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Influence of drum inlet air conditions on drying process in a domestic tumble dryer

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Abstract

This study examines how the inlet air temperature, relative humidity and flow rate influence the textile drying process in an open cycle tumble dryer. An experimental setup was prepared by connecting a domestic tumble dryer to an external system for controlled heating, humidification and transport of air. Experiments were conducted by drying cotton textiles (8 kg dry mass) at different air inlet conditions. On the basis of measured data, correlations for determination of the total drying time, the moisture evaporation rate during the constant drying rate and the area-mass transfer coefficient were developed. The process in the drum was modeled by using an established moisture evaporation model, based on sorption isotherms. A commonly used and a recently reported sorption isotherm for cotton were used with the model. Agreement between calculated and measured drying curves was better in case of the commonly used sorption isotherm, but final moisture content was better predicted by the recently reported sorption isotherm.

1. Introduction

Electric laundry dryers are becoming increasingly popular in modern households. In the EU, trends show a growing number of sold units per year and a reduced energy consumption of the average sold unit [1]. Integration of a heat pump into a closed-cycle dryer enabled energy consumption to drop by more than 50% compared to a conventional condenser dryer with electrical heater [1]. Besides energy efficiency, other dryer parameters such as drying capacity, condensation efficiency and noise have also been improved. However, trends in drying time are more difficult to detect. Reduction of drying time is achieved by increasing heat and mass transfer rates, which is a challenging task in both conventional and heat pump tumble dryers.

Tumble dryers function by blowing heated air into a rotating drum, where textiles are located. The tumble dryers can be roughly divided in two types: 1) a vented or open-cycle type, where moist air from the drum is exhausted to ambient and 2) a condenser type, where drying air circulates in a closed cycle and a condenser is included for continuous dehumidification of the air. In conventional dryers the condenser is cooled by air, whereas in heat pump dryers it is cooled by evaporation of

coolant. In any case, the contact between the textiles and the air is of great importance in order to attain an efficient drying process. However, its optimization is a difficult task due to the complex motion and interaction of textile pieces in the rotating drum. A more straightforward approach to reaching higher drying rates is to raise the parameters of the inlet air. Increasing the drying air temperature and flow rate and decreasing its relative humidity results in better drying kinetics and therefore shorter drying times. Measurement and analyses of these effects was the main goal of the study presented in this paper.

Several authors have studied tumble dryers in order to find ways for reducing energy use or drying time. Lambert et al. [2] made a model of the tumble dryer that agreed well with the experimental data. The model was fitted by manipulating transfer coefficients for the constant drying rate period and sorption-isotherms for the falling rate drying period. Many researchers adopted the sorption-isotherm approach when developing their models. Conde [3] used a similar model to study options for heat recovery in the exhaust air. Deans [4] developed a model to investigate effects of fabric type, operating conditions and air recirculation on the vented dryer performance. He concluded that energy consumption rates were primarily influenced by the ambient air temperature and its relative humidity, whereas the drying time was controlled by the mass of the drying load and the power supplied to the dryer heater. Similar findings were reported by Do et al. [5] but based on experiments conducted on a condensing type tumble dryer with an air-to-air condenser. Highest reduction of drying time was achieved by increasing the heater power, whereas the drying air flow rate and the cooling air flow rate had a significantly smaller effect on drying performance. Yadav and Moon [6] conducted a theoretical and experimental analysis of a vented tumble dryer and developed a simulation model of the dryer. They used experimental results to evaluate the effect of different parameters on the dryer energy efficiency and found the mass per unit area of the clothes to have the most significant effect. Bassily and Colver [7] conducted 32 experiments on a vented tumble dryer to evaluate its energy performance by changing operating parameters including heater power, fan speed, drum speed, weight, and initial moisture content of the clothes. Based on these experiments Bassily and Colver [8] developed a correlation for the area mass transfer coefficient and concluded that the area of mass

transfer is a function of the weight of clothes and drum speed while the mass transfer coefficient is a function of the air flow rate, the relative humidity at the dryer outlet, the dry bulb temperature at the dryer outlet, and the material of clothes. Stawreberg and Nilsson [9] established a model based on 19 experiments in a condensing tumble dryer to be used for improving the energy efficiency and leakage ratio. The results showed that a high power supply to the heater, a high internal (drying) airflow and a low external airflow (for cooling the condenser) give the highest energy efficiency values. Stawreberg and Nilsson [10] developed a mathematical model for studying alternative control strategies for the venting tumble dryer in order to increase the energy efficiency of drying small loads. Optimized control strategy resulted in improved energy efficiency for small loads, but could not reach the same energy efficiency as for the maximum load. Huelsz et al. [11] conducted a total energy balance for venting tumble dryers to demonstrate energy flows and potential areas for energy savings in the dryers. Wei et al. [12] developed a prototype vented tumble dryer with individual control of heater power, air flow rate and drum speed. By optimizing these parameters for each of the drying stages they managed to reduce energy consumption and improve fabric smoothness.

Many recent studies were aimed at improving the condensing tumble dryer performance by including novel designs for different components. Bansal et al. [13] showed that increased energy efficiency and reduced drying time can be achieved by replacing the electrical heater with a heat exchanger that utilizes hot water to heat the air. Cochran et al. [14] demonstrated improvements in dryer efficiency by replacement of the air-to-air plate heat exchanger condensing surface with a surface tension element. Jian and Zhao [15] carried out experiments to study the drying performance of a condensing tumbler clothes dryer with a unique water cooled plate-fin heat exchanger under different working conditions. Rezk and Forsberg [16] showed benefits of air duct geometry optimization by using CFD simulations. Stawreberg and Nilsson [9] and Stawreberg et al. [17] measured and modelled air leakage in a condensing tumble dryer. They conclude that the main amount of leakage is found between the heater and the drum. Benefits of including a heat pump to the dryer have been first calculated by Braun et al. [18]. A more recent paper by Bengtsson et al. [19] reports modelled and measured effects of different heat pump compressor sizes on the dryer energy performance. Heat

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3 pump tumble dryers were recently also investigated by Erdem and Heperkan [20], who modelled
4 effects of using CO₂ as a refrigerant, and Ganjehsarabi et al. [21], who conducted an exergoeconomic
5 analysis of the dryer.
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9 A vast majority of the reviewed studies combines mathematical modelling of the tumble drying
10 process with experimental data. Probably the most difficult part of the process with regard to
11 modelling is the heat and mass transfer process in the drum, which decides the rate of moisture
12 evaporation from the textiles. This process is most commonly modelled by using several
13 simplifications and an approach introduced by Lambert et al. [2], which requires data on the sorption-
14 isotherms of relevant textiles and the textile-to-air mass transfer coefficient. This coefficient is
15 unique to the dryer design and operating parameters and therefore has to be determined by
16 measurements. The aim of this study is to relate this coefficient and the total drying time to different
17 air conditions at drum inlet. In contrast to most existing studies, the experiment was conducted on a
18 modern domestic tumble dryer with parameters, relevant for the current and future generations of
19 tumble dryers. In this respect, our study involved a large capacity dryer with a large load (8 kg dry
20 mass) with drying air parameters being representative of the conditions that are present in the heat
21 pump tumble dryers. The selection of a large capacity heat pump tumble dryer as a technology of the
22 near future was based on trends observed in the EU [1], which show that the capacity and energy
23 efficiency of an average sold tumble dryer is continuously increasing.
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40 **2. Experimental setup**

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42 The tumble dryer used for this investigation was a modified commercial heat pump model with a 9
43 kg dry load capacity. For the purpose of this study however, the built-in heat pump was removed and
44 an external, separate system for air conditioning was attached. This enabled independent setting of
45 drum inlet temperature, relative humidity and flow rate. Essentially this means that the tumble dryer
46 was converted to a vented-type dryer, functioning in an open air cycle with constant drum inlet air
47 conditions.
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The basic structure of the experimental system is shown on Fig. 1. The air conditioning system consisted of three main units: a radial fan, electrical heaters and a steam humidifier. The radial fan was used in series with the tumble dryer's internal fan to transport the air through the system. The internal fan was driven by the dryer's AC motor at constant rpm, whereas the external fan was driven by a variable frequency drive and was used to control the actual air flow rate through the dryer. The electrical heaters had a total power of 4,8 kW and were manually controlled by setting supply voltage on an autotransformer. Humidification of the air was performed by using a Defensor Mk5 steam humidifier with nominal capacity of 5 kg/s steam. Control of steam production was done manually by setting control voltage for the humidifiers internal proportional controller. The produced steam was led through a flexible hose into the main pipe, where it mixed with the hot air. Mixing was promoted by a static mixer element, installed in the pipe downstream. When highest air temperatures and relative humidities, together with high air flow rates were required, the heating and humidification system capacity was insufficient and a partial recirculation of air was implemented (Fig. 1). By recirculating warm and moist air from the drum outlet it was possible to further increase the inlet air temperature and humidity and in this way measure the entire range of operating points.

The tumble dryer was modified for the experiment by removing the heat pump and blocking the conduits that normally lead the air through heat exchangers. A connection to the external heating and humidification system was made at the back of the dryer (Fig. 2), coaxially to the dryer's internal fan. The production impeller was replaced by a 3D printed impeller to change the suction side from the usual front side to the back side, where a circular hole in the fan casing was made. The exhaust of air from the drum to ambient was made by removing the front cover, which is normally used to access the heat pump filter (Fig. 2).

2.1 Measurement system

Measurement locations are presented on Fig.1. Air temperature and relative humidity were measured at drum inlet and outlet. Air flow rate was measured at the orifice plate, installed between the fan and the heaters. To accurately determine air density at the orifice plate, air temperature was also measured upstream the orifice plate.

All temperatures were measured by using standard 4-wire Pt100 resistance temperature detectors and all relative humidities (RH) by using the Honeywell HIH-4000-003 capacitive sensors with 3.5% measurement uncertainty. Two temperature and RH sensors were installed in the air conduit upstream the drum inlet and three temperature and RH sensors were installed at drum outlet, in the air channel downstream of the lint filter. Multiple sensors were installed to compensate for the potential non-uniform conditions in the conduits.

Air flow rate was computed from the measured pressure drop at the orifice plate according to the ISO standard [22]. The Endress+Hauser Deltabar PMD235-KUBA1EA1C pressure transducer was used to measure the pressure drop. Air density was determined from the measured temperature at the orifice plate and from ambient pressure.

Mass of wet textiles before and after drying was determined by weighing it on the Kern FKB 15K0.5A scale with a resolution of 0.5 g.

All sensors were connected to the National Instruments Fieldpoint modular I/O system. Data was sampled once every second and communicated to a PC through the RS-232 connection. Data monitoring, processing and storage was performed on a PC by using the LabView software.

2.2 Measurement procedure

The measurement procedure was identical for all the measured operating points and was defined in line with standards for tumble dryer performance measurements [23]. Before each measurement the lint filter was cleaned. Then the following steps were taken:

- 1) Starting and warming-up of the heating and humidification system with the dryer door open. The warming up was made close to the required drum inlet temperature in order to provide stable temperature conditions from the very start of drying.

- 2) Wetting of textiles to $60\% \pm 1\%$ moisture content. A standard mix of cotton textiles (sheets, pillowcases, hand-towels) [23] with 8032 g conditioned dry mass was used for the experiment. The conditioned dry mass was determined according to [23] as an equilibrium moisture content for

textiles placed in a chamber, where ambient temperature and relative humidity were maintained at $20\pm 2^{\circ}\text{C}$ and $65\pm 5\%$, respectively. Wetting to the initial moisture content was done in a washing machine to assure uniform distribution of moisture in textiles.

3) Folding and loading of textiles according to Gorenje company in-house procedures. Special care was taken to always fold the textile pieces in the same manner and then place them in the drum in the sequence that assured an even distribution of sheets, pillowcases and hand-towels over the drum volume. The folding and distributing the textile pieces was performed in order to minimize tangling of textiles during drying.

4) Closing of the drum and immediate start of drying with a modified program that kept the drum rotation at constant direction. The automatic program stop was disabled.

5) Maintaining operating parameters for inlet air temperature, relative humidity and flow rate. The fan rotational speed was set at the start of drying and was not changed until the end of drying. This resulted in 1-minute averaged volume flow rates to deviate up to $\pm 4\%$ of the average value, which was assumed to be sufficient. However, manual adjustments to heater power and humidifier capacity were done during the drying cycle to maintain constant conditions. Minor adjustments were mostly needed to compensate for variations in flow rate that occurred primarily as a consequence of dryness-dependent textile volume and consequently pressure drop in the drum. More significant adjustments were required in case of partial air recirculation, where drum outlet conditions affected drum inlet conditions. Generally, it was possible to control the 1-minute averages of drum inlet temperature and RH within $\pm 0.5\text{ K}$ and $\pm 1\%$ of the set values, respectively.

6) Stopping the drying cycle. The drying was considered completed when the difference in air temperature at the drum inlet and outlet reached 8 K . In that instant the drying program was manually stopped and textiles were immediately taken out of the drum and weighed. Drying cycle was considered valid if final moisture content of textiles was $0\pm 3\%$, which is according to [23]. The temperature difference criteria was decided on the basis of preliminary tests, where it was determined that the required final textile moisture was typically achieved at 8 K difference.

2.3 Operating points

Each operating point was defined by three parameters of air at the drum inlet: normed volumetric flow rate $\dot{V}_n$, temperature T_{in} and relative humidity RH_{in} . The measured volume flow rates were multiplied by the ratio of measured air density, determined at conditions upstream orifice plate, and reference density of 1.2 kg/m³. The resulting normed volume flow rates $\dot{V}_n$ eliminate the effect of variable air density and could also be easily transformed to mass flow rates. The following air parameter values were considered to be relevant for the modern heat pump tumble dryers and were therefore chosen for the experiment: $\dot{V}_n = 230 \text{ m}^3/\text{h}, 260 \text{ m}^3/\text{h}, 290 \text{ m}^3/\text{h}$; $T_{in} = 55 \text{ }^\circ\text{C}, 60 \text{ }^\circ\text{C}, 65 \text{ }^\circ\text{C}$; $RH_{in} = 6 \%, 13 \%, 20 \%$. The parameter values were defined on the basis of data, provided by the manufacturer as state of the art, however, similar values are reported in studies that deal with heat pump tumble dryers [19 - 21]. By combining all the values a total of 27 different operating points was obtained. All the operating points were measured at least once.

3. Model

3.1 Water balance calculation

The measured data was first evaluated by performing water balance calculations. The water evaporation rate in the drum can be expressed from the difference of outflowing and inflowing water flow rates:

$$\dot{m}_{ev} = \dot{m}_{out} - \dot{m}_{in} \quad (1)$$

Mass flow of water at drum inlet and drum outlet is calculated by multiplying mass flow of dry air with the corresponding humidity ratio:

$$\dot{m} = \frac{\dot{V}\rho}{1+x_a} x \quad (2)$$

Humidity ratio is obtained from the measured temperatures and relative humidities by applying the following relations:

$$x = 0.622 \frac{p_w}{(p_a - p_w)} \quad (3)$$

$$p_w = p_{w,sat}RH \quad (4)$$

The water vapor saturation pressure was calculated according to the Antoine equation [24]:

$$p_{w,sat} = 10^{\left(10.196213 - \frac{1730.63}{233.426 + T}\right)} \quad (5)$$

Combination of Eq. (1) and (2) gives:

$$\dot{m}_{ev} = \frac{\dot{V}\rho}{1+x_a}(x_{out} - x_{in}) \quad (6)$$

Humidity ratios for air at drum inlet (x_{in}), drum outlet (x_{out}) and for ambient air (x_a) are obtained from Eq. (3)-(5). Since measured data was sampled once every second, the actual mass of evaporated water for each second can be obtained by multiplying Eq. (6) by $t = 1s$. Summation of evaporated water quantities over selected time gives the total mass of evaporated water:

$$m_{ev} = \sum_{i=1}^{t_{end}} \frac{\dot{V}_i \rho_i}{1+x_{a,i}}(x_{out,i} - x_{in,i}) t \quad (7)$$

3.2 Constant drying rate

A constant drying rate period, indicated by nearly constant air temperature and relative humidity at drum outlet, can be observed for all of the measured cases. The constant drying rate period is established after a short initial warming-up period and typically lasts for about one half of the total drying time. A period between 500 and 1500 seconds drying time was selected as representative for further analysis since this time interval corresponds to the constant drying rate for all the measured cases. The maximum and average increase in drum outlet temperature during this period, observed among all the measured cases, was 1.8 K and 0.75 K, respectively.

3.3 Moisture evaporation model

The most common approach for modelling of moisture evaporation rate from textiles, reported in literature [2, 10, 19], is by using the mass transfer equation in the following form [25]:

$$\dot{m}_{ev} = h_m A_t (p_t - p_{in}) \quad (8)$$

The product of the moisture transfer coefficient and the textile area, $h_m A_t$, is specific to the operating conditions such as the airflow rate, the drum speed, the drying load and on the distribution and motion of textiles in the drum. It is most commonly determined experimentally [8, 10, 19], but could also be calculated from the Lewis number [4, 6]. In this study the $h_m A_t$ was determined from measured data for each operating point by taking average values of $\dot{m}_{ev}$, p_t and p_{in} in a 1000-second interval during the constant drying rate period.

The difference in vapor pressures at textile surface and drum inlet ($p_t - p_{in}$) represents the driving force for moisture transfer. Instead of vapor pressures, some researchers [3, 4, 6, 8] use the corresponding difference in humidity ratios. Here, the vapor pressure at drum inlet, p_{in} , was calculated from measured temperature and relative humidity according to Eq. (4) and (5). The vapor pressure at textile surface, p_t , was determined from the relations presented by Lambert et al. [2]:

$$p_t(X_t, T_t) = p_{sat}(T_t) a(X_t) \quad (9)$$

Where the water activity, a , was

$$a(X_t) = \frac{\beta X_t + \delta}{1 + \delta(\gamma X_t)} \quad (10)$$

The temperature of textiles, T_t , was assumed to be equal to the outlet air temperature. Eq. (10) is an approximation of the sorption isotherm and depends on the type of fabrics being dried. For cotton it is common to use the coefficients introduced by Lambert et al. [2]: $\beta = 18$, $\gamma = 30$, $\delta = 2$. During the heating and the constant drying rate periods when the water content in textiles (X_t) is still high, the water activity equals unity. The surface of the cotton fabric is saturated and the corresponding vapor pressure, p_t , is equal to the saturation vapor pressure. Only in the later stages of the drying process, when the water content in textiles has dropped to around 30%, the water activity starts to decrease notably. This results in lower vapor pressure at textile surface (Eq. 9) and consequently lower moisture evaporation rates (Eq. 8), which is typical for the falling rate drying period.

The remaining moisture content in textiles is determined by subtracting evaporated water mass from the initial water mass:

$$X_t = \frac{m_w - \sum \dot{m}_{ev} t}{m_t} \tag{11}$$

3.4 Drying time correction

Tests were considered valid when the moisture content of textiles at the end of drying laid in the range between -3% and +3%. Also, the initial moisture content could be within 60%±1%, therefore the evaporated water mass varies from test to test and the actual drying time as seen in the drying curves (Figs. 3-5) is not directly comparable. For this reason a correction of the measured drying time is performed according to the standard [23]. The corrected drying time takes into account the nominal mass of water that would be evaporated when drying 8 kg of textiles from 60 % to 0 % moisture content and the actual evaporated mass of water:

$$t_{cor} = t \frac{m_{w,nom}}{m_w} = t \frac{0.6 \cdot 8 \text{ kg}}{m_w} \tag{12}$$

4. Results and discussion

4.1 Measured drying curves

A total of 27 different operating points with varied drum inlet conditions were measured to analyze the influence of volume flow rate, inlet air temperature and inlet air humidity on textile drying kinetics. The following figures compare measured air conditions at drum outlet (T , RH) and textile moisture content, computed from Eq. (7), for sets of three different operating points. The influence of volume flow rate is presented on Figure 3 for cases with constant $T_{in}=60^{\circ}\text{C}$ and $RH_{in}=13\%$. The temperature curves indicate a short initial warm-up period, followed by a period of nearly constant temperature. This period can be regarded as the constant drying rate period, despite the trends of rising temperatures and falling humidities actually indicate a slightly falling drying rate. The

influence of volume flow rate is visible throughout the drying time, which is most clearly reflected in the computed textile moisture, which takes into account both the air temperature, relative humidity and flow rate. Influence of flow rate on the outlet air thermodynamic state is evident mostly in the air humidity levels, whereas the air temperature levels remain unaffected. Figure 4 presents the influence of inlet air temperature by comparing 3 cases with constant $\dot{V}_n=260 \text{ m}^3/\text{h}$ and $RH_{in}=13\%$. The higher inlet air temperature results in higher outlet air temperature and lower outlet air relative humidity. The thermodynamic state of outlet air does not clearly indicate the positive effect of higher inlet air temperature, however the computed textile moisture curves and the total drying time reflect the higher drying rates. Figure 5 shows the influence of inlet air relative humidity on drying by comparing 3 cases with constant $\dot{V}_n=260 \text{ m}^3/\text{h}$ and $T_{in}=60 \text{ }^\circ\text{C}$. The measured outlet air parameters indicate a constant drying rate period with significant differences in both temperature and humidity levels. The case with the highest inlet relative humidity also has the highest outlet relative humidity and temperature levels and has the lowest water evaporation rate, as reflected by the textile moisture curves. Consequently, the total drying time is strongly dependent on the inlet air relative humidity.

The measured outlet air temperature and relative humidity curves (Figs. 3-5) reflect the set inlet air conditions and more importantly, the textile drying kinetics in the drum. The trends in curves are similar as in other reported experiments on tumble dryers [3-7,10,11], but also specific due to the fact that the inlet air conditions were kept constant by an external air conditioning system. Also specific to the presented data is the presence of high oscillations in the measured data, especially RH . However, such oscillations are a normal consequence of a highly dynamic textile motion in the drum that affects the heat and mass transfer processes. The process dynamics could be well captured due to a low response time of the RH sensors and a relatively fast sampling rate.

When comparing the resulting $X_i(t)$ curves from different tests it has to be noted, that the average inlet conditions could vary slightly within groups of tests that were supposed to have identical conditions. Furthermore, the initial and final textile moisture content also varied within specified intervals, therefore the total mass of evaporated water was different in each test. To compare the

influence of inlet air conditions on drying it is therefore necessary to choose parameters that take these effects into account.

4.2 Drying time

Table 1 contains the corrected drying time (Eq. 12) per mass of dry textiles for all the operating points. A correlation for prediction of drying time as a function of process parameters was developed. The correlation is based on data from Table 1 and time averaged inlet air conditions (T_{in} , RH_{in} , $\dot{V}_n$). An exponential function was chosen for the drying time correlation as it provided a good fit to data. The coefficients were determined numerically by using the least squares method. No outliers were found in the data and therefore all the operating points were taken into account. The resulting correlation is defined as

$$t = 39434 \dot{V}_n^{-0.9065} T_{in}^{-0.8338} RH_{in}^{0.1598} \quad (\text{min/kg}) \tag{13}$$

Values of corrected measured time and calculated time according to Eq. (13) are compared on Figure 6. Scatter of points around the line of equality is small, which indicates a good fit between measured and modelled data. This is confirmed by the high value of coefficient of determination (R^2) for the model, which in this case is 0.968.

4.3 Moisture evaporation model – constant drying rate

The moisture evaporation model, described in detail in chapter 3.3, enables calculation of water evaporation rate for the entire drying cycle. The key unknown parameter in the model is the product of the moisture transfer coefficient and the textile area, $h_m A_t$. The $h_m A_t$ was calculated from Eq. (8) for each operating point by taking average values of $\dot{m}_{ev}$, p_t and p_{in} during the constant drying rate period. Table 2 contains the resulting values of $h_m A_t$ as well as the average values of $\dot{m}_{ev}$, that were determined from measured data according to Eq. (6) and were required to calculate the $h_m A_t$.

It is difficult to compare the resulting $h_m A_t$ values to other reported research, since each study involved different dryer constructions, textile loads and air parameters. Perhaps the most comparable in this respect is a recent study [19] that employed a heat pump tumble dryer with the following

parameters: 5.3 kg of dry textiles, 200 m³/h air volume flow, 45-65 °C air temperature. In this case the $h_m A_t$ was determined at 3.6e-7 kg/sPa, which corresponds very well to the here determined values.

Correlations for both the $\dot{m}_{ev}$ and the $h_m A_t$ were developed by using the same approach as in the case of the drying time model (Eq. 13). The resulting correlations are given with the following equations:

$$\dot{m}_{ev} = 5.53 \cdot 10^{-7} \dot{V}_n^{0.8305} T_{in}^{0.8066} RH_{in}^{-0.1914} \quad (14)$$

$$h_m A_t = 9.06 \cdot 10^{-8} \dot{V}_n^{0.6824} T_{in}^{-0.5561} RH_{in}^{-0.0293} \quad (15)$$

The R^2 values for the first (Eq. 14) and the second (Eq. 15) correlation are 0.966 and 0.850, respectively. In the case of $h_m A_t$, the effect of RH in the correlation is relatively small, as indicated by the value of corresponding exponent. Removal of RH from the equation and re-calculation of coefficients resulted in Eq. (18) with the R^2 value of 0.819.

$$h_m A_t = 7.98 \cdot 10^{-8} \dot{V}_n^{0.6916} T_{in}^{-0.5545} \quad (16)$$

The simple relation between the $\dot{m}_{ev}$ and the $h_m A_t$ (Eq. 8) and steady conditions during the constant drying rate period could be used as a presumption, that both the $\dot{m}_{ev}$ and $h_m A_t$ correlations will show a similar dependence on the chosen inlet air parameters. However, the R^2 values indicate that the $\dot{m}_{ev}$ correlation presents a better fit to data than the $h_m A_t$ model. Moreover, the exponent values in the $\dot{m}_{ev}$ correlation reflect the expected influences of air inlet parameters, where higher water evaporation is attained by increasing the $\dot{V}_n$ and T_{in} and decreasing the RH_{in} . In case of the $h_m A_t$ correlation, the effect of temperature becomes reversed, meaning that higher temperatures reduce the moisture mass transfer coefficient. Such relation seems questionable, but has a valid explanation. The $h_m A_t$ is calculated from Eq. (8) by dividing the $\dot{m}_{ev}$ by the difference of vapor pressures at textile surface and drum inlet ($p_t - p_{in}$). This difference represents the driving force for moisture transfer and is already a function of both T_{in} and RH_{in} . Measured data shows, that ($p_t - p_{in}$) increases when T_{in} is increased at constant RH_{in} . However, the $\dot{m}_{ev}$ does not increase linearly with the increase in ($p_t - p_{in}$) and the $h_m A_t$ compensates for this.

When evaluating the $h_m A_t$ correlation it is necessary to consider the possible introduction of error due to assumptions in the moisture evaporation model. Namely, to calculate p_t it is assumed that the textile temperature is equal to the outlet air temperature and that air at textile surface is saturated during the constant drying period. This is a common approach [10, 17] since the actual textile conditions in the rotating drum are difficult to measure accurately. The $h_m A_t$ could also be affected by other factors that were not controlled or measured during the experiments, such as leakage of air and heat losses through drum walls. Last but not least, the measurement uncertainty in T and RH can have a significant influence on calculation of air thermodynamic state and all dependent quantities. However, this influence is not limited to the $h_m A_t$ but affects all data based on experiments.

4.4 Moisture evaporation model – complete drying cycle

The moisture evaporation model (Eq. 8-11) enables calculation of the instantaneous $\dot{m}_{ev}$ for the entire drying time by assuming that the $h_m A_t$ value is constant. Figures 7 and 8 compare X_t as a function of time for different operating points. The X_t is calculated according to Eq. (11), where the $\dot{m}_{ev}$ was obtained from either measurements (water balance calculation, Eq. 6) or was calculated from the moisture evaporation model (Eq. 8). The model calculations were done by using the actual $h_m A_t$ values (Table 2). Water activities (Eq. 10) were determined according to two different sorption isotherms for cotton: the sorption isotherm introduced by Lambert et al. [2] and a more recently published desorption isotherm by Bhourri et al. [26]. The sorption isotherm from [2] is used throughout the reviewed literature [2-4, 6, 18, 19] and is defined by Eq. (10) and coefficients $\beta = 18$, $\gamma = 30$, $\delta = 2$. Bhourri et al. [26] have measured sorption isotherms for cotton yarn and knitted fabric at temperatures of 25 °C and 35 °C. The desorption isotherm for knitted fabric at 35 °C was taken for this study and fitted to Eq. (10), which resulted in coefficients $\beta = 16.62$, $\gamma = 43.06$, $\delta = 2.02$. In addition to the drying curves, Figures 7 and 8 also contain the final X_t determined by weighing the textiles.

Figure 7 shows drying curves for 2 operating points, both with $T_{in}=60^{\circ}\text{C}$ and $RH_{in}=13\%$, where $\dot{V}_n=230\text{ m}^3/\text{h}$ for the first and $\dot{V}_n=290\text{ m}^3/\text{h}$ for the second point. The excellent agreement of curves

during the constant drying rate period is an expected result of determining the $h_m A_t$ with regard to the measured $\dot{m}_{ev}$. In the falling drying rate period the curves separate. Calculation with the sorption isotherm [2] predicts a shorter constant drying rate period and a higher final X_t than calculation with the sorption isotherm [26]. Similar trends can be seen on Figure 8, which compares 2 tests, both with $T_{in}=60^\circ\text{C}$ and $\dot{V}_n=260\text{ m}^3/\text{h}$, but $RH_{in}=6\%$ for the first and $RH_{in}=20\%$ for the second test.

Drying curves for all the measured operating points are not plotted due to space constraints. Instead, agreement between the curves is evaluated by calculating the R^2 values, where the measured curve is taken as a reference. The curves are further evaluated by comparing their final moisture content ($X_{t,fin}$) to the reference final moisture content ($X_{t,ref}$), determined by weighing. The data is presented in Table 3 and Figure 9. The R^2 values indicate better agreement between the measured and calculated drying curves in the case of sorption isotherm [2]. However, the resulting $X_{t,fin}$ values are in better agreement with the weighed $X_{t,ref}$ in case of the sorption isotherm [26] instead of the sorption isotherm [2]. Average difference between the $X_{t,fin}$ and the weighed $X_{t,ref}$ is smallest in the case of measured water balance, but scatter in this case is highest. The smallest scatter in $X_{t,fin} - X_{t,ref}$ can be seen in case of using the sorption isotherm [26]. The point with an 8% deviation from the $X_{t,ref}$ in the case of measured water balance could be considered as an outlier, however in all other analyses (regression of drying time and $h_m A_t$) this operating point never stood out with above average residuals, therefore it was considered valid and kept in the dataset.

Presented results confirm capability of the moisture evaporation model to predict drying curves for the entire drying time with reasonable accuracy. Current study was aimed at improving the accuracy of such predictions by modeling the $h_m A_t$ as a function of inlet air parameters and by including an alternative sorption isotherm for cotton. Sorption isotherms for cotton can vary with temperature and mass transfer direction (adsorption/desorption), as shown by Bhourri et al. [26]. The conditions at which the commonly used sorption isotherm [2] was determined are not reported, contrary to the more recent sorption isotherm [26], where temperature and mass transfer direction are exactly defined. However, comparison of measured and modelled drying curves (Table 3, Fig. 9) does not provide a universal answer to the question of which sorption isotherm performed better.

Improvements in modeling of textile drying in tumble dryers could be achieved by further investigating processes in the drum. Distribution and motion of textiles in the drum changes significantly as it gets dry, affecting aerodynamic conditions and heat and mass transfer processes. Therefore, current assumption of $h_m A_t$ being constant during the entire drying time could be an important source of error. In this respect, another factor that must be considered is occurrence of textile tangling. Textile tangling is typically detected by unexpected rises in T_{out} and simultaneous drops in RH_{out} , which means temporary lower drying rates and lower values of $h_m A_t$. It has to be noted that this study considered drying runs to be valid only when no textile tangling was detected in plots of measured data.

Optimization of the drying process in the drum affects drying time and other dryer performance parameters, such as energy consumption, fabric smoothness [12] and others. Experiments in this study were designed to measure and model drying kinetics in the drum and did not provide relevant data for analysis of energy performance. However, the developed relations should enable improved modeling of the complete tumble dryer cycle, including the heat pump process, which has a decisive influence on the energy efficiency of the complete system [19, 20].

5. Conclusions

An experimental study of textile drying in an open cycle tumble dryer was performed to analyze the influence of air inlet conditions on drying kinetics. The air parameters and the dryer capacity were chosen to be representative of the latest generations of heat pump tumble dryers. Results provide relations for modeling and optimization of the drying process, primarily in form of reducing the relatively long drying times that are typical for the heat pump tumble dryers. Alternatively, they enable developers to estimate the effects of heat pump process optimizations on the textile drying kinetics.

Results of the experiment enabled description and modeling of the drying process that was based on the established models, but with several novel contributions. A correlation for prediction of drying time was developed with good fit to measured data. The correlation clearly shows the influence of

drum inlet conditions on total drying time. Higher flow rate and temperature and lower relative humidity of drum inlet air can be used effectively to reduce drying time. Furthermore, the constant drying rate period was analyzed to determine the area-mass transfer coefficient ($h_m A_t$), which is needed for modeling of moisture evaporation rate during the entire drying time. The $h_m A_t$ was correlated as a function of air inlet parameters, where the flow rate and temperature showed the highest influence.

Drying curves were modeled by using two different sorption isotherms for cotton and compared to measured data. Agreement between the curves was better in case of the commonly used sorption isotherm [2], but final moisture content was better predicted by the sorption isotherm [26].

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		$\dot{V}_n=230 \text{ m}^3/\text{h}$			$\dot{V}_n=260 \text{ m}^3/\text{h}$			$\dot{V}_n=290 \text{ m}^3/\text{h}$			unit
		RH_{in}			RH_{in}			RH_{in}			
		6 %	13 %	20 %	6 %	13 %	20 %	6 %	13 %	20 %	
T_{in}	55 °C	12.7	14.2	16.2	11.3	12.8	14.8	10.5	11.6	12.8	min/kg
	60 °C	11.7	13.0	14.5	10.4	12.0	13.5	9.7	11.0	12.1	min/kg
	65 °C	11.4	12.1	13.9	10.0	11.5	12.9	9.1	10.4	11.1	min/kg

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		$\dot{V}_n=230 \text{ m}^3/\text{h}$			$\dot{V}_n=260 \text{ m}^3/\text{h}$			$\dot{V}_n=290 \text{ m}^3/\text{h}$			unit
		RH_{in}			RH_{in}			RH_{in}			
		6 %	13 %	20 %	6 %	13 %	20 %	6 %	13 %	20 %	
		Values of $\dot{m}_{ev}$ in g/s									
T_{in}	55 °C	0.96	0.83	0.73	1.05	0.93	0.84	1.15	0.97	0.91	g/s
	60 °C	1.05	0.88	0.81	1.14	0.99	0.88	1.21	1.11	0.98	g/s
	65 °C	1.10	0.97	0.85	1.23	1.05	0.91	1.35	1.17	0.93	g/s
		Values of $h_m A_t \cdot 10^7$									
T_{in}	55 °C	3.83	3.77	3.65	4.07	3.88	3.74	4.24	4.14	4.42	kg/sPa
	60 °C	3.51	3.59	3.47	3.87	3.69	3.81	4.53	4.11	3.98	kg/sPa
	65 °C	3.53	3.38	3.22	3.68	3.64	3.20	3.96	3.96	3.90	kg/sPa

	measured (water balance)	sorption isotherm [2]	sorption isotherm [26]
min R^2		0.972	0.918
max R^2		0.999	0.997
average R^2		0.990	0.983
min $(X_{t,fin} - X_{t,ref})$	-3.2 %	1.4 %	-0.3 %
max $(X_{t,fin} - X_{t,ref})$	8.0 %	5.8 %	3.2 %
average $(X_{t,fin} - X_{t,ref})$	0.5 %	3.8 %	1.4 %

Figure & Table caption list

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