need to be considered in the energetic evaluation.

The efficiency of the drying process is evaluated through the specific moisture evaporation rate (SMER), which states how much water can be removed from the product with the utilization of 1 kWh. The SMER is defined by Eq. (9).

$$SMER = \frac{amount\ of\ evaporated\ water\ (kg)}{energy\ used\ (kWh)} \tag{9}$$

Fig. 9 shows the simulation results for the SMER values. During the first hour, the transient peak and drop of the SMER result from the sensitive controller of the compressor, when the air temperature in the drying chamber settles down towards the drying temperature set point.

The maximum SMER values for a drying temperature of 50 °C are 0.22 kg/kWh for the open loop dryer and 0.78 and 0.89 for the 20% and 80% bypass heat pump dryers, respectively. For 60 °C the SMER for the open loop system is 0.24 kg/kWh and 0.83 and 0.89 for the heat pump-assisted systems. For 70 °C, 0.27 kg/kWh is the highest SMER value for the open loop case, while the heat pump systems offer up to 0.84 and 0.93, respectively. For all three investigated cases, the heat pump systems offer a 3–4 times higher SMER value than the open loop system.

It shall be noted that the results for the SMER are generally low in comparison to typical SMER values found in literature [6,20]. The latent heat of water evaporation is 0.63 kWh (2250 kJ). This amount of energy is required to evaporate 1 kg of water, leading to a reciprocal value of 1.6 kg/kWh. Therefore, Fig. 9 shows that the SMER for the open loop benchmark system is already quite low. This indicates rather inefficient drying conditions that were measured and used to validate the model in the first place. The usage of a batch dryer generally appears to be a rather inefficient way for apple drying, as cabinet dryers offer only limited throughput and drying often occurs not to be uniform throughout the drying space [5].

4. Conclusion and further work

A dynamic model for the drying cabinet was developed in Modelica and successfully validated towards measurement data. The model was used to demonstrate the potential energy saving effects for drying processes at common food drying temperatures. The simulations showed that the desired drying conditions can be reached with the suggested R744 heat pump system. It was shown that the utilization of a heat pump to provide heating and cooling for a drying process can decrease the energy demand by 77–84% while offering a SMER value that is 3–4 times higher than for open loop systems. However, this results in an increased drying time by 29–69%. These results were achieved with a non-optimal controlled CO_2 heat pump and a constant uncontrolled bypass ratio.

The simulation results allowed better understanding of the design parameters and drying characteristics and enable more sophisticated dimensioning and design of future drying systems. Several adjustments of the heat pump system hold great potential for further efficiency improvements, especially regarding more sophisticated control strategies.

The CO_2 heat pump needs to be controlled at a more efficient operating point. Furthermore, the heat transfer properties between CO_2 and moist air are a limitation towards the overall system efficiency. The implementation of two intermediate water or glycol sub-cycles between the CO_2 and moist air cycle could enhance the overall heat transfer and provide more control in adjusting the dryer inlet temperature, with the utilization of water tanks to store heat and cold, respectively.

As for the drying process, a more flexible adjustment of drying temperature and humidity offers further potential for both energetic and product quality improvements. The various phases of a typical drying process need to be taken into consideration, with their differing demand for particularly hot air during the beginning and particularly dry air during the time period of constant drying rate. A variably controlled bypass in the air cycle can be used to adjust the humidity level in the drying cabinet for both drying rate reasons and to prevent surface cracking for product quality reasons.

CrediT authorship contribution statement

Michael Jokiel: Data curation, Formal analysis, Investigation, Methodology, Software, Validation, Visualization, Writing - original draft, Writing - review & editing. **Michael Bantle:** Conceptualization, Funding acquisition, Investigation, Project administration, Resources, Supervision. **Christian Kopp:** Data curation, Investigation, Software, Visualization, Writing - original draft. **Espen Halvorsen Verpe:** Data curation, Formal analysis, Investigation, Validation, Visualization, Writing - original draft, Writing - review & editing.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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